WASTE HEAT MANAGEMENT GUIDEBOOK

NBS HANDBOOK 121

U.S. DEPARTMENT OF COMMERCE/National Bureau of Standards
and
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Waste Heat Management Guidebook

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CONTENTS

WASTE HEAT MANAGEMENT GUIDEBOOK

K. G. Kreider and M. B. McNeil, Editors

Sources of waste (i.e., discarded) heat in industrial processes are reviewed, and an overview of off-the-shelf technology available for its use is given. Discussions of waste heat measurement technology and economics are included, as are 14 case studies of successful industrial waste-heat recovery installations.

Key words: Boilers; economics; energy conservation; instrumentation; recuperators; waste heat.

INTRODUCTION

Statement of the Problem

Fuel energy powers today's civilization.

We depend on low price fuel to farm the midwest plains, heat our northern cities, provide the transportation to allow our complex division of labor, and manufacture the steel and concrete on which our society is built.

The industrial sector of the Nation's economy uses over 40 percent of the total U.S. energy requirements. The economic health of the United States depends upon its ability to satisfy the energy requirements of industry. Unfortunately, while our country is experiencing increasing demands for energy, domestic oil and natural gas production is declining (8% in 1974). While coal reserves are adequate, substantial increases in production are not projected for the near term. Thus, consumers must expect increasing energy prices. Understanding the Nation's energy situation leads to the conclusion that industrial conservation of energy is extremely important [1-7].[1]

Energy conservation, in the context of this work, means reduction of energy waste or the increased efficiency of energy utilization. Conservation is not to be confused with curtailment. The purpose of this guidebook is to help the engineer or engineering manager to conserve energy by better utilization of waste heat, thereby increasing his profits.

It is often possible to maintain or increase profits by improved heat management. The first steps in such a profit-enhancing approach are overall awareness of the problem and specific management actions, such as those recommended in the NBS Handbook 115 "Energy Conservation Program Guide for Industry and Commerce" (EPIC). These management actions result in a systematic program for identifying and correcting wasteful practice in industrial plants. The EPIC program, for example, includes organizing an energy conservation team with top management commitment, taking an energy audit of the plant and its energy intensive processes, identifying energy conservation opportunities, analyzing and implementing the profitable energy conserving opportunities, and providing for a continuing program to measure energy consumption and optimize efficiency. The program plan stresses the use of energy balances on all energy intensive systems and processes and demonstrates the need for improved measurement and control of energy.

One of the benefits of the EPIC program in a plant is identification of high energy consumption areas such as furnaces and ovens where considerable energy savings may be achieved by routine "housekeeping" measures as well as by actual technological innovation. In addition to opportunities for fundamental improvements to specific industrial processes or improvements in combustion efficiency or even new industrial plant designs, EPIC type surveys also uncover potentials for recovering "waste heat." Waste heat is heat which is generated in a process but then is "dumped" to the environment even though it could still be reused for some useful and profitable purpose.

The waste heat is normally available in the form of sensible heat of the work (i.e., the extra heat content of work which is hotter than the

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[1] Figures in brackets indicate the literature references.

environment) or jacket heat losses of the furnace. It is normally released to and degraded in the surroundings and quickly loses its value. In fact, it is the value of that heat being discharged which should be identified and if possible recovered. The utilization may include a simple heat exchanger for reheating combustion air, water or reactants, or it could include the operation of a heat engine such as a waste heat boiler or turbine to obtain steam or electricity.

The purpose of this book is to help the reader assess the value of that waste heat (which depends on its form and location) and decide how that value can be used to earn dollars. Therefore, the message of this handbook, like that of the EPIC handbook, is saving energy in order to save dollars. Naturally, there may be other considerations in whether an investment in heat recovery equipment is to be made, but it is felt that a sound estimate of the financial picture for the investment is vitally important. This is the approach of this Handbook.

The problem addressed in this guide is that of converting waste heat into useful heat in such a way as to increase profits. The recovery of the value of waste heat in industrial processes is a major way in which engineering can reduce fuel bills and raise profits.

The essential quality of heat addressed in this volume is not *amount*, but *value*. This distinction becomes apparent when one compares the value of recovering similar amounts of heat at 200 F or 1000 F, or of a given amount of heat in a corrosive or in an inert gas stream. For example, if you want process steam, waste heat in clean flue gas at 1000 F is quite useful; the same heat in a dirty flow at 200 F might not be worth bothering with, and certainly would be much harder to use. The examples chosen for this book demonstrate the conversion of waste heat into useful heat needed elsewhere, with a consequent saving in fuel. Complicating factors are discussed and the economics of the true value of the heat are carefully considered. Large energy savings are possible through waste heat utilization.

In 1975, four industry groups that appeared to offer particularly favorable opportunities for waste heat utilization (paper, food, stone-clay-glass, and primary metals) were estimated to use 7.7×10^{15} Btu, or 7.7 quad/yr.[2] Results detailed later in this guide indicate that a typical plant which installs waste heat recovery equipment can save at least 20 percent on its fuel. If only 2/3 of the plants in these four classes could save 20 percent on their fuel, the saving would be 1 quad/yr, equivalent to more than 400,000 bbl/d.

The reader will ask, if such savings are available, why has waste heat recovery equipment not already been installed in every plant that can benefit from it? Basically, the reason is that when fuels were cheap and readily available, waste heat recovery was not important and payback periods for such equipment were long. Consequently, although a few industries were well aware of waste heat opportunities, the level of awareness of waste heat problems among engineers and managers in other industries was low. The increases in the prices of hydrocarbon fuels in recent years and the prospects for further increases, together with fuel allotments, have fundamentally changed the economics of waste heat recovery. The purpose of this book is to help the engineer or manager learn about available options in waste heat recovery, to decide whether he can increase profits by installing waste heat equipment, and to help him choose the best available equipment for this purpose. Emphasis is placed on the solution of problems using off-the-shelf technology. It appears that development of new waste heat technology is less important at present than efficient use of existing technology and equipment that is already available and has been tried.

How to Use the Guidebook

Chapter 1 on "Sources and Uses of Waste Heat" includes an orientation so that the reader may relate the guidebook to his particular plant or his interest. The sources discussed are separated according to the temperature of the hot exhaust. This is followed by a section on how to use waste heat, which enumerates the various possible applications. This chapter is completed by a discussion on organizing and implementing a waste heat management program.

Chapter 2 is intended to assist in making an engineering assessment of a plant's waste heat recovery situation. It includes sections on the principles behind performing a heat balance and explicit information and examples of how to make a heat balance. This is followed by a discussion of engineering problems such as stack gas

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[2] "Energy Consumption—Manufacturing" (Ballinger Publishing Co., Cambridge, Mass., 1974).

condensation which constrain the selection of the parameters of the heat recovery system. A simplified treatment of the bases for calculation of heat transfer quantities is also included. Many readers may prefer to skim this chapter and then return to it if it appears that they must use this material in their particular waste heat problems.

The engineering discussion is followed by chapter 3 on "Economics of Waste Heat Recovery." Here the basic approaches to a cost-benefit analysis are presented to allow the reader to determine the profits in applying waste heat recovery. Several techniques are described with the advantages and disadvantages of each explained so that the reader may select an approach which fits his situation. Starting with simple payback procedures the work builds toward a complete analysis incorporating income tax effects, inflation, and risk and uncertainty. Illustrative examples are used to assist the reader in calculating his own potential financial benefits.

Chapter 4 includes 14 actual case studies of companies which have recently installed profitable waste heat recovery systems. The manager may not only find interesting applications but can also discover the various approaches used in solving a number of waste heat recovery problems. Perhaps one example may be very similar to his own plant.

In order to provide more complete coverage several additional chapters have been included in the guidebook. Chapter 5 "Commercial Heat Recovery Equipment" is a rather detailed discussion of the types of equipment which can be used in waste heat systems. This chapter may be particularly useful in appraising the value of the various approaches proposed by the companies selling the equipment. The reader might begin by finding a particular type of equipment on the chart on page 153 and then knowing the general parameters of performance read the description in the chapter of how it works and what to expect.

The editors have also included chapter 6 on instrumentation for measuring quantities of importance in waste heat management. These measurements relate directly to determining the heat balance of the operation which was explained in chapter 2. Temperature, pressure, fluid flow, combustion chemistry, and radiative heat flow measurement techniques are discussed.

Chapter 7 "Engineering Data for Waste Heat Recovery" has been chosen by the authors to include that information which would simplify the reader's task. In summary, the first part of the guidebook outlines and describes the necessary steps an engineer should accomplish in demonstrating the value of waste heat recovery to himself and to his management. Part two, the case studies, helps him to relate the experience of others with similar problems to his own. The editors feel part three is also very useful in that the specific information on the hardware that will be installed and information on how to make the measurements that are needed for the heat balance are critical inputs in achieving a good waste heat recovery system.

Because English units (e.g., Btu's) are still widely used in production work, the guidebook is written in those units. For the convenience of readers who have converted to SI metric units, a conversion table (7.7) is at the end of chapter 7. An index of notation is given in the appendix.

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Chapter 1

SOURCES AND USES OF WASTE HEAT

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Definitions

Waste heat has been defined as heat which is rejected from a process at a temperature enough above the ambient temperature to permit the manager or engineer to extract additional value from it. Sources of waste energy can be divided according to temperature into three temperature ranges. The high temperature range refers to temperatures above 1200 F. The medium temperature range is between 450 F and 1200 F, and the low temperature range is below 450 F.

High and medium temperature waste heat can be used to produce process steam. If one has high temperature waste heat, instead of producing steam directly, one should consider the possibility of using the high temperature energy to do useful work before the waste heat is extracted. Both gas and steam turbines are useful and fully developed heat engines.

In the low temperature range, waste energy which would be otherwise useless can sometimes be made useful by application of mechanical work through a device called the heat pump. An interesting application of this is in petroleum distillation, where the working fluid of the heat pump can be the liquid being distilled. (This application was developed by the British Petroleum Co.)

Sources of Waste Heat

The combustion of hydrocarbon fuels produces product gases in the high temperature range. The maximum theoretical temperature possible in atmospheric combustors is somewhat under 3500 F, while measured flame temperatures in practical combustors are just under 3000 F. Secondary air or some other dilutant is often admitted to the combustor to lower the temperature of the products to the required process temperature, for example to protect equipment, thus lowering the practical waste heat temperature.

Table 1.1 below gives temperatures of waste gases from industrial process equipment in the high temperature range. All of these result from direct fuel fired processes.

TABLE 1.1

Type of Device	*Temperature F*
Nickel refining furnace	2500 - 3000
Aluminum refining furnace	1200 - 1400
Zinc refining furnace	1400 - 2000
Copper refining furnace	1400 - 1500
Steel heating furnaces	1700 - 1900
Copper reverberatory furnace	1650 - 2000
Open hearth furnace	1200 - 1300
Cement kiln (Dry process)	1150 - 1350
Glass melting furnace	1800 - 2800
Hydrogen plants	1200 - 1800
Solid waste incinerators	1200 - 1800
Fume incinerators	1200 - 2600

Table 1.2 gives the temperatures of waste gases from process equipment in the medium temperature range. Most of the waste heat in this temperature range comes from the exhausts of directly fired process units. Medium temperature waste heat is still hot enough to allow consideration of the extraction of mechanical work from the waste heat, by a steam or gas turbine. Gas turbines can be economically utilized in some

cases at inlet pressures in the range of 15 to 30 lb/in^{2}g. Steam can be generated at almost any desired pressure and steam turbines used when economical.

TABLE 1.2

Type of Device	*Temperature F*
Steam boiler exhausts	450 - 900
Gas turbine exhausts	700 - 1000
Reciprocating engine exhausts	600 - 1100
Reciprocating engine exhausts (turbocharged)	450 - 700
Heat treating furnaces	800 - 1200
Drying and baking ovens	450 - 1100
Catalytic crackers	800 - 1200
Annealing furnace cooling systems	800 - 1200

Table 1.3 lists some heat sources in the low temperature range. In this range it is usually not practicable to extract work from the source, though steam production may not be completely excluded if there is a need for low pressure steam. Low temperature waste heat may be useful in a supplementary way for preheating purposes. Taking a common example, it is possible to use economically the energy from an air conditioning condenser operating at around 90 F to heat the domestic water supply. Since the hot water must be heated to about 160 F, obviously the air conditioner waste heat is not hot enough. However, since the cold water enters the domestic water system at about 50 F, energy interchange can take place raising the water to something less than 90 F. Depending upon the relative air conditioning lead and hot water requirements, any excess condenser heat can be rejected and the additional energy required by the hot water provided by the usual electrical or fired heater.

TABLE 1.3

Source	*Temperature F*
Process steam condensate	130 - 190
Cooling water from:	
Furnace doors	90 - 130
Bearings	90 - 190
Welding machines	90 - 190
Injection molding machines	90 - 190
Annealing furnaces	150 - 450
Forming dies	80 - 190
Air compressors	80 - 120
Pumps	80 - 190
Internal combustion engines	150 - 250
Air conditioning and refrigeration condensers	90 - 110
Liquid still condensers	90 - 190
Drying, baking and curing ovens	200 - 450
Hot processed liquids	90 - 450
Hot processed solids	200 - 450

How to Use Waste Heat

To use waste heat from sources such as those above, one often wishes to transfer the heat in one fluid stream to another (e.g., from flue gas to feedwater or combustion air). The device which accomplishes the transfer is called a heat exchanger. In the discussion immediately below is a listing of common uses for waste heat energy and in some cases, the name of the heat exchanger that would normally be applied in each particular case. Commercially available types of waste heat exchangers are reviewed in chapter 5.

The equipment that is used to recover waste heat can range from something as simple as a pipe or duct to something as complex as a waste heat boiler. Chapter 5 categorizes and describes some waste recovery systems that are available commercially suitable for retrofitting in existing plants, with lists of potential applications for each of the described devices. These are developed technologies which have been employed for years in some industries.

(1) Medium to high temperature exhaust gases can be used to preheat the combustion air for:

Boilers using air-preheaters
Furnaces using recuperators
Ovens using recuperators
Gas turbines using regenerators

(2) Low to medium temperature exhaust gases can be used to preheat boiler feedwater or boiler makeup water using *economizers*, which are simply gas-to-liquid water heating devices.

(3) Exhaust gases and cooling water from condensers can be used to preheat liquid and/or solid feedstocks in industrial processes. Finned tubes and tube-in-shell *heat exchangers* are used.

(4) Exhaust gases can be used to generate steam in *waste heat boilers* to produce electrical power, mechanical power, process steam, and any combination of above.

(5) Waste heat may be transferred to liquid or gaseous process units directly through pipes and ducts or indirectly through a secondary fluid such as steam or oil.

(b) Waste heat may be transferred to an intermediate fluid by heat exchangers or waste heat boilers, or it may be used by circulating the hot exit gas through pipes or ducts. Waste heat can be used to operate an absorption cooling unit for air conditioning or refrigeration.

Organizing a Waste Heat Management Program

Every plant has some waste heat. A waste heat management program, that is, a systematic study of the sources of waste heat in a plant and opportunities for its use, would normally be undertaken as part of a comprehensive energy conservation program.

The organization and management of a waste heat recovery program is an integral part of the overall energy conservation program, but the engineering effort and the capital requirement for waste heat recovery are considerably greater than those for most other energy saving opportunities. Thus, decisions about individual projects become more difficult to make. Expenses for engineering studies and economic analysis are substantial and thus a greater commitment to optimum energy utilization is demanded. On the other hand the rewards in the form of reduced energy costs may also be greater and this constitutes the incentive for committing resources to waste heat recovery. Sections 1 and 2 of the EPIC manual should be consulted for aid in setting up the energy management program within your organizational structure.

Implementing Waste Heat Management

The first steps to be taken are to survey the plant's process units in order to discover opportunities for recovering and using waste heat. On the next page is a survey form, suitable for direct fired or unfired units, which will contain all the information needed to obtain a heat balance for any industrial process unit. The flow chart for the process and its heat balances are then studied to determine where opportunities for waste heat recovery exist. The results of engineering and economic studies for each process unit are next evaluated and summarized, using appropriate additional information and proposals from manufacturers of waste heat recovery equipment. Whenever possible, the individual processes should be submetered for fuel consumption and instrumented so as to monitor equipment performance. It is essential if full benefit is to be obtained from the capital investment that the equipment be kept in optimum operating condition and this can only be assured through adequate instrumentation and an active testing program.

References

NBS Handbook 115 (1974): Energy Conservation Program Guide for Industry and Commerce (EPIC), by R. Gatts, R. Massey, and J. Robertson.

WASTE HEAT SURVEY

SURVEY FORM FOR INDUSTRIAL PROCESS UNITS

NAME OF PROCESS UNIT ____________ INVENTORY NUMBER ____________

LOCATION OF PROCESS UNIT, PLANT NAME ____________ BUILDING ____________

MANUFACTURER ____________ MODEL ____________ SERIAL NUMBER ____________

				TEMPERATURE OF			FLUE GAS COMPOSITION % VOLUME				
	NAME	FIRING RATE	HHV	COMB. AIR	FUEL	STACK	CO_2	O_2	CO	CH	N_2
PRIMARY FUEL											
FIRST ALTERNATIVE											
SECOND ALTERNAT.											

	FLOW PATH 1	FLOW PATH 2	FLOW PATH 3	FLOW PATH 4
FLUID COMPOSITION				
FLOW RATE				
INLET TEMPERATURE				
OUTLET TEMPERATURE				
DESCRIPTION				

ANNUAL HOURS OPERATION ____________ ANNUAL CAPACITY FACTOR, % ____________

ANNUAL FUEL CONSUMPTION: PRIMARY FUEL ________; FIRST ALTERN. ________ SEC. ALTERN. ________

PRESENT FUEL COST: PRIMARY FUEL ________; FIRST ALTERN. ________ SEC. ALTERN. ________

ANNUAL ELECTRICAL ENERGY CONSUMPTION, KWHR. ____________

PRESENT ELECTRICAL ENERGY RATE ____________

Chapter 2

DETERMINATION OF WASTE HEAT REQUIREMENTS

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Introduction

The economic recovery of waste heat depends on five factors. First, one must have a use for the waste heat. This point will not be treated in the present chapter; it will be assumed that such a use has been located. Second, one must have an adequate *quantity* of waste heat; to estimate the quantity of waste heat available one uses the first law of thermodynamics. Third, the heat must be of adequate quality for the purpose in question; for example, heat available at 300 F cannot be used directly to heat steam to 400 F. The problems of heat quality and availability are treated using the second law of thermodynamics. Fourth, the heat must be transferred from the waste stream to the material or work piece where it is to be used. This is a problem in heat transfer. Fifth and last, the waste heat must be used profitably; this is a question of economics.

The present chapter treats three of the five issues just mentioned; energy quantity (first law of thermodynamics), energy quality (second law of thermodynamics) and heat transfer. The treatments are of necessity somewhat theoretical, and the reader interested only in the "how to" of applications would be well advised to pass on directly to chapter 4. For more detailed thermodynamic information, the reader is referred to references [3] and [4]. The economics of waste heat use is treated in chapter 3.

First Law of Thermodynamics

The quantity of waste heat available is determined from heat balances of process units, or of whole plants, or of groups of plants.

The basis of heat balance equations is the first law of thermodynamics: that energy can neither be created nor destroyed. For engineering purposes we shall neglect the conversion of energy into mass or vice versa; for our purposes this law may be stated: The net increase in the energy content of a particular system in a given period is equal to the energy content of the material leaving the system, plus the work done on the system, plus the heat added to the system. For example, if water with a heat content of 100 Btu and 3000 Btu of heat enter a system, and the system does 200 Btu of work and delivers steam with a heat content of 2000 Btu, we know that (3100−2200) = 900 Btu have been lost through surface losses and blowdown. To estimate the quantity of waste heat we shall consider a system of known volume (e.g., a boiler, into which and out of which material is flowing). If we let U_{in} denote the internal energy of the material entering the system, U_{out} as the internal energy of the material leaving the system, Q as the heat added to the system, and W' as the work done *by* the system (the negative of the work done *on* the system), the first law of thermodynamics may be written:

$$Q - W' = U_{in} - U_{out}. \qquad (2.1)$$

Here we have neglected several energy terms such as kinetic energy, electrical energy, magnetic energy, and gravitational energy, which may be of importance in specific applications but which need not be considered for the simple applications in this chapter.

It is convenient to work in terms of time rates, so we shall use the notation that a quantity written with a dot over it is the time rate of change of that quantity (e.g., $\dot{U}$). We shall treat many quantities on a per pound basis and we shall denote values of various quantities per pound by lower case letters; e.g., u will denote internal energy per pound. Finally, in rewriting eq (2.1) we shall segregate W' into two factors: W'' the work which is done in moving volumes into and out of the system against pressure and W, which is all other work. W'' is $(p\dot{V})_{\mathrm{out}} - (p\dot{V})_{\mathrm{in}}$

Using the changes, we may rewrite eq (2.1) as a rate equation:

$$\dot{Q} - \dot{W} = \dot{U}_{\mathrm{in}} - \dot{U}_{\mathrm{out}} + (p\dot{V})_{\mathrm{in}} - (p\dot{V})_{\mathrm{out}}. \qquad (2.2)$$

Here $(p\dot{V})_{\mathrm{in}}$ represents the rate of work that must be done on the system to push the inbound volume flow $\dot{V}$ into the system, and $(p\dot{V})_{\mathrm{out}}$ is the rate at which the system does work on its surroundings pushing material out at the volumetric rate $\dot{V}_{\mathrm{out}}$. If we define the energy term enthalpy as $H = U + pV$, this equation can be rewritten:

$$\dot{Q} - \dot{W} = \dot{H}_{\mathrm{in}} - \dot{H}_{\mathrm{out}}. \qquad (2.3)$$

If, further, we assume that the system is in a steady state in the sense that the rate of mass flow in is the same as the rate of mass flow out, and define q as Q divided by the mass of the mass throughput and similarly define w, h, etc., we have

$$\dot{q} - \dot{w} = \dot{h}_{\mathrm{in}} - \dot{h}_{\mathrm{out}}. \qquad (2.4)$$

This equation describes the rate at which enthalpy per pound is added to the material passing through the system in terms of the heat added to a pound of throughput and the work done per pound of throughput.

The first law of thermodynamics does not impose limitations on the fraction of the available energy which can be used in a desired way. The second law of thermodynamics has the effect of imposing such limitations.

Second Law of Thermodynamics

The second law of thermodynamics says that the entropy of an isolated system cannot decrease; that is, that in an isolated system forces and temperature differentials will tend to dissipate themselves, and in any event will not grow spontaneously. For instance, the first law of thermodynamics would permit us to use waste heat in spent steam at 250 F to melt steel. The second law says we cannot; that unless we put energy into the system as a whole the heat will flow from the hot steel to the warm steam, not vice versa.

In most waste heat engineering applications, this qualitative understanding is all that is required. However, for some special cases an "availability analysis" is useful and the following discussion is included to familiarize the reader with the language of such analyses. The concept of availability is a tool for analyzing usefulness of energy in our waste heat (in second-law terms). The availability is defined (assuming velocity, gravitational, etc., contributions to energy can be neglected) by:

$$B = H - T_0 S \qquad (2.5)$$

where B is the availability, H the enthalpy, S the entropy, and T_0 the temperature of the lowest-temperature reservoir we have available for "dumping" heat. For most purposes T_0 is the ambient temperature, or the temperature of a cooling pond. Eq (2.5) may be written as a rate equation:

$$\dot{B} = \dot{H} - T_0 \dot{S}. \qquad (2.6)$$

On a per-pound basis, we may write the rate of change of availability as:

$$\dot{b} = \dot{u} + p\dot{v} - T_0\dot{s} \qquad (2.7)$$

where again we have followed the convention of using lower-case letters for per-pound quantities and using the dot to indicate time rate of change. In deriving (2.7) we have used the equation for h (i.e., for H per unit mass):

$$h = u + pv \qquad (2.8)$$

The availability has particular significance because the change in the availability of something (e.g., a pound of steam) in a given process is the maximum possible useful work or heat which can be extracted from it in the given process. An example of the use of this concept is as follows. Suppose we are using exhaust air at 100 F as a source of heat for warming intake air. If the outside temperature is 95 F we can retrieve very little heat; if the outside air is a 0 F we can get quite a lot.

To examine the usefulness of this concept for our purposes, consider a system in which we have $\dot{m}$ pounds per unit time passing through some device which we expect to use to recover useful energy, and we wish to know what the maximum power is that we can develop. If we denote this power as $\dot{W}_{\mathrm{max}}$ (since power is the rate of doing work), we have:

$$\dot{W}_{\mathrm{max}} = -\dot{m}(b_{\mathrm{out}} - b_{\mathrm{in}}). \qquad (2.9)$$

or

$$\dot{W}_{\mathrm{max}} = \dot{B}_{\mathrm{in}} - \dot{B}_{\mathrm{out}}. \qquad (2.10)$$

Written out, eq (2.10) is:

$$W_{\mathrm{max}} = \dot{m}\,[u_{\mathrm{in}} + (pv)_{\mathrm{in}} - T_0 s_{\mathrm{in}} - u_{\mathrm{out}} - (pv)_{\mathrm{out}} + T_0 s_{\mathrm{out}}]. \qquad (2.11)$$

The equation tells us that the larger the specific internal energy and pressure at inlet, the more work is possible; while the more entropy is produced within the system, the less work is possible. Thus high quality of energy is characterized by high internal energy and high pressure, but the quality is reduced by increased entropy. The internal energy is a function of temperature, so high temperatures represent high internal energy. In the next portion of this chapter we shall apply elementary thermodynamics to a few practical problems.

The great preponderance of our energy requirements are met by burning fossil fuels at atmospheric pressure. When the chemically correct amount of air is mixed with hydrocarbon fuels, the resulting temperature is in the range above 2000 F. If the combustion takes place in a boiler where steam is generated at 150 lb/in^{2}g, (that is at 366 F), we can hope conservatively to transfer 85 percent of the chemical energy in the fuel to energy in the steam. We have lost only 15 percent of the total input energy but the availability has been decreased by 66 percent. How could we have avoided the destruction of that potential to do work? And furthermore what difference does it make?

The answer to the first question is that we could have carried out the combustion process at a higher pressure, say 5 or 6 atmospheres, expanded the products through a gas turbine and produced a quantity of work. We could then have passed the turbine exhaust gases through a waste heat boiler to generate steam at the same pressure and temperature at which we had produced it previously. Whether this scheme is practical depends upon the relative needs for mechanical work and for process heat in the plant and the respective demand profiles for the two commodities. Although it is institutionally awkward, it is sometimes possible to sell generated power to neighbor plants or to the local electrical utility. This approach to making use of the work potential of fossil fuels is one that will become more prevalent as its implications are better understood, as arrangements for sharing power are developed, and as fuel costs continue to rise.

The answer to the second question posed above is that it is important to make use of the work potential of fossil fuel combustion because work is more scarce and more expensive than heat. Since the average efficiency of all power plants in this country is about 30 percent, and other heat engines are also very inefficient, mechanical work is worth over three times as much as an equivalent quantity of heat. Somehow we have missed the economic impact of that fact until recently, but we will not be able to ignore it in the future.

As a simple example let us analyze the energy requirements for a hypothetical small industrial plant [1]. We shall, in treating power plants, use the abbreviations bbl/d = barrels per day and SCF = cubic feet at standard T and P.

"Let's assume that we must design a plant whose only requirements are 50,000 lb/h of 150 lb/in^{2}g saturated steam, 30,000 lb/h of 20 lb/in^{2}g saturated steam, and 3500 hp of mechanical drives. We have determined that our cost for oil to generate this steam will be \$0.60/MBtu and have a contract with our utility to provide electricity at \$0.012/kWh.

"The most straightforward technique would be to provide a boiler capable of generating steam at 150 lb/in^{2}g and then providing the required amount of 20 lb/in^{2}g steam by reducing its pressure from 150 lb/in^{2}g. The simplified sketch in figure 2.1 shows this system and includes a water heater to heat the boiler feedwater before it goes to the boiler drum.

"To provide these services our boiler must produce 95,600 lb/h of steam which will cost \$648,000/yr and consume 485 bbl/d of oil. Our electricity bill will be \$322,900/yr and the utility must consume about 115 bbl/d of oil to produce this electricity. Thus, at an annual cost of nearly one million dollars, the total energy requirements for this plant are about 600 bbl/d of oil.

"A system which would use energy more efficiently generates steam at a higher pressure

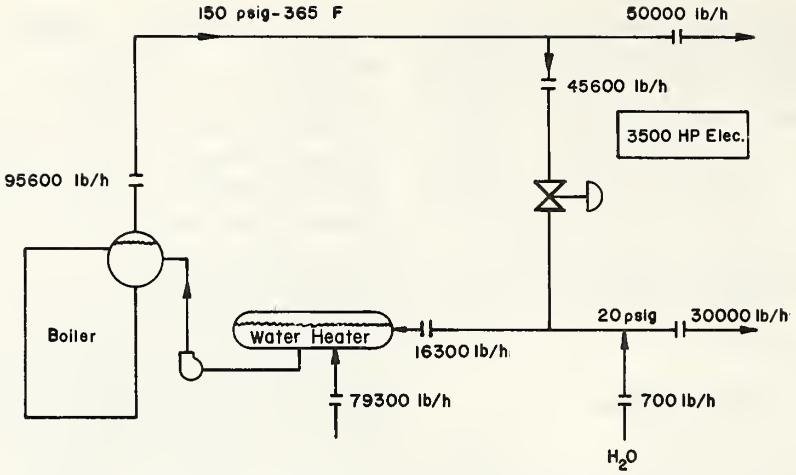

FIGURE 2.1 *Diagram of hypothetical small industrial steam generator.* Reprinted from Hydrocarbon Processing, Vol. 52, No. 7 (July 1973) by permission of Gulf Publishing Co., Houston, Texas.

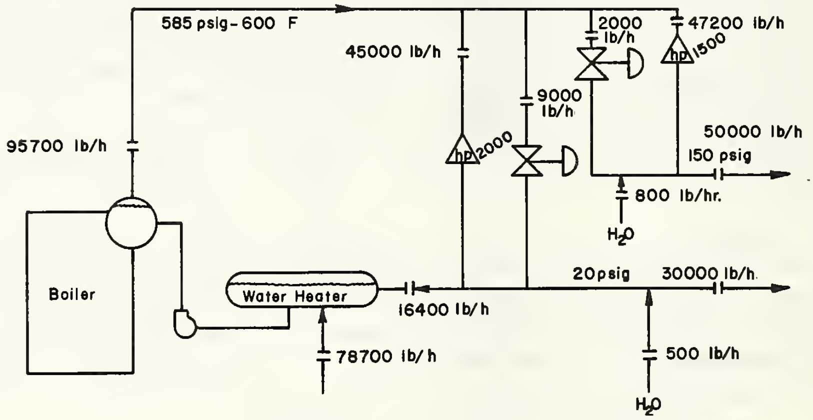

FIGURE 2.2 *Diagram of a small industrial steam generator designed for better utilization of available energy.* Reprinted from Hydrocarbon Processing, Vol. 52, No. 7 (July 1973) by permission of Gulf Publishing Co., Houston, Texas.

and temperature. For this illustration (fig. 2.2) we have selected 585 lb/in^{2}g and 600 F.

"Mechanical energy is obtained through steam turbine drives with the exhaust from these turbines used for the process needs. The schematic shows that the turbine exhausts provide most of the steam requirements for the processes.

"Additional requirements are provided through pressure reducing valves. Such a system requires that our boilers generate 95,700 lb/h of steam at annual cost of $707,000 and will consume 530 bbl/d of oil. We are providing our own mechanical power, so no electrical energy is used. Annual energy cost is cut by more than 1/4 million dollars. Total energy requirements have been reduced 70 bbl/d of oil, or a reduction of 11.6 percent."

This example points up the value of considering a thermomechanical energy system from the standpoint of the second law of thermodynamics and availability. Although a first law analysis would not have shown this opportunity for savings, the high availability of the fuel could lead to substantial cost savings. These savings come from use of higher pressures, which permit turbine energy extraction.

Deriving the Heat Balance

A heat balance is an analysis of a process which shows where all the heat comes from and where it goes. This is a vital tool in assessing the profit implications of heat losses and proposed waste heat utilization projects. The heat balance for a steam boiler, process furnace, air conditioner, etc., must be derived from measurements made during actual operating periods. Chapter 6 provides information on the instrumentation available to make these measurements. The measurements that are needed to get a complete heat balance involve: energy inputs, energy losses to the environment, and energy discharges.

Energy Input

Energy enters most process equipment either as chemical energy in the form of fossil fuels, of sensible enthalpy of fluid streams, of latent heat in vapor streams, or as electrical energy.

For each input it is necessary to meter the quantity of fluid flowing or the electrical current. This means that if accurate results are to be obtained, submetering for each flow is required (unless all other equipment served by a main meter can be shut down so that the main meter can be used to measure the inlet flow to the unit). It is not necessary to continuously submeter every flow since temporary installations can provide sufficient information. In the case of furnaces and boilers that use pressure ratio combustion controls, the control flow meters can be utilized to yield the correct information. It should also be pointed out that for furnaces and boilers only the fuel need be metered. Tests of the exhaust products provide sufficient information to derive the oxidant (usually air) flow if accurate fuel flow data are available. For electrical energy inflows, the current is measured with an ammeter, or a kilowatt hour meter may be installed as a submeter. Ammeters using split core transformers are available for measuring alternating current flow without opening the line. These are particularly convenient for temporary installations.

In addition to measuring the flow for each inlet stream it is necessary to know the chemical composition of the stream. For air, water, and other pure substances no tests for composition are required, but for fossil fuels the composition must be determined by chemical analysis or secured from the fuel supplier. For vapors one should know the quality—this is the mass fraction of vapor present in the mixture of vapor and droplets. Measurement of quality is made with a vapor calorimeter which requires only a small sample of the vapor stream.

Other measurements that are required are the entering temperatures of the inlet stream of fluid and the voltages of the electrical energy entering (unless kilowatt-hour meters are used).

The testing routines discussed above involve a good deal of time, trouble, and expense. However, they are necessary for accurate analyses and may constitute the critical element in the engineering and economic analyses required to support decisions to expend capital on waste heat recovery equipment.

Energy Losses

Energy loss from process equipment to the ambient environment is usually by radiative and convective heat transfer. Radiant heat transfer, that is, heat transfer by light or other electromagnetic radiation, is discussed in the section of chapter 6 dealing with infrared thermography. Convective heat transfer, which takes place by hot gas at the surface of the hot material being displaced by cooler gas, may be analyzed using Newton's law of cooling.

$$\dot{q} = h_{cr} A \, (T_s - T_0) \qquad (2.12)$$

where

$\dot{q}$ = rate of heat loss in energy units Btu/h
h_{cr} = heat transfer coefficient in Btu/h·ft²·F
A = area of surface losing heat in ft²
T_s = surface temperature
T_0 = ambient temperature

Although heat flux meters are available, it is usually easier to measure the quantities above and derive the heat loss from the equations. The problems encountered in using the equation involve the measurement of surface temperatures and the finding of accurate values for the heat transfer coefficient.

Unfortunately the temperature distribution over the surface of a process unit can be very nonuniform so that an estimate of the overall average is quite difficult. New infrared measurement techniques, which are discussed in chapter 6, make the determination somewhat more accurate. The heat transfer coefficient is not only a strong function of surface and ambient temperatures but also depends on geometric considerations and surface conditions. Thus for given surface and ambient temperatures a flat vertical plate will have a different h_{cr} value than will a horizontal or inclined plate. Table 7.1 in chapter 7 contains data on heat transfer coefficients which can be used for estimating heat losses.

Energy Discharges

The composition, discharge rate and temperature of each outflow from the process unit are required in order to complete the heat balance. For a fuel-fired unit, only the composition of the exhaust products, the flue gas temperature and the fuel input rate to the unit are required to derive:

(1) air input rate
(2) exhaust gas flow rate
(3) energy discharge rate from exhaust stack

The composition of the exhaust products can be determined from an Orsat analysis, a chromatographic test, or less accurately from a determination of the volumetric fraction of oxygen or CO_2. Figure 7.1 in chapter 7 is an example of a chart that can be used for determining the quantity of excess or deficiency or air in the combustible mixture. It is based on the fact that chemical reactions occur with fixed ratios of reactants to form given products.

For example a natural gas with the following composition:

CO_2	-	0.7% volume
O_2	-	0.0
CH_4	-	92.0
C_2H_6	-	6.8
N_2	-	0.5
		100.00

is burned to completion with the theoretical amount of air indicated in the volume equation below:

FUEL (1 ft³) $0.92\,CH_4 + 0.068\,C_2H_6 + 0.007\,CO_2 + 0.005\,N_2$

plus

AIR (9.891 ft³) $2.078\,O_2 + 7.813\,N_2 \rightarrow$

yields

$$\text{DRY PRODUCTS}\quad 1.063\,CO_2 + 7.818\,N_2 + 2.044\,H_2O \tag{2.13}$$

The equation is based upon the laws of conservation of mass and elemental chemical species. The ratio of N_2 and O_2 in the combustible mixture comes about from the approximate volumetric ratio of N_2 to O_2 in air, i.e.,

$$20.9\%\ O_2,\ 79.1\%\ N_2 \text{ or } \frac{79.1}{20.9} = 3.76 = \frac{\text{Volume } N_2}{\text{Volume } O_2}.$$

For gases the coefficients of the chemical equation represent relative volumes of each species reacting. Ordinarily excess air is provided to the fuel so that every fuel molecule will react with the necessary number of oxygen molecules even though the physical mixing process is imperfect. If 10 percent excess air were supplied, this mixture of reactants and products would give a chemical equation appropriately modified as given below:

FUEL (1 ft³) $0.92\,CH_4 + 0.068\,C_2H_6 + 0.007\,CO_2 + 0.005\,N_2$

plus

AIR (10.880 ft³) $2.86\,O_2 + 8.594\,N_2 \rightarrow$

yields

$$\text{DRY PRODUCTS}\quad 1.063\,CO_2 + 0.208\,O_2 + 8.599\,N_2 + 2.044\,H_2O \tag{2.14}$$

where an additional term representing the excess oxygen appears in the products, along with a corresponding increase in the nitrogen.

As an example let us assume that an oxygen meter has indicated a reading of 7 percent for the products of combustion from the natural gas whose composition was given previously and that the exhaust gas temperature was measured as 700 F. Figure 7.1 is used as indicated to determine that 45 percent excess air is mixed with fuel. The combustion equation then becomes:

FUEL (1 ft^3) $0.92\ CH_4 + 0.068\ C_2H_6 + 0.007\ CO_2 + 0.005\ N_2$

plus

AIR $3.013\ O_2 + 11.329\ N_2 \rightarrow$

yields

DRY PRODUCTS (15.376 ft^3) $1.063\ CO_2 + 0.935\ O_2 + 11.334\ N_3 + 2.044\ H_2O$ (2.15)

For each 1 ft^3 of fuel, 14.342 ft^3 of air is supplied and 15.376 ft^3 of exhaust products (at mixture temperature) are formed. Each cubic foot of fuel contains 1055 Btu of energy, so the fuel energy input is 250,000 × 1055 = 263,750,000 Btu/h. From figure 7.2 we compute the exhaust gas losses at 700 F as:

CO_2: 250,000 × 1.063 × 17.5 = 4,651,000 Btu/h
H_2O: 250,000 × 2.044 × 14.0 = 7,154,000
O_2: 250,000 × 0.935 × 12.3 = 2,875,000
N_2: 250,000 × 11.334 × 11.5 = 32,585,000

47,265,000 Btu/h

or 18 percent of the fuel energy supplied. Some of this could be recovered by suitable waste heat equipment.

Heat Balance on a Boiler

Let us consider a further example, a heat balance on a boiler. A process steam boiler has the following specifications:

- Natural gas fuel with HHV = 1001.2 Btu/ft^3
- Gas firing rate = 2126.5 ft^3/min
- Steam discharge at 150 lb/in^2g saturated
- Steam capacity of 100,000 lb/h
- Condensate returned at 180 F

The heat balance on the burner is derived from measurements made *after* the burner controls had been adjusted for an optimum air/fuel ratio corresponding to 10 percent excess air. All values of the heat content of the fluid streams are referred to a base temperature of 60 F. Consequently the computations for each fluid stream entering or leaving the boiler are made by use of the equation below:

$$\dot{H} = \dot{m}\ (h - h_0) \tag{2.16}$$

where

$\dot{H}$ = the enthalpy rate for entrance or exit fluids
$\dot{m}$ = mass flow rates for entrance or exit fluids
h = specific enthalpy at the fluid temperature of the fluid entering or leaving the entrance or exit
h_0 = specific enthalpy of that fluid at the reference temperature T_0 = 60 F

The first law of thermodynamics for the boiler is expressed as: Sum of all the enthalpy rates of substances entering = Sum of all enthalpy rates of substances leaving + q where q is the rate of heat loss to the surroundings. This can be expressed as

$$\Sigma_{\text{in}} H_i = \Sigma_{\text{out}} H_i + q \tag{2.17}$$

where Σ is the summation sign, and H_i is the enthalpy rate of substance i.

For gaseous fuels the computation for the heat content of the gases is more conveniently expressed in the form:

$$H = J_0 C_{pm} (T - T_0) \tag{2.18}$$

where

J_0 = the volume rate of the gas stream corrected back to 1.0 atmosphere and 60 F (T_0 = 60 F); and
C_{pm} = the specific heat given on the basis of a standard volume of gas averaged over the temperature range ($T - T_0$) and the gas mixture components.

$$C_{pm} = \Sigma_i X_i C_{pi} \tag{2.19}$$

where

X_i = percent by volume of a component in one of the flow paths.
C_{pi} = average specific heat over the temperature range for each gas component.

From eq (2.14) we derive the volume fractions of each gas component as follows:

Component	*X*
CO_2	8.9
H_2O	17.2
N_2	72.2
O_2	1.7
	100.0

The average specific heat is found (using fig. 7.3)

$$\begin{array}{rcrcl} C_{pm} = 0.089 & \times & 0.0275 & = & 0.00245 \\ 0.172 & \times & 0.0220 & = & 0.00378 \\ 0.722 & \times & 0.0186 & = & 0.01344 \\ 0.017 & \times & 0.0195 & = & 0.00033 \\ \hline & & & & 0.02 \text{ Btu/Scf}\cdot\text{F} \end{array}$$

The combustion equation also tells that when the products are at standard conditions 11.88 ft^3 fuel and air generate 11.915 ft^3 of products and that for every cubic foot of gas burned, 10.880 ft^3 of air is introduced. Thus for a firing rate of 2126.5 ft^3/min the air required is 2126.5 × 10.88 = 23,136 ft^3/min or 23,136 × 60 = 1,388,179 ft^3/h. The total fuel and air flow rate is then almost exactly equal to 1,388,179 + 2126.5 × 60 = 1,515,769 ft^3/h. This corresponds to a flue gas discharge rate of 1,519,128 ft^3/h.

For a flue gas discharge rate of 1,519,128 ft^3/h and a temperature of 702 F, the total exhaust heat rate is found as:

$$\dot{H}_{Ex.Gas} = 1{,}519{,}128 \ \frac{ft^3}{h} \times 0.02 \ \frac{Btu}{ft^3 \cdot F} \times (702 - 60)\ F = 19{,}505{,}604 \ Btu/h$$

For the steam leaving the boiler (100,000 lb/h at 150 lb/in^{2}g saturated) the energy flow rate is found using the Steam Tables, table 7.2 in Chapter 7 and the equation below:

$$\begin{aligned} \dot{H}_{Steam} &= \dot{m}\ (h - h_0) \\ &= 100{,}000 \text{ lb/h } (1195.6 - 28.08) \text{ Btu/lb} \\ &= 116{,}752{,}000 \text{ Btu/h} \end{aligned}$$

where 1195.6 is the specific enthalpy of saturated steam at 150 lb/in^{2}g and 28.08 is the specific enthalpy of saturated liquid water at 60 F since we are using that temperature as our standard reference temperature for the heat balance. We have used 60 F as a reference temperature, this is not universal practice and in the boiler industry 70 F is more common, whereas in other areas 25 C is normal.

Chemical Energy in Fuel

To determine the heat content of the chemical energy in the fuel, find the higher heating value (HHV) for the fuel and multiply it by the volumetric flow rate for a gaseous fuel or the mass flow rate for a liquid or solid fuel. The assumed higher heating value for the natural gas used in the boiler of our example is 1001.1 Btu/ft^3 and the heat content rate is then:

$$\begin{aligned} \dot{H} &= 2126.5 \ \frac{ft^3}{min} \times 60 \ \frac{min}{h} \times 1001.1 \ \frac{Btu}{ft^3} \\ &= 127{,}743{,}000 \ \frac{Btu}{h} \end{aligned}$$

The enthalpy rates for the condensate return and make up water are derived from data in the steam tables where the specific enthalpy of the compressed liquids are taken to be almost exactly equal to the specific enthalpy of the *saturated* liquid found at the same *temperature*.

The complete heat balance derived in the manner detailed above is presented in figure 2.3.

Waste heat is available from the combustion products leaving the stack. It amounts to 19,505,604 Btu/h at a temperature of 702 F. Some energy is also available from the condensers.

There may be a practical limit to the amount of heat that can be removed from the stack gases, however, figure 2.4 (reproduced for convenience on the next page) gives the relationship between the percent excess air and the dew point temperature (temperature at which the water vapor in the exhaust products will condense) for the case we are considering.

If the water vapor, which is formed by burning the hydrogen in the fuel, is cooled to the point where it condenses, then part of the condensed liquid may collect on the stack liner and cause rusting or corrosion. This corrosion may increase maintenance and replacement costs for the stack to the point where additional waste heat recovery made possible by reduction of temperatures below the saturation temperature becomes uneconomical.

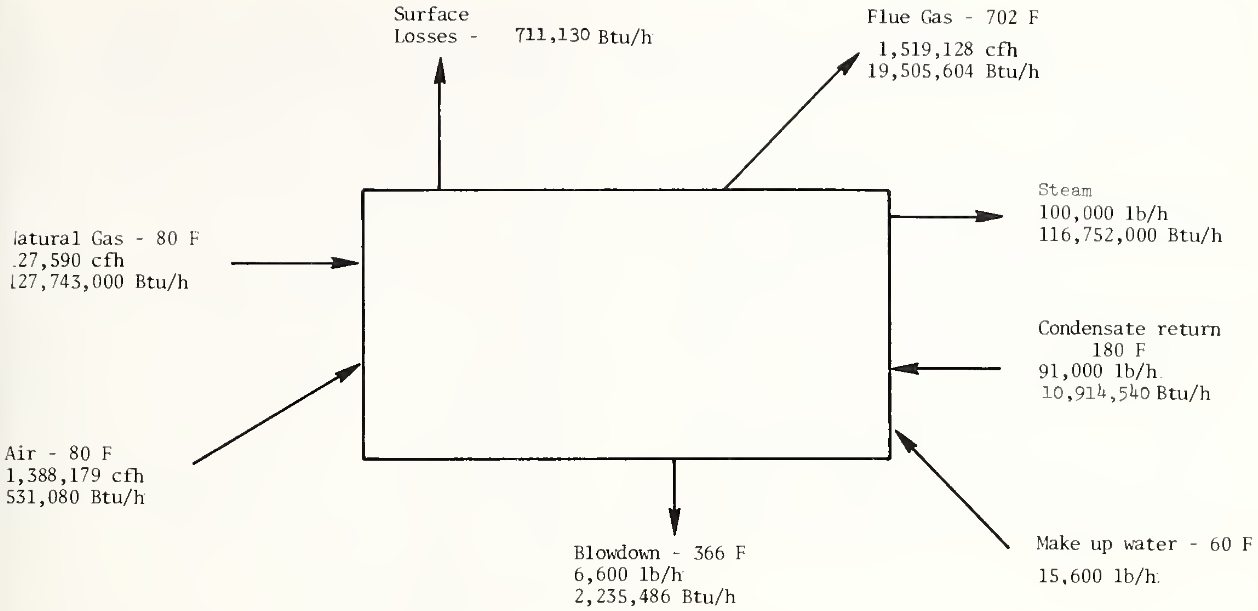

Figure 2.3 *Heat balance for a simple steam generator burning natural gas with 10 percent excess air.*

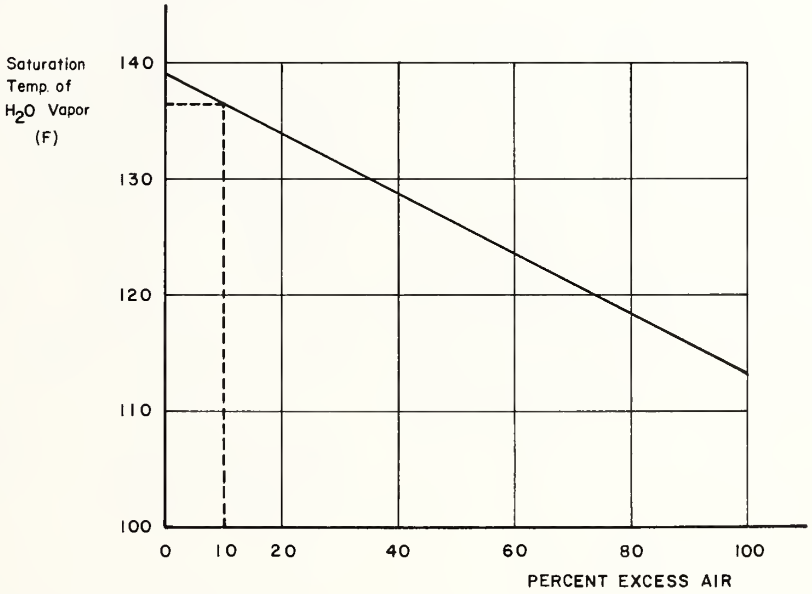

Figure 2.4 *Saturation temperature of water vapor in exhaust products from a gas fired boiler as a function of percent excess air.*

Stack Gas Condensation

To use figure 7.4 for the illustrative example enter the graph at 1001.2 Btu/ft^3 along one of the bottom abscissa and run a vertical line to the gaseous fuel curve. From the point of intersection run a horizontal line to the left ordinate and then follow the curve intersecting the ordinate at that point downward and to the right until an intersection with a vertical line representing the percent excess air is made. From that point move horizontally to the left and read the dew point temperature. For the present example enter at 1001.2 Btu/ft^3, follow a vertical to the intersection of the gaseous fuel curve and then horizontally to the left to read 140 F. Follow the intersecting curve downward to the right until it intersects with the vertical from the top ordinate representing 10 percent excess air. Move horizontally to the left and read 137 F.

Thus the condensation temperature for the exhaust products from this boiler are 137 F when operated with 10 percent excess dry air. Since the stack liner may exist at a temperature 50 to 75 F lower than the average gas stream temperature, it is prudent in this example to limit the lowest stack gas temperature to at least 212 F. The effect of water vapor in the entering air on this condensation temperature is small. If the combustion air entering the boiler were at a relative humidity of 100 percent at 80 F, then the saturation temperature would increase only to 140 F and the minimum stack temperature then must be held above 214 F. As mentioned before, 250 or 300 F is better for safety.

Waste Heat Recovery

The energy exhausted to the atmosphere is energy which has already been paid for and which should not be discarded until the last penny of profit has been extracted. A portion of it however can be recovered by using a heat exchanger. Any requirement for energy at a temperature in excess of 200 F can be satisfied. It is necessary to identify the prospective uses for the waste energy; make an economic analysis of the costs and savings involved in each of the options; and decide among those options on the basis of the economics of each. An important option in every case is that of rejecting all options if none proves economic. For the illustrative example, let us assume that the following uses for the waste heat have been identified.

- Preheating the combustion air
- Preheating the boiler feedwater
- Heating the domestic hot water supply (370 gal/h from 50 to 170 F)
- A combination of the preceding

The first two are practices which are standard in energy-intensive high technology industries (e.g., electric companies).

Two rapid calculations give the waste heat available in the exhaust gases between 702 F and 220 F and the heat requirements for the domestic hot water supply to be 14,063,000 Btu/h and 369,900 Btu/h respectively. Since the latter constitutes only 2.6 percent of the available waste energy, we should reject that option. The remaining options are to preheat the combustion air, the feedwater, or to divide the waste energy between those two options. For a small boiler one probably would not find the purchase of two separate heat exchangers an economic option, so that we shall limit ourselves to one of the other of the first two options. Without going into a detailed analysis, we may note that preheating the combustion air or the feedwater provides a double benefit. Since the preheated air or water requires less fuel to produce the same steam capacity, a direct fuel saving results. But the smaller quantity of fuel means a smaller air requirement, and this in turn means a smaller quantity of exhaust products and thus smaller stack loss at a given stack temperature.

The economic benefits can be estimated as follows. The air preheater is estimated to save 6 percent of the fuel. With an average boiler loading of 60 percent for the 8760 h in a year this amounts to an annual fuel saving of

$$0.06 \times 8760 \times 0.60 \times 127{,}590 = 40{,}103{,}000 \ \frac{\text{ft}^3}{\text{yr}}$$

which is worth, at an average rate of \$1.10/1000 ft^3, an annual dollar saving of

$$\frac{40{,}103{,}000}{1000} \times \$1.10 = \$44{,}000.00$$

If the costs for installing the air preheater are assumed to be

Cost of preheater	$26,000
Cost of installation	28,600
New burners, air piping, controls and fan	23,300
	$77,900

For the feedwater heater (or economizer) the fuel savings is 9.2 percent or a total annual fuel saving of:

$$0.092 \times 127{,}600 \times 8760 \times 0.60 = 61{,}696{,}000 \ \frac{\text{ft}^3}{\text{yr}}$$

and the economic benefit is

$$\frac{61{,}696{,}000 \ \frac{\text{ft}^3}{\text{yr}}}{1000} \times \$1.10 = \$68{,}000/\text{yr}.$$

The cost of the economizer installed is estimated to be $67,000. This is clearly the best option for this particular boiler, especially as there is no need in this case for modifications to the boiler and accessories beyond the heat exchanger retrofit. There are several reasons for its superiority. The first is that in the case of air preheating one is exchanging the waste heat in the gases to the incoming air which has almost the same mass flow rate and almost the same specific heat. Thus one can expect that the final temperature of the preheater air will be almost the arithmetic mean of the ambient air temperature and the exhaust gas temperature entering the economizer. In the case of the feedwater the mass flow rate times specific heat is over four and one quarter times that of the combustion air. Thus we can expect to transfer more energy to the water and end up with a lower flue gas temperature leaving the stack.

The second reason is that preheating the air quite often (as in this case) affects the boiler accessories which requires additional modifications and thus related capital expenditures.

In the preceding example of the process steam boiler, we are analyzing an efficient process unit. The heat available in the product (process steam) constituted a large percentage of the energy introduced in the fuel. The efficiency in percentage terms is computed as

$$\eta = \frac{\text{Useful output}}{\text{Energy input}} = \frac{H_{\text{steam}}}{H_{\text{fuel}}} \times 100$$

$$\eta = \frac{100{,}000 \text{ lb/h } (1195.6 - 148) \times 100}{127{,}590 \text{ ft}^3\text{/h} \times 1001.2 \text{ Btu/ft}^3}$$

$$= \frac{1.0476 \times 10^{10}}{1.2775 \times 10^{8}} = 0.82, \text{ or } 82\ \% \qquad (2.20)$$

Heat Recovery in Steel Tube Furnace

As a second and very different example note the steel tube furnace illustrated below in figure 2.5.

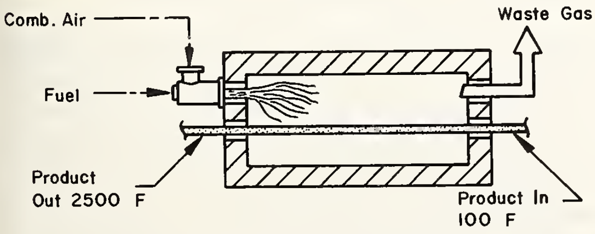

FIGURE 2.5 *Continuous steel tube furnace.*

The tubing enters the furnace from the right at a temperature of 100 F.

The specifications for the steel tube heating furnace are:

Product capacity - 50 ton/h
Product specifications - 0.23 percent carbon steel
Final product temprature - 2000 F
Air/Fuel inlet temperature - 100 F
Air/Fuel mixture - 10 percent excess air
Fuel - No. 5 fuel oil gravity API° - 16
Fuel firing rate 48.71 gpm at 240 F
(factory usage)
Utilization factor - 0.62

The useful heat leaves the furnace in the steel at 2000 F. This is called the useful furnace output and equals:

$$\begin{aligned} H_{\text{prod}} &= \dot{m}_{\text{prod}} C_P (T_{\text{out}} - T_{\text{in}}) \\ &= 50 \ \frac{\text{ton}}{\text{h}} \times 2200 \text{ lb/ton} \\ &\times 0.179 \text{ Btu/lb} - \text{F } (200 - 100) \\ &= 0.3741 \times 10^{8} \ \frac{\text{Btu}}{\text{h}} \end{aligned}$$

0.179 is used as the average specific heat of steel over the 100 F-2000 F range, based on data in 7.3. The heat input to the furnace is the chemical energy in the fuel oil. The heating value for No. 5 fuel oil is found from table 7.4. The heat input is determined by:

$$\begin{aligned} H_{\text{fuel}} &= 48.71 \text{ gpm} \times 142{,}300 \ \frac{\text{Btu}}{\text{gal}} \\ &= 4.159 \times 10^{8} \text{ Btu/h} \end{aligned}$$

The percent efficiency of the furnace is:

$$\begin{aligned} \eta &= \frac{\text{output} \times 100}{\text{input}} \\ &= \frac{\text{Enthalpy added to steel} \times 100}{\text{Enthalpy entering with fuel}} \\ &= \frac{0.3741 \times 10^{8}}{4.159 \times 10^{8}} \times 100 = 9 \text{ percent.} \qquad (2.21) \end{aligned}$$

This means that of the 415.9 MBtu introduced per hour to the furnace, that 378.4 million are released to the atmosphere. At the present average cost of \$0.349/gal for No. 5 fuel oil, the heat wasted is

$$q_{\text{waste}} = 48.71 \text{ gpm} \times (1 - 0.09) \frac{\text{gal wasted}}{\text{gal used}}$$
$$\times\ 60 \text{ min/h} \times \$0.349/\text{gal} = \$928/\text{h}$$
$$\text{or as much as } \$928 \times 8760 \text{ h/yr}$$
$$= \$8{,}129{,}280/\text{yr}$$

In order to construct the combustion equation we must note from table 7.5 that the carbon-hydrogen ratio is about 7.3. Thus:

$$C_{6.08}H_{10} + 9.44\ O_2 + 35.49\ N_2 \rightarrow 6.08\ CO_2 + 5\ H_2O + 0.86\ O_2 + 35.49\ N_2.$$

Again referring to table 7.4 the density of the liquid fuel is 7.99 lb/gal. Therefore the above equation represents the reaction for 100 lb fuel/7.99 lb/gal = 12.51 gal and for 35.49 + 9.44 or 44.93 lb mol of air. 44.93 lb mol of air weighs 1301 lb (using the molecular weight for air found in table 7.6). The specific volume of air at standard conditions is found from the same table as 378.5 ft^3/lb mol. Therefore the volume of air required to burn 12.51 gal of fuel is $44.93 \times 378.5 = 17{,}000\ \text{ft}^3$ at standard temperature and pressure. The air input is then:

$$Q = \frac{17{,}000\ \text{ft}^3}{12.51\ \text{gal}} \cdot 48.71\ \text{gpm} = 66{,}200\ \text{ft}^3/\text{m}$$

or

$$Q_{\text{air}} = 66{,}200 \times 60 = 3{,}972{,}000\ \text{ft}^3/\text{h}$$

$$H_{\text{air}} = 1.2\ \text{Btu/ft}^3 \times 3{,}972{,}000$$
$$= 4{,}766{,}000\ \text{Btu/h}$$

The enthalpy of the entering air is 1.2 Btu/ft^3 at 100 F.

The volume ratio of stack gases at standard conditions to air inlet rate is found from the combustion equation as:

$$\frac{6.08 + 5 + 0.86 + 35.49}{9.44 + 35.49} = \frac{47.43}{44.93} = 1.056$$

$$Q_{\text{ex}} = 1.056 \times 3{,}972{,}000$$
$$= 4{,}194{,}000\ \text{ft}^3/\text{h}$$

The volume fractions of the stack gas components are found from the combustion equation as:

	X
CO_2	0.128
H_2O	0.106
O_2	0.018
N_2	0.748
	1.000

The specific enthalpy of the stack gas components may be calculated by use of figure 7.2 for the exhaust temperature of 2200 F leading to the average specific enthalpy as indicated below:

				h
CO_2	0.128	× 69.3	=	8.870
H_2O	0.106	× 54.0	=	5.724
O_2	0.018	× 45.4	=	0.817
N_2	0.748	× 43.4	=	32.463
				47.874

Therefore

$$H_{\text{ex}} = 4{,}194{,}000\ \text{ft}^3/\text{h} \times 47{,}874\ \frac{\text{Btu}}{\text{ft}^3}$$
$$= 200{,}784{,}000\ \frac{\text{Btu}}{\text{h}}.$$

The heat losses from the surface equal the fuel enthalpy input less the exhaust gas enthalpy and the produce enthalpy.

$$q = 415{,}880{,}000 + 4{,}766{,}000 - 200{,}784{,}000 - 37{,}410{,}000 = 182{,}451{,}270\ \frac{\text{Btu}}{\text{h}}.$$

The complete heat balance diagram appears in figure 2.6. Now suppose that we install a metallic recuperator and preheat the combustion air to 750 F. The new heat balance is shown in figure 2.7.

The economic analysis of this situation is as follows.

The waste heat saved is equal to the fuel savings times the heating value of the fuel.

$$q_{\text{saved}} = (48.71 - 31.66)\ \text{gpm} \times 142{,}300\ \frac{\text{Btu}}{\text{gal}}$$

$$= 145{,}554{,}000 \quad \text{or} \quad 35.0 \text{ percent saving}$$

which equals

$$0.35 \times 48.71\ \text{gpm} \times 60 \times \$0.349/\text{gal} = \$357/\text{h}$$

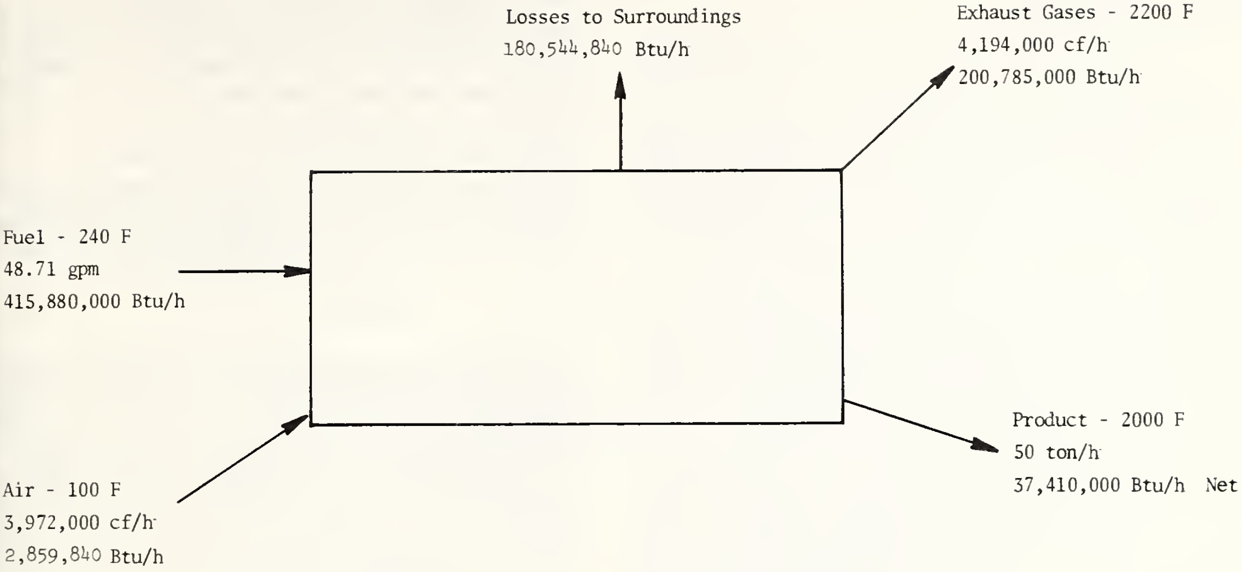

FIGURE 2.6 *Heat balance for a simple continuous steel tube furnace.*

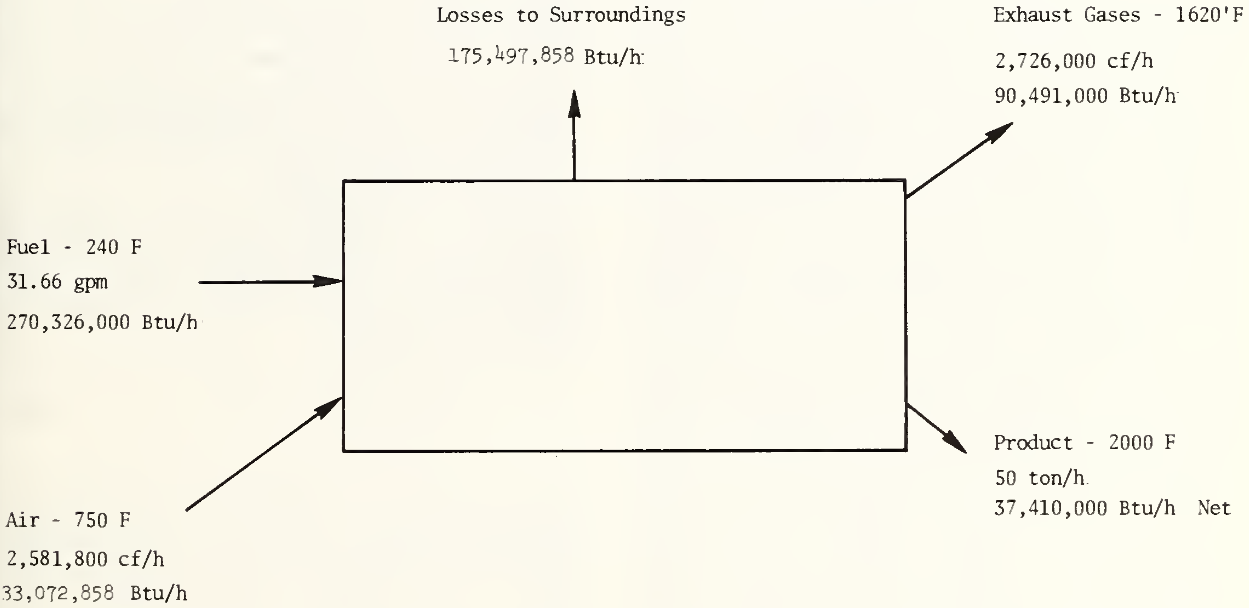

FIGURE 2.7 *Heat balance for a continuous steel tube furnace equipped with a recuperator preheating combustion air to 750 °F.*

In using figures 7.5 and 7.6 remember that the combustion air volume (in SCF) is reduced directly proportional to the reduction in firing rate. If we maintain the same air/fuel ratio for the furnace (10% excess air), the air rate is reduced from the initial value of 3,972,000 ft^3/h at a firing rate of 48.71 gpm to a rate of 31.66 gpm/48.71 gpm × 3,972,000 ft^3/h = 2,581,800 ft^3/h with the lower firing rate of 31.66 gpm when the recuperator is used.

If as in our example the furnace is used 0.62 × 8760 or 5431 h/yr then the annual saving is 5431 × 357 = $1,940,000/yr. Since a metallic recuperator can be purchased for a fraction of $1.9 million, the savings will allow a payoff in less than a year.

Heat Transfer

Up to this point, we have treated the estimation of the quantity of waste heat (first law of thermodynamics) and quality of waste heat (second law of thermodynamics); now we shall consider briefly heat transfer, which is the necessary step in transferring the waste heat from the flow in which it is of value to the stream in which it is of value.

For the most part, waste heat recovery involves the transfer of heat from a fluid which can give up heat which would otherwise be wasted to another fluid which must be heated for the process involved and whose heating would require the use of additional heat energy. To accomplish this, a device known as a heat exchanger would be used. These are discussed in detail in Chapter 5, and heat transfer is treated in references [5] and [6]; for the reader's convenience a few points are mentioned here. Heat exchangers are divided into three broad groups. The simplest type directly mixes the hot and cold fluids directly; in the second type, heat is directly transferred from one fluid to the other fluid; in the third type, the heat is intermittently stored. The first type, sometimes called an open heater, has very limited usefulness in waste heat recovery. The second type of heat exchanger transmits the heat from one fluid to another through a separating wall, and requires both fluids to be flowing simultaneously. Examples of this type are recuperators, boilers, condensers and evaporators. The third type involves periodically storing heat which has been extracted from the hotter fluid, and then allowing the colder fluid to be heated by the stored heat in the exchanger. The hotter and colder fluid must switch, on a more or less regular basis, in their flow through the heat exchanger. Such heat exchangers are usually called regenerators. Examples are the brick type regenerators of open hearth furnaces and glass furnaces. Another form of regenerator is the wheel-type air preheater described in Chapter 5. These are used as air preheaters in steam boilers and regenerators in gas turbine cycles. For waste heat recovery applications, these regenerators are called heat wheels. Presently, recuperators are much more common than regenerators for waste heat recovery applications.

To analyze heat exchanger applications, one must use the basic principles of heat transfer. For details of the application of these principles, heat transfer textbooks should be consulted. Here we shall limit ourselves to a brief overview of the three mechanisms of heat transfer, conduction, convection, and radiation; some further material on this heading is in Chapter 6.

The heat transferred by conduction, q, is proportional to the area A normal to the heat flow, and to the temperature gradient dT/dx, in the direction of the heat flow. This equation is written:

$$q = -kA\,(dT/dx) \tag{2.22}$$

The thermal conductivity k is a property of the material through which the heat is being transferred.

Convection, that is, transfer of heat by flow of material, is governed by Newton's law of cooling, written:

$$q = h_c A\,(T_s - T_f) \tag{2.23}$$

Here q and A are as defined as above for conduction. The temperatures T_s and T_f are for the solid surface and the fluid, respectively. The factor h_c is the coefficient of convection. It is dependent on several of the physical properties of the fluid, as well as the geometrical arrangement of the surface and the fluid.

Radiation is the transfer of heat energy by electromagnetic means between two materials whose surfaces "see" each other. The governing equation is known as the Stefan-Boltzmann equation, and is written:

$$q = \sigma F_e F_a A\,(T^4_{abs\,1} - T^4_{abs\,2}) \tag{2.24}$$

Again q and A are as defined for conduction and convection and $T_{abs\,1}$ and $T_{abs\,2}$ are the absolute temperatures of the two surfaces involved. The factor F_e is a function of the condition of the radiation surfaces and in some cases the areas of the surfaces. The factor F_a is the configuration factor and is a function of the areas and their positions. Both F_e and F_a are dimensionless. The Stefan-Boltzmann constant, σ is equal to 0.1714 Btu/h•ft^2•R^4. Although radiation is usually associated with solid surfaces, certain gases can emit and absorb radiation. These include the so-called nonpolar molecular gases such as H_2O, CO_2, CO, SO_2, NH_3 and the hydrocarbons. Some of these gases are present in every combustion process.

It is convenient in certain calculations to express the heat transferred by radiation in the form of

$$q = h_r A \ (T_1 - T_2) \qquad (2.25)$$

where by comparison to eq (3.23) the coefficient of radiation, h_r, is defined as

$$h_r = \frac{\sigma F_e F_a A \ (T^4_{abs\,1} - T^4_{abs\,2})}{(T_1 - T_2)} \ . \qquad (2.26)$$

Note that h_r is still dependent on the factors F_e and F_a, and also on the absolute temperatures to the fourth power. The main advantage is that in eq (2.26) the rate of heat flow by radiation is a function of the temperature difference and can be combined with the coefficient of convection to determine the total heat flow to or from a surface.

Practical heat transfer calculations for heat exchangers are beyond the scope of this book. For information on this subject the reader is referred to reference [3].

References

[1] Moore, F. G., Save Energy in Plant Operations, *Hydrocarbon Processing*, Vol. 52, No. 7, pp. 67-71, July 1973. Used by permission of copyright owner.

[2] Mangiore, B. J. and Petkorsek, J. C., *Economics Favor Industrial Power Generation with Steam from High Pressure Boilers*, ASME Paper 75-IPWR, Industrial Power Conference, Pittsburgh, Pa., May 19-20, 1975.

[3] VanWylen, G. and Sonntag, R. E., *Fundamentals of Classical Thermodynamics*, 2d edition, J. Wiley, New York, 1973.

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[5] Holman, J. P., *Heat Transfer*, 4th edition, McGraw-Hill, New York, 1976.

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Chapter 3

ECONOMICS OF WASTE HEAT RECOVERY

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Introduction

This chapter of the manual deals with concepts and analytical techniques which can guide the industrial plant manager in evaluating the economic efficiency of investments in waste heat utilization. It explains in textbook fashion, and demonstrates in simplified but realistic examples, the use of alternative methods of evaluating and comparing energy-saving investments. The methods which are treated range from very simple, first-level techniques such as determination of the payback period, to more comprehensive techniques such as benefit-cost analysis, life-cycle cost analysis, and the internal rate of return method. The discussion covers the treatment of taxes, inflation, and uncertainty in data estimates and assumptions. The kinds of financial data needed to perform the economic analyses are identified, and the appropriateness of the different evaluation methods for analyzing various kinds of investment decisions is explained.

Some of the methods described here are employed in the case studies chapter which follows this one. Here we examine them along with additional methods for the following reasons: some of the methods used in chapter 4 are not completely explained in terms of how to apply them; the rationale for using the particular methods in the case studies is not always clear; and some of the methods to be presented here are not used in the case studies but appear relevant for evaluating energy-saving investments. Chapter 4 demonstrates evaluations which certain firms have made of the economic efficiency of their investments in waste heat recovery. This chapter, on the other hand, is intended as a reference and guide for plant analysts and decision makers in analyzing their own investment decisions.

Details of a financial analysis will, of course, be unique to each firm. The general procedures and types of data requirements, however, will be essentially uniform for the many different types of firms or industries which might wish to evaluate investment in waste heat recovery. Hence, this chapter is intended to serve the requirements of commercial firms of all types.

The evaluation methods described here are appropriate to treating investment problems of varying complexity. In the case where there is only one type of waste heat recovery equipment available for the job at hand, and little choice of equipment sizes in the relevant range, the investment decision is simple: Should the equipment be obtained or not? In other cases where alternative types and sizes of equipment are possible, the question will be not only whether the equipment should be obtained, but which among the alternative types and sizes is most efficient.

This discussion of investment evaluation techniques presupposes that a prime objective of a business firm is profit maximization.[1] Successful investment in waste heat recovery increases profits generally by reducing fuel costs, and in some cases by generating revenue. It is assumed that the expected profitability of investment in waste heat recovery systems is a critical factor in determining if firms will adopt the systems.

The material is presented in five main sections. The first discusses the kinds of benefits and costs associated with waste heat recovery. The second treats several "partial" (i.e., incomplete), but popular methods for evaluating investment alternatives. These methods, e.g., the payback

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[1] Other possible objectives of a business firm include cost minimization for a given outcome, minimization of risk of loss, maximization of sales, and creation of a desirable public image. Most firms probably have several objectives, but profit maximization is likely to be one of the most important.

method and return on investment method, are explained, illustrated, and evaluated in terms of their advantages and disadvantages.

The third section covers more comprehensive methods for evaluating investment alternatives. These methods take into account costs and benefits over the life of the waste heat recovery system and discount cash flows to a common time for comparison. They require somewhat more data and effort than the partial methods, but generally result in more correct assessments of investment opportunities than do the partial methods.

The fourth section discusses some of the complicating factors which may arise in economic evaluations, including income tax effects, inflation, and risk and uncertainty. Methods of dealing with these factors are outlined and illustrated.

The fifth section points out the different kinds of investment decisions, such as making choices among mutually exclusive projects. It discusses the nature of the decision when one investment project is prerequisite to other activities of the firm, such as might be the case where curtailment of fuel allocations causes investment in waste heat recovery to be needed to prevent production cutbacks. The appropriate evaluation method for each situation is given.

Benefits and Costs

In general, the motivation for industrial firms to invest in waste heat recovery is that they expect the resulting benefits to exceed investment costs. Factors that have recently made such investments attractive are rising fuel costs and curtailment of regular fuel sources which threaten production cutbacks and changeover to other energy sources. In addition, mandatory pollution controls and rising labor costs cut into profits and cause firms to look more closely for ways to control costs.

The kinds of potential benefits which may result from waste heat recovery are listed in table 3.1. These benefits were suggested by a preliminary look at existing applications; however, only one, fuel savings, was found in every case examined. The other benefits, savings in capital and maintenance costs on existing equipment, pollution abatement, labor savings, product improvement, and revenue from sales of recovered heat, appear limited to certain applications.

TABLE 3.1 *Possible benefits from waste heat recovery*[a]

Fuel savings
Reduced size, hence lower capital cost, of heating/cooling equipment
Reduced maintenance costs for existing equipment
Reduced costs of production labor
Pollution abatement
Improved product
Revenue from sales of recovered heat or energy

[a] Not all of these benefits will necessarily result from investment in waste heat recovery; in fact, fuel savings may be the only benefit in many applications. However, examples of the other kinds of benefits shown were found in existing applications.

Fuel savings result when waste heat is recovered and used in substitution for newly generated heat or energy. For example, heat from stack flue gas may be recovered by an economizer and used to preheat the input water, thereby reducing the amount of fuel needed for steam generation.

Savings in capital costs for certain items of existing equipment (i.e., regular equipment apart from that required for waste heat recovery) may be possible if recovered heat reduces the required capacity of the furnace or other heating/cooling equipment.[2] For example, installation of rooftop thermal recovery equipment on buildings with high ventilation requirements can enable significant reductions in the size, and cost, of the building's heating and cooling system. This potential for savings is probably limited to new plant installations, and does not appear to have received much consideration in industrial applications of heat exchangers.

Reduced maintenance and repair on certain items of existing equipment may, in some instances, be a further benefit of investment in waste heat recovery. The principal impact on the maintenance of existing equipment is likely to result from the planning, engineering, and installation phases of investment in waste heat recovery, when the existing equipment and plant processes are often scrutinized. Existing faults may be identified and corrected; and improved maintenance practices may be extended to existing equipment. While these same effects could be achieved independently of waste heat recovery by a separate inspection of the existing equipment, planning for waste heat recovery provides a catalyst for the inspection. Furthermore, cost of the informational gain is probably significantly

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[2] On the other hand, use of heat recovery equipment may increase capital costs of regular equipment by imposing higher temperature loads on it. This effect of increasing costs is included in the listing of costs.

reduced when inspection is performed jointly with the planning for waste heat recovery. Additional effects from waste heat recovery which may reduce maintenance costs include a lowering of the temperature of stack gases.

Another kind of benefit which may result from investment in waste heat recovery is savings in labor costs. Labor savings can result, for example, from a lowering of furnace changeover time (i.e., the time needed to alter furnace temperatures required for a change in production use) by preheating combustion air with waste heat. Savings may also result from faster furnace start-ups, accomplished by similar means. By reducing the amount of labor "downtime," unit labor costs are reduced. (A tradeoff may exist between idling the furnace at higher temperatures during off-duty hours and incurring labor "downtime" during furnace start-ups. If the existing practice is to idle the furnace at high temperatures in order to avoid "downtime," the savings from using waste heat recovery to preheat air would be in terms of fuel reductions rather than lower labor costs.)

Pollution abatement is a beneficial side effect which may result from recovery of waste heat. For example, the pollution abatement process in textile plants will often be facilitated by waste heat recovery. Pollutants (plasticizers) are usually collected by circulating air from the ovens (where fabrics are coated or backed with other materials) through electrostatic precipitators. The air must, however, be cooled to accomplish collection of pollutants. Recovery of waste heat from the air leaving the furnace prior to its entering the precipitators will, consequently, not only provide heat or energy which can be used elsewhere to reduce fuel costs, but will also aid in the collection of pollutants. If it were not for heat recovery, it would be necessary to cool the air by other means, which would generally entail additional fuel consumption. Thus, there is a twofold impact on fuel use from this application of heat recovery.[3] Another instance of pollution abatement as a side effect of waste heat recovery occurs if pollutants are reduced by the higher furnace temperatures resulting from preheating combustion air with waste heat.

The pollution abatement side effects represented by the two preceding examples are distinguishable from the use of systems to recover heat from a pollution abatement process, where recovery of heat does not in itself contribute to pollution abatement. For example, the recovery of waste heat from the incineration of polluting fumes is a method of reducing the cost of pollution abatement by producing a useful by-product from the abatement process. However, the waste heat recovery does not itself contribute to the pollution abatement process and therefore does not yield multiple benefits; the only benefit is the value of the fuel savings from using the recovered heat in other processes.

Product improvement is a further potential side effect of waste heat recovery. For example, by achieving a more stable furnace temperature and a reduction in furnace aeration, use of a recuperator to preheat combustion air may reduce the undesirable scaling of metal products. In absence of preheating combustion air, it would be necessary to invest in improvements to furnace controls or in some other means of preventing scaling; to secure the same product quality.

A final potential benefit from waste heat recovery, as suggested by existing applications, is the generation of revenue from sales of recovered waste heat or energy. In some cases, the recoverable waste heat cannot all be used by the plant itself. Recovery may still be advantageous if there are adjoining plants which are willing to purchase the recovered heat. In this case, the potential benefits are revenue-generating, rather than cost-reducing, and would be measurable in terms of dollars of revenue received.

With the following information—records of past operating levels and expenses, the efficiency of the proposed heat recovery equipment, the level of expected furnace operation, the demand for recycled heat, and the expected price of fuel—it should be possible to predict fairly closely the savings in fuel costs that would result from substituting waste heat for newly-generated heat or energy. Certain of the other potential benefits, such as labor cost savings and product improvement, might be more difficult to estimate.

To evaluate the desirability of an investment, measures of costs are needed to compare with the benefits. Table 3.2 shows the type of costs which may arise in connection with waste heat recovery.

As may be seen, costs may begin before the waste heat recovery system is installed and extend

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[3] Here we consider only the benefits in fuel savings resulting from pollution abatement effects of waste heat recovery, and not the benefits to the surrounding area from cleaner air emanating from the plant. The emphasis is on private benefits and costs, i.e., those accruing directly to the firm, because private decision makers have traditionally not taken into account all social benefits and costs associated with their investment decisions, i.e., those benefits and costs that accrue to society at large. With the advent of environmental impact statements, however, pollution abatement benefits have become more important to private decision makers.

Table 3.2. *Potential costs to consider in investing in waste heat recovery* [a]

Type of costs	Examples of costs
1. Pre-engineering and planning costs	Engineering consultant's fee; in house manpower and materials to determine type, size, and location of heat exchanger.
2. Acquisition costs of heat recovery equipment	Purchase and installation costs of a recuperator.
3. Acquisition costs of necessary additions to existing equipment	Purchase and installation costs of new controls, burners, stack dampers, and fans to protect the furnace and recuperator from higher temperatures entering the furnace due to preheating of combustion air.
4. Replacement costs	Cost of replacing the inner shell of the recuperator in *N* years, net of the salvage value of the existing shell.
5. Costs of modification and repair of existing equipment	Cost of repairing furnace doors to overcome greater heat loss resulting from increased pressure due to preheating of combustion air.
6. Space costs	Cost of useful floor space occupied by waste heat steam generator; cost of useful overhead space occupied by evaporator.
7. Costs of production downtime during installation	Loss of output for a week, net of the associated savings in operating costs.
8. Costs of adjustments (debugging)	Lower production; labor costs of debugging.
9. Maintenance costs of new equipment	Costs of servicing the heat exchanger.
10. Property and/or equipment taxes of heat recovery equipment	Additional property tax incurred on capitalized value of recuperator.
11. Change in insurance or hazards costs	Higher insurance rates due to greater fire risks; increased cost of accidents due to more hot spots within a tighter space.

[a] In addition, attention should be given to the length of intended use, expected lives of related equipment, and the flexibility of alternative equipment to future modification and expansion.

throughout the period of continued plant operation. In most cases, the major cost item is likely to be the acquisition and installation of the heat exchanger, and should be relatively easy to estimate.

It is important that only those costs and benefits which are attributable to an investment be included in the analysis of that investment. For example, if a plant is required by mandate to add a pollution control apparatus, the decision to add a waste heat recovery system to the pollution control system should not be influenced by the costs of the pollution control system. As a further example, costs of equipment replacement or repair not necessitated by the addition of the waste heat recovery system should not be incorporated into the waste heat evaluation, although it may be undertaken jointly for convenience.

Partial Methods of Evaluation

The simplest procedures which are used by firms to try to evaluate alternative kinds and amounts of investments are visual inspection,[4] payback period, and return on investment—approaches which are termed "partial" here because they do not fully assess the economic desirability of alternatives. These partial methods may be contrasted with the more complete techniques, discussed in the following section, which take into account factors such as timing of cash flows, risk, and taxation effects—factors which are required for full economic assessment of investments.

Despite their shortcomings, the partial techniques of analysis may serve a useful purpose. They can provide a first level measure of profitability which is, relatively speaking, quick, simple, and inexpensive to calculate. They may therefore be useful as initial screening devices for eliminating the more obviously uneconomical

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[4] There are some investments whose desirability is apparent merely by inspection, and which do not require further economic analysis. An example is an investment characterized by negligible or low costs and a highly certain return. But actions which require significant initial investment and yield benefits over time—as recovery of waste heat is typically characterized —usually require more extensive analysis than visual inspection.

investments. These partial techniques (particularly the payback method) may also provide needed information concerning certain sensitive features of an investment. But where partial methods are used, the more comprehensive techniques may also be needed to verify the outcome of the evaluations, and to rank alternative projects as to their relative efficiency.

Following are descriptions, examples, and limitations of the payback method and the return on investment method, two of the more popular partial methods.

Payback Method[5]

The payback (also known as the payout or payoff) method determines the number of years required for the invested capital to be offset by resulting benefits. The required number of years is termed the payback, recovery, or break-even period.

The measure is popularly calculated on a before-tax basis and without discounting, i.e., neglecting the opportunity cost of capital.[6] Investment costs are usually defined as first costs, often neglecting salvage value. Benefits are usually defined as the resulting net change in incoming cash flow, or, in the case of a cost-reducing investment like waste heat recovery, as the reduction in net outgoing cash flow.

The payback period is usually calculated as follows:[7]

$$\text{Payback Period (PP)} = \frac{\text{First Cost}}{\text{Yearly Benefits}-\text{Yearly Costs}} \qquad (3.1)$$

For example, the payback period for a furnace recuperator which costs $10,000 to purchase and install, $300/yr on average to operate and maintain, and which is expected to save by preheating combustion air an average of 2000 Mft^3 of burner gas per year at $0.70/$\text{Mft}^3$ (i.e., $1400/yr), may be calculated as follows:

$$\text{PP} = \frac{\$10{,}000}{\$1400-\$300} = 9.1 \text{ yr.}$$

The *disadvantages* of the payback method which recommend against its use as a sole criterion for investment decisions may be summarized as follows:

(a) The method does not give consideration to cash flows beyond the payback period, and thus does not measure the efficiency of an investment over its entire life.

Consider, for example, the two alternative investments A and B, presented in table 3.3. Using the undiscounted payback method, a firm would prefer Investment A, which has a payback period of 1.7 yr, to Investment B, which has a payback of 2.2 yr. Yet, depending upon the true opportunity cost of capital (i.e., the discount rate), Investment B, which continues to yield benefits beyond Investment A, may be a more profitable choice. (For example, with an opportunity cost of 10 percent, Investment A would yield $20,832 in total benefits, and Investment B, $22,383 in total benefits in present value terms.)

(b) The neglect of the opportunity cost of capital, that is, failing to discount costs occurring at different times to a common base for comparison, results in the use of inaccurate measures of benefits and costs to calculate the payback period, and, hence, determination of an incorrect payback period. This problem is illustrated by the example of two alternative investments shown in table 3.4. Payback analysis using undiscounted values would result in indifference between Investments C and D. They both have a payback of 2 yr, and yield total benefits, undiscounted, of $25,000. But because Investment D yields more benefits toward the beginning than Investment C, and thereby allows the investor to realize a larger return on earnings, Investment D would be the preferred choice. In present value terms, with an opportunity cost of 10 percent, Investment C would yield total benefits of $20,697, and Investment D, $21,524.

In short, the payback method gives attention to only one attribute of an investment, i.e., the number of years to recover costs, and, as often calculated, does not even provide an accurate

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[5] The payback method is treated in greated detail in Eugene L. Grant and W. Grant Ireson, *Principles of Engineering Economy*, 4th Ed., pp. 347, 528-529 (New York: The Ronald Press Co., 1970).

[6] The opportunity cost of capital is the return which could be earned by using resources for the next best available investment purpose (e.g., from an investor's standpoint, this might be the earning of interest on savings accounts), rather than for the purpose at hand. It represents an extra return, beyond merely covering other costs, which is necessary in order to make an investment competitive with other opportunities. This concept is discussed further in the following section of this chapter.

[7] This method has the implicit assumption that the expected proceeds from an investment are constant from year-to-year. If expected yearly proceeds are not equal, the customary approach is either (1) to average yearly benefits and costs and use the above formula, or (2) to add the proceeds in successive years until their total equals the first cost.

TABLE 3.3. *An illustration that payback analysis does not take into account cash flows beyond the payback period*

Investment	First Cost	Yearly benefits			Payback period	Total present value benefits[a]
		Year 1	Year 2	Year 3		
Investment A	$20,000	$12,000	$12,000	$0	1.7 years	$20,832
Investment B	20,000	9,000	9,000	9,000	2.2 years	22,383

[a] Calculated for a discount rate of 10 percent, compounded annually.

TABLE 3.4. *An illustration that the undiscounted payback method can result in inaccurate measures*

Investment	First cost	Yearly benefits			Payback period	Total present value benefits[a]
		Year 1	Year 2	Year 3		
Investment C	$20,000	$5,000	$15,000	$5,000	2 years	$20,697
Investment D	20,000	15,000	5,000	5,000	2 years	21,524

[a] Calculated for a discount rate of 10 percent, compounded annually.

measure of this. It is a measure which many firms appear to overemphasize, tending toward shorter and shorter payback requirements. Firms' preference for very short payback to enable them to reinvest in other investment opportunities may in fact lead to a succession of less efficient, short-lived projects.

Despite its limitations, the payback period has *advantages* in that it may provide useful information for evaluating an investment. There are several situations in which the payback method might be particularly appropriate:

(a) A rapid payback may be a prime criterion for judging an investment when financial resources are available to the investor for only a short period of time.

(b) The speculative investor who has a very limited time horizon will usually desire rapid recovery of the initial investment.

(c) Where the expected life of the assets is highly uncertain, determination of the break-even life, i.e., payback period, is helpful in assessing the likelihood of achieving a successful investment. (This use of the payback method is discussed later in the section on risk, uncertainty, and sensitivity analysis.)

The shortcomings that result from failure to discount costs and the omission of important cost items can be overcome simply by using a more accurate calculation of payback. Essentially what is desired is to find the number of years, R, for which the value of the following expression is equal to zero:

$$C = \sum_{j=1}^{R} \frac{B_j - P_j}{(1 + i)^j} \tag{3.2}$$

where

C = Initial investment cost,
B_j = Benefits in year j,
P_j = Costs in year j,
R = Break-even number of years, and
i = Discount rate.

Where yearly net benefits are uneven, an iterative process can be used to determine the solution. If, on the other hand, yearly net benefits are expected to be about uniform, the following formula[8] can be used to facilitate the calculation:

$$R = \frac{-\log\left(1 - \frac{iC}{M}\right)}{\log(1+i)}, \qquad (3.3)$$

where

R = Break-even number years,
M = Yearly net benefits,
C = Initial investment cost, and
i = Discount rate.

Return on Investment Method[9]

The return on investment (ROI) or return on assets method calculates average annual benefits, net of yearly costs such as depreciation, as a percentage of the original book value of the investment.

The calculation is as follows:

Return on Investment (ROI)

$$= \frac{\text{Average Annual Net Benefits}}{\text{Original Book Value}} \times 100. \qquad (3.4)$$

As an example, the calculation of the ROI for an investment in a waste heat economizer is as follows:

Original Book Value = \$15,000
Expected Life = 10 yr
Annual Depreciation,
using a straight-line method = $\frac{\$15,000}{10}$ = \$1500

Yearly Operation, Maintenance
and Repair Cost = \$200

Expected Annual Fuel Oil Savings = \$5000

$$\text{ROI} = \frac{\$5000 - (\$1500 + \$200)}{\$15,000} \times 100$$
$$= 0.22 \times 100 = 22 \text{ percent}$$

The return on investment method is subject to the following principal *disadvantages*, and, therefore, is not recommended as a sole criterion for investment decisions:

(a) Like the payback method, this method does not take into consideration the timing of cash flows, and thereby may incorrectly state the economic efficiency of projects.

(b) The calculation is based on an accounting concept, original book value, which is subject to the peculiarities of the firm's accounting practices, and which generally does not include all costs. The method, therefore, results in only a rough approximation of an investment's value.

The *advantages* of the return on investment method are that it is simple to compute and a familiar concept in the business community.[10]

Comprehensive Methods for Evaluating Investment Alternatives

There are additional methods of financial analysis which avoid the problems of the partial methods by taking into account total costs and benefits over the life of the investment and the timing of cash flows by discounting. The methods of this type discussed here are the present value of net benefits method, the annual net benefits method, the benefit/cost ratio method, and the internal rate of return method. Before defining each of these, let us consider an element common to them, the discounting of costs.

Discounting of Costs

Investment in waste heat recovery systems, like many capital investments, will generally require a number of expenditures spread over a period of time and will result in cost savings (or revenue receipts) also spread over time. To evaluate correctly the profitability of such investments, it is necessary to convert the various expenditures and receipts to a common basis, because dollars spent or received at different times are not of equal value. If, for instance, the firm's earning

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[8] Educational Facilities Laboratories, *The Economy of Energy Conservation in Educational Facilities*, pp. 67-68, Library of Congress Catalog No. 73-83011, (New York: FFL, Inc., 1973).

[9] This method is described in greater detail in the Electric Energy Association's Manual, *The Financial Analysis Section of Alternative Choice Comparison for Energy System Selection*, (New York, n.d.).

[10] A variation of this method is the return on average investment method, in which the average yearly benefit, before taxes and net of depreciation, is divided by the average book value over the life of the project, (defined as the original book value plus the final book value, divided by 2. Thus, with no remaining salvage value at the end of 10 yr, the return on average investment in the example would be 44 percent.

opportunity is 8 percent/yr compound annually,[11] \$1.00 received today would grow to \$1.47 in 5 yr and would about double in 9 yr. Deferral of the receipt of \$1.00 for 5 yr would mean that the earnings on it over the interim 5 yr would not be realized, and it would be worth less than \$1.00 received today. It would, in fact, be equivalent to receiving only \$0.68 today. Therefore, other things equal, a firm will prefer early income and deferred expenditures. A real opportunity cost of capital exists even when there is no inflation[12] or deflation and even when equity (nonborrowed) funds are used, as long as alternative productive investments are possible.

Cash flows occurring at different times can be converted to a common basis, i.e., discounted, by means of discounting formulas (also known as interest formulas). The six basic discounting formulas are shown in table 3.5, along with an abbreviated guide for their use.

The discounting of costs can be greatly simplified by the use of tables of discount factors, such as those in tables 3.10 through 3.16 at the end of this chapter. The discount factors are calculated from the discount formulas for various time periods and opportunity costs (expressed as a rate of discount or interest). In the sample tables provided, factors are calculated for 50 to 100 interest periods (which could be years, months, or any other time interval desired), and for selected discount rates ranging from 6 to 25 percent per period. Factor tables covering other time periods and interest rates are available in most engineering economics textbooks.[13]

To find the comparable value of a given sum of money if it were received (or disbursed) at a different time, the appropriate formula from table 3.5 may be used, or the corresponding discount

Table 3.5. *Discounting formulas*

Standard nomenclature	Use when	Standard notation	Algebraic form
Single compound amount formula	Given P; to find F	(SCA, i%, N)	$F = P(1+i)^N$
Single present worth formula	Given F; to find P	(SPW, i%, N)	$P = F\dfrac{1}{(1+i)^N}$
Uniform compound amount formula	Given A; to find F	(UCA, i%, N)	$F = A\dfrac{(1+i)^N - 1}{i}$
Uniform sinking fund formula	Given F; to find A	(USF, i%, N)	$A = F\dfrac{i}{(1+i)^N - 1}$
Uniform capital recovery formula	Given P; to find A	(UCR, i%, N)	$A = P\dfrac{i(1+i)^N}{(1+i)^N - 1}$
Uniform present worth formula	Given A; to find P	(UPW, i%, N)	$P = A\dfrac{(1+i)^N - 1}{i(1+i)^N}$

Where:

P = a present sum of money.
F = a future sum of money, equivalent to P at the end of N periods of time at an interest of i.
i = a discount or interest rate to reflect the opportunity cost of capital.
N = number of interest periods.
A = an end-of-period payment (or receipt) in a uniform series of payments (or receipts) over N periods at i interest rate, usually annually.

Source: Gerald W. Smith, *Engineering Economy*, p. 47.

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[11] In economic studies, interest often is taken into account as though compounding occurs once a year, although in practice it may occur more frequently. (In the case of business investments, continuous compounding may be more descriptive than annual compounding.) Since the results will, in any case, generally not be appreciably affected by the choice, annual compounding is assumed here for simplicity. For an explanation of continuous compounding and a table of conversion factors, see Gerald W. Smith, *Engineering Economy: Analysis of Capital Expencitures*, 2d ed., p. 541 (Ames, Iowa: The Iowa State University Press, 1973).

[12] Inflation is discussed in the following major section of this chapter.

[13] See, for example, Gerald Smith, *Engineering Economy: Analysis of Capital Expenditures*, and Eugene L. Grant and W. Grant Ireson, *Principles of Engineering Economy*.

factor may be multiplied by the given sum. For example, if a firm's opportunity earning potential is 10 percent annually, \$500 to be received 5 yr from now would be equivalent to receiving \$310 today, (i.e., Present value = \$500 [SPW, i = 10%, N = 5 yr] =

$$\$500 \frac{1}{(1+0.10)^5} = \$500\ (0.6209) = \$310)$$

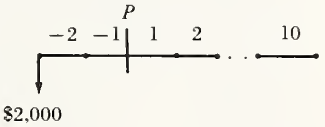
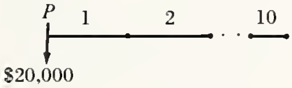
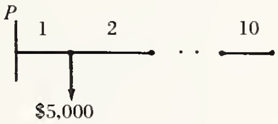
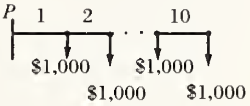
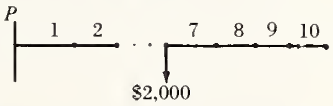
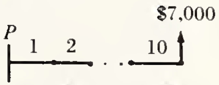
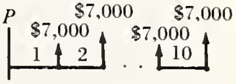
Alternatively, receiving \$500, 5 yr from now, would be equivalent to receiving equal payments of \$82 in each of the 5 yr, (i.e., Annual Payment Equivalent = \$500 [USF, i = 10%, N = 5 yr] =

$$\$500 \frac{0.10}{(1+0.10)^5 - 1} = \$500\ (0.1638) = \$82).$$

Table 3.6 illustrates the conversion to a present equivalent and an annual equivalent of various costs and benefits representative of an investment in waste heat recovery. The examples assume that the investing firm has an opportunity cost of 15 percent and a 10-yr time horizon. The first column gives the kind of cost or benefit incurred. The second column uses a cash flow diagram to describe the pattern of cash outflow or inflow associated with the cost or benefit. The horizontal line of the cash-flow diagram is a time scale, where P indicates the present, the progression of time moves from left to right, and the numbers between the points represent time intervals (in this case, years). The downward arrows represent expenditures (cash outflows), and the upward arrows represent receipts (cash inflows), as viewed from the standpoint of the firm.

Once all costs or benefits are discounted to the present, they may be combined in order to find the total. Thus, in this simplified example, total costs

TABLE 3.6. *Illustrative discounting of repesentative costs and benefits*

Kind of cost or benefit	Cash flow pattern	Present equivalent	Annual equivalent
Planning for waste heat recovery		P=\$2,000 (SCA, 15%, 2 yr) =\$2,000 (1,322) =\$2,644	A=\$2,000 (SCA, 15%, 2 years) (UCR, 15%, 10 years) =\$2,000 (1.322) (0.1993) =\$527
Purchase and installation of recuperator		P=\$20,000 (1) =\$20,000	A=\$20,000 (UCR, 15%, 10 years) =\$20,000 (0.1993) =\$3,986
Cost of production downtime		P=\$5,000 (SPW, 15%, 1 yr) =\$5,000 (0.8696) =\$4,348	A=\$5,000 (SPW, 15%, 1 year) (UCR, 15%, 10 years) =\$5,000 (0.8696) (0.1993) =\$867
Net increase in annual operating and maintenance cost due to the recuperator		P=\$1,000 (UPW, 15%, 10 yr) =\$1,000 (5.019) =\$5,019	A=\$1,000 (1)
Replacement of parts		P=\$2,000 (SPW, 15%, 7 yr) =\$2,000 (0.3759) =\$752	P=\$2,000 (SPW, 15%, 7 years) (UCR, 15%, 10 years) =\$2,000 (0.3759)(0.1993) =\$150
Salvage value of heat recovery equipment, end of 10 years		P=\$7,000 (SPW, 15%, 10 yr) =\$7,000 (0.2472) =\$1,730	A=\$7,000 (USF, 15%, 10 years) =\$7,000 (0.0493) =\$345
Annual fuel savings, (assuming no change in future fuel prices) [a]		P=\$7,000 (UPW, 15%, 10 yr) =\$7,000 (5.019) =\$35,133	A=\$7,000 (1)

[a] Evaluation of costs and benefits when price changes are expected is discussed in the following major section.

of the investment amount to $31,033 in present value terms, and total present value benefits amount to $35,133 over the 10 yr (found by aggregating all the costs and all the benefits in the Present Equivalent Column of table 3.6). This is equivalent to incurring costs of $6185 annually and realizing benefits of $7000 annually for an annual benefit of $815 in each of the 10 yr (found by aggregating the costs and the benefits in the Annual Equivalent Column).

Net Present Value (Net Benefits) Method

This method calculates the difference between the present value of the benefits and the costs resulting from an investment. The difference between benefits and costs is the net present value of the investment. A positive net present value means that the financial position of the investor will be improved by undertaking the investment; a negative net present value means that the investment will result in a financial loss.[14]

To use this method in the evaluation of waste heat recovery investments, the benefits would be defined as positive cash flows (as, for example, would result from sales of surplus, recovered waste heat), and/or reductions in cash outflows (as would result from substitution of recovered waste heat for newly-generated heat).[15]

The formula for calculating net present value, or net benefits, is as follows:

$$\mathrm{NPV} = \sum_{j+1}^{n} \frac{(S_j + R_j) - (I_j - V_j + M_j)}{(1+i)^j}, \quad (3.5)$$

where

NPV = Net present value benefits,
n = Number of time intervals over which the investment is analyzed,
S_j = Energy cost savings in year j,
R_j = Revenue from sale of excess energy received in year j,
I_j = Investment costs in year j,
V_j = Salvage value in year j,
M_j = Maintenance and repair costs in year j, and
$\frac{i}{(1+i)^j}$ = Single present value discount formula.

The acceptance criteria of a project, as evaluated with the net present value method, are that (1) only those investments having positive net benefits will be accepted (unless the project is mandatory), and (2) when selecting among mutually exclusive investments, the one with the highest positive net benefits will be chosen (or the one with the lowest negative net benefits if none of the alternatives has positive net benefits and the project is mandatory).

In using the net present value method to compare alternative investments, it is important to evaluate the costs and benefits of each alternative over an equal number of years. This may be done in any of the following ways, depending upon the nature of the investment. (1) The costs and benefits can be measured over a time period that is a common multiple of the economic lives of the alternatives. For example, to compare heat exchange A with a life of 5 yr against heat exchanger B with a life of 10 yr, alternative A could be evaluated on the basis of one replacement, and alternative B, on the basis of no replacements, such that benefits and costs of both systems would be computed for 10 yr. (2) Alternatively, benefits and costs of each alternative can be calculated in annual cost terms, based on 5 yr for A and 10 yr for B, and the annual benefits and costs can then be used to calculate the present value of benefits and costs for the desired number of years of service (for example, 12 yr service for both systems).[16] This avoids the need to find a common multiple of system life. (3) If either system will be used for only a limited period of time, which is less than a common multiple of the economic lives of the alternatives, the estimated cash flows associated with each system over the period of analysis can be discounted to present value, making sure to take into account the expected remaining value of each system at the time of terminated use. For example, the problem might be to choose between

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[14] The net benefits of a given project will be greatest at that level where marginal benefits (the additional benefits from the last increment of the investment) are just equal to the cost of the investment at the margin, i.e., where the *marginal* benefit-cost ratio is 1.

[15] The definition of cost reductions as benefits is a useful means of converting the data for an investment for which cash flows are all negative into a format to which benefit-cost methods can be applied.

[16] Since annual cost in this procedure is based on the economic life of the system, the procedure assumes that if the point of termination of use is different from the economic life of the system, the full value of the remaining life is realizable in terms of salvage value.

heat exchanger A, with its economic life of 5 yr, and heat exchanger B, with its economic life of 10 yr, where intended use of either would be only 7 yr due to expected closure of the plant at the beginning of the eighth year. System A would require one replacement because its expected life is only 5 yr and it is needed 7 yr. The remaining value of the replacement after 2 yr of use would be discounted to present value and deducted from present value cost of the system. Similarly, an estimate would be required of the value of the remaining 3 yr of life of system B. If removal costs are prohibitive, or if there is no good resale market for the equipment, the remaining value of both systems at the end of 7 yr should be evaluated at zero dollars. This holds true even though the equipment could provide additional years of service if the existing operation were continued. (4) An investment can also be evaluated on the basis of a perpetual, or indefinite period of use by "capitalizing" renewal costs and expected benefits. Present value benefits of an investment in perpetuity are calculated by dividing the expected annual benefits by the discount rate. Present value costs in perpetuity are calculated by converting all costs other than first costs to an annual equivalent, dividing by the discount rate, and adding the amount to the first cost. Thus, if first costs were \$5000; operation costs \$1000 yearly; and renewal costs \$4000 every 5 yr, the present value capitalized cost of the above heat exchanger A in perpetual service would be equal to \$5000 + (\$1000 ÷ i) + [\$4000 (USF, i, 5 yr) ÷ i].

The choice among these four approaches to measuring present value is often not critical to the outcome; the particular nature of the investment will generally determine which approach is used. For example, in the case of a short-lived investment with unrecoverable salvage, the third approach explained above would be preferred.

The use of the net present value method to compare alternative investments in waste heat recovery equipment is illustrated in table 3.7. It is assumed that the investment choice is between Plan X, the addition of a large recuperator designed to preheat combustion air to a high temperature, and Plan Y, the addition of a smaller recuperator capable of a lesser amount of preheating but sized to avoid the replacement of fans, etc., on the existing equipment. It is further assumed that Plan Y has longer expected economic life due to lower temperatures.

TABLE 3.7. *Illustration of net present value method*

Investment alternatives	First cost	Expected economic life	Annual fuel savings	Net present value [a]
Plan X	\$20,000	6 years	\$7,000	\$12,847
Plan Y	12,000	10 years	5,000	16,256

[a] Based on a discount rate of 10 percent, an investment horizon of 8 years, and the assumption that the full value of any remaining life after 8 years can be salvaged.

Plan Y is found to provide \$3409 more in net benefits than Plan X. The additional fuel savings from the larger recuperator is, therefore, not sufficient to justify its additional costs.

In the case where alternative investments are expected to provide the same level of benefits, the net present value method becomes equivalent to a net present cost method, often referred to as life-cycle costing, or cost-effectiveness analysis. In this case, the most efficient alternative may be identified as the one with the least present value of costs alone. This approach is also often used when the benefit levels cannot be quantified.

A feature of the net present value method which may be a *disadvantage* in some applications is that, in focusing only on net benefits, it does not distinguish between a project involving relatively large benefits and costs, and one involving much smaller benefits and costs, as long as the two projects result in equal net benefits. Thus a project requiring \$10,000 in present value costs, and resulting in \$11,000 in present value benefits (i.e., \$1000 in net benefits) will yield the same net benefits as a project costing \$1000 and resulting in \$2000 in total benefits (again \$1000 in net benefits). A way to avoid this problem is to compute benefit-to-cost ratios for further evaluation of the projects. Choices will be most efficient if independent projects are chosen in the order of their benefit-cost ratios, starting with the highest and working down until the budget is exhausted. (A later section of this chapter, entitled "Application of Evaluation Methods to Different Kinds of Decisions," discusses the appropriateness of each evaluation method to different decisions and suggests ways to avoid potential problems.)

Another possible disadvantage of the net present value method is that the results are quite sensitive to the discount rate, and failure to select the appropriate rate may alter or even reverse the efficiency ranking of the alternatives. For example, with too low a rate an alternative with benefits spread far into the future may unjustifiably appear more profitable than an alternative whose benefits are more quickly realized but of lower amount in undiscounted terms. Since changing the discount rate can change the outcome of the evaluation, the rate used should be considered carefully.

As was explained earlier, the discount rate which a firm should use to discount cash flows of an investment is the firm's opportunity cost of capital, expressed as an interest rate; i.e., the rate of return which will be foregone by using the funds (resources) for the investment under consideration instead of the next best investment opportunity available to it.

If the firm is uncertain as to the appropriate discount rate to use, it may wish to compute the net benefits of an investment based on several alternative discount rates to test for sensitivity of the outcome to the choice of rates.[17]

The net present value method has the *advantage* of measuring the net effect of an investment over its life, taking into account the opportunity cost of capital. The method is particularly useful for determining the efficient scale or size of an investment project.

Net Annual Value (Net Annual Benefits) Method

This method takes essentially the same form as the net present value method. The difference is that all costs and benefits of the net annual benefits method are converted to a uniform annual basis instead of to present value. The formula is as follows:

$$A = \sum_{j=1}^{n} \frac{(S_j + R_j) - (I_j - V_j + M_j)}{(1+i)^j} \; \frac{i(1+i)^n}{(1+i)^n - 1} \qquad (3.6)$$

where

A = Annual value of net benefits,

n = Number of time intervals over which the investment is analyzed,

S_j = Energy cost savings in year j,

R_j = Revenue received in year j,

I_j = Investment costs in year j,

V_j = Salvage value in year j,

M_j = Maintenance and repair costs in year j,

$\frac{1}{(1+i)^j}$ = Single present value discount formula,

and

$\frac{i(1+i)^n}{(1+i)^n - 1}$ = Uniform compound amount formula.

If alternative investments have different life expectancies, either of two approaches may be taken to compare the alternatives; (1) It may be assumed that whichever alternative is chosen will be needed for an indefinite period of time and, hence, will be renewed as needed. In this case, the annual cost of each system may be simply calculated for its expected economic life, regardless of the fact that the lives of the alternatives may be different. (2) If the use of the investment alternatives is required for only a limited time, it is necessary to calculate costs based on the planned investment period, estimating the salvage value of each alternative at the planned time of terminated use. (This is similar to the third approach described under Net Present Value.)

With the cash flows of the preceding example (described in table 3.7), and the assumption that either plan will be used for an indefinite period of time, the net annual value of Plan X is $2408, and of Plan Y is $3047. As we would expect, this method shows Plan Y to be the most efficient choice; as did the present value example.

This method, like the net present value method, has the *disadvantage* of failing to distinguish between projects of unequal magnitudes which yield equal net benefits. However, analysis of benefit-cost ratios can be used to overcome this problem. A possible *advantage* of the net annual value method, as compared with the net present value method, is that the concept of an equivalent annual amount may be easier to understand than the concept of a present equivalent of all cash flows over the period of analysis.

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[17] For a more detailed discussion of the appropriate value of i, see Gerald W. Smith, *Engineering Economy: Analysis of Capital Expenditures*, pp. 452-467, 582-588.

Benefit/Cost Ratio Method[18]

The benefit/cost ratio method expresses benefits as a proportion of costs, where benefits and costs are discounted to either a present value or an annual value equivalent. The formula (with benefits and costs discounted to present value) is as follows:

$$B/C = \sum_{j=1}^{n} \left[\frac{S_j + R_j}{(1+i)^j}\right] \Big/ \sum_{j=1}^{n} \left[\frac{I_j - V_j + M_j}{(1+i)^j}\right] \quad (3.7)$$

where

B/C = Benefit-cost ratio,
n = Number of time intervals over which the investment is analyzed,
S_j = Energy cost savings in year j,
R_j = Revenue received in year j,
I_j = Investment costs in year j,
V_j = Salvage value in year j, and
M_j = Maintenance and repair costs in year j.

While the net present value and the net annual value methods require that discounted benefits minus discounted costs be positive in order for an investment to be worthwhile, the benefit/cost ratio method requires that the ratio of discounted benefits to costs be greater than 1.

A *disadvantage* of the benefit/cost ratio method is that the ratio is influenced by the decision as to whether an item is classified as a cost or as a disbenefit, i.e., whether it appears in the numerator or denominator of the ratio. For many cost or benefit items, this is simply an arbitrary decision, but one which can lead to confusion as to the real efficiency of a project.

Another problem with the benefit/cost ratio method is that it is subject to be misapplied in determining the efficient scale of a given project. It pays to expand a project up to the point that the ratio for the last increment of the investment is equal to 1.0, assuming no alternative investment is available with a higher ratio. Because the benefit/cost ratio for the overall investment declines as the investment is expanded towards the most efficient level, a smaller less efficient size of a project may have a higher ratio for the total investment than a larger more efficiently sized project. This problem can be avoided by applying the benefit/cost ratio method to evaluate the efficiency of increments of an investment, rather than the total investment.

Internal Rate of Return Method

This method (not to be confused with the ROI method evaluated earlier) calculates the rate of return an investment is expected to yield. This may be contrasted with the net present value, the net annual value, and the benefit/cost ratio methods which calculate the net dollar value of the investment based on a predetermined required rate of return. The internal rate of return method expresses each investment alternative in terms of a rate of return (a compound interest rate). The expected rate of return is the interest rate for which total discounted benefits become just equal to total discounted costs, i.e., net present benefits or net annual benefits are equal to zero, or for which the benefit/cost ratio equals one. The criterion for selection among alternatives is to choose the investment with the highest rate of return.

The rate of return is usually calculated by a process of trial and error, whereby the net cash flow is computed for various discount rates until its value is reduced to zero. As a simple example, let us calculate the internal rate of return for a heat exchanger which will cost \$10,000 to install, will last 10 yr, and will result in fuel savings of \$3000 each yr.

We wish to find that i which will equate the following:

$$\$10{,}000 = \$3000\ (\text{UPW},\ i = ?,\ 10\ \text{yr})$$

To do this, we calculate the net present value (NPV) for various i values, selected by visual inspection:

$$\begin{aligned} \text{NPV}\ \ 25\% &= (\$3000)\ (3.571) - \$10{,}000 \\ &= \$10{,}713 - \$10{,}000 \\ &= \$713 \\ \text{NPV}\ \ 30\% &= (\$3000)\ (3.092) - \$10{,}000 \\ &= \$9276 - \$10{,}000 \\ &= \$724 \end{aligned}$$

For i = 25 percent, net present value is positive; for i = 30 percent, net present value is negative. Thus, for some discount rate between 25 and 30 percent, present value benefits are equated to

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[18] For a more detailed treatment of the benefit/cost ratio method, see Eugene L. Grant and W. Grant Ireson, *Principles of Engineering Economy*, pp. 138-143.

present value costs. To find the rate more exactly, without the benefit of a complete set of discount tables, one may interpolate between the two rates as follows:

$$\begin{aligned} i &= 0.25 + 0.05 \left(\frac{\$713}{\$703 - (-\$724)}\right) \\ &= 0.275, \text{ or } 27.5 \text{ percent.} \end{aligned}$$

To decide whether or not to undertake this investment, it would be necessary for the firm to compare the expected rate of return of 27.5 percent with its minimum attractive rate of return.

Use of this method has the *advantage* of generally resulting in conclusions consistent with the three other comprehensive methods discussed in this section. However, there are several possible *disadvantages* which might arise. For one thing, under certain circumstances there may be either no rate of return solution or multiple solutions.[19] Secondly, confusion may arise when this method is used to choose among mutually exclusive alternatives. For example, if the compared alternatives are different sizes of the same project (e.g., different capacity heat exchangers), the rate of return on the larger scale project may be lower than on the smaller scale project, causing the larger to appear less efficient than the smaller. However, additional investment in the larger project may nonetheless yield a positive rate of return in excess of the minimum attractive rate of return. This problem, which is comparable to the problem which was described for the benefit/cost method, can be avoided by analyzing incremental changes in an investment. A third problem is that the rate of return may be somewhat more cumbersome to calculate than the other methods.[20]

Special Factors to Consider in Investment Analysis

This section discusses the treatment of income taxes, inflation, and uncertainty in investment analysis.

Treatment of Income Taxes[21]

Income taxes can have a profound impact on optimum investment decisions. By changing the effective values to the firm of the revenues and costs associated with an investment, taxes can reverse the relative profitability of alternative projects as evaluated apart from taxes, and they can alter the optimal size of investments. Taxes are, therefore, an important element in the economic evaluation of investment in waste heat recovery systems.

If, for example, an investment results in increased taxable revenue, the effective value of the additional revenue to the firm is reduced by taxation. Over the life of an investment, the effective present value of after-tax revenue ($\bar{R}$) can be calculated as follows:

$$\bar{R} = \sum_{j=1}^{n} \frac{R_j(1 - t_j)}{(1 + i)^j} , \qquad (3.8)$$

where

n = the number of time intervals over which the investment is analyzed,

R_j = the taxable revenue resulting from the investment in year j, and

t_j = the firm's tax rate at the margin in year j,

i = the discount rate.

Similarly, if an investment results in benefits in the form of cost savings, the after-tax value of the benefits will be lessened by the loss of any related income tax deductions; i.e., the cost savings are generally not tax free. The present value of after-tax cost-savings ($\bar{C}$) from an investment can be calculated as follows:

$$\bar{C} = \sum_{j=1}^{n} \frac{C_j \ (1 - a_j t_j)}{(1 + i)^j} , \qquad (3.9)$$

where

C_j = the cost savings from the investment in year j,

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[19] An example of a type of investment which gives rise to multiple solutions is one characterized by a net benefits stream which is first negative, then positive and finally negative again.

[20] Computer programs, however, may be used to facilitate the computations. For example, a FORTRAN program for solution of the internal rate of return appears in E. Paul DeGarmo, *Engineering Economy*, 4th Edition, Appendix C, pp. 476-7 (New York: The Macmillan Co., 1969).

[21] Thorough treatment of taxation is beyond the scope of this chapter. The level of detail given here is not intended for computation of a specific firm's actual tax liabilities, but rather to demonstrate the general approach. Property taxes were earlier identified as a type of cost in table 3.2, and are not discussed in this section. They would enter into the *income tax* analysis only as a possible deductible item for purpose of computing after-tax costs.

a_j = that percentage of the cost savings which were deductible from taxable revenue, and

t_j = the firm's tax rate at the margin in year j.

On the other hand, investment costs are also reduced somewhat by the value of corresponding tax deductions. Effective after-tax costs in present value terms can be calculated from eq (9), by redefining C, C_j, and a_j in terms of "costs" rather than "cost savings."

An economic analysis of investment in waste heat recovery systems might involve each of these three tax effects. For instance, system equipment costs may give rise to tax deductions in terms of depreciation allowances and/or investment tax credits; savings in fuel costs will reduce deductions from taxable income of current operating costs; and sale of excess heat or energy to other firms will result in additional taxes on the revenue received.

The following example compares the before-tax and after-tax annual net benefits from an investment in a recuperator, given the assumptions below.

Given

Investing Firm's Tax Rate, at the Margin: 40%
Opportunity Cost of Capital: 10%, after taxes

Source of Capital: Equity Funds

Method of Equip. Depreciation: Straight-line Method

First Cost of Recuperator: $25,000
Annual Maintenance Cost: $500
Expected Salvage Value in 10 yr: $0
Expected Annual Fuel Savings: $5000

Benefits

Annual Reductions in Fuel Costs, Before Taxes = $5000
(minus)
Annual Value of Tax Deductions Lost = 0.40 ($5000) = 2000

(equals)
Annual Reductions in Fuel Costs, After Taxes = $3000

Costs:

Annual Capital Cost of Equipment = $25,000 (UCR; 10%; 10 yr) = $25,000 (0.1628) = $4070
(plus)
Annual Maintenance Cost = 500

(equals)
Total Annual Increase in Equipment and Maintenance Costs, Before Taxes = $4570
(minus)
Annual Value of Tax Deductions of Equipment and Maintenance Cost = 0.40 ($25,000/10 + $500) = $1200

(equals)
Annual Increase in Equipment and Maintenance Costs, After Taxes = $3370

Net Benefits: Before and After Taxes:

Net Benefits of Investment, Before Taxes = ($5000 − $4570) $ 430
Net Benefits of Investment, After Taxes = ($3000 − $3370) − $ 370

In this case, before-tax evaluation of the investment in the recuperator indicates a positive annual net cost savings of $430. But the after-tax evaluation indicates a negative cost savings of $370, which means the investment would not be profitable.

Even though depreciation is not itself a cash flow, it affects the firm's cash flow through the deductions from taxable income. By its choice of depreciation methods (e.g., the declining balance method as opposed to the straight-line method of depreciation), an investing firm can affect the timing of the tax deduction and thereby influence the effective amount of income tax payments. The choice of a depreciation method which yields higher depreciation allocations in the early years will increase the present value of depreciation deductions, thereby decreasing the present value of income tax payments and raising the profitability of the investment to the firm.[22]

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[22] The choice of a depreciation method (i.e., the desired pattern of tax deduction) is also influenced by such factors as the presence or absence of carry-forward losses, the expected pattern of revenue, and the expected future tax rate. Hence, the most rapid depreciation may not always be most profitable.

Inflation

For simplicity, changes in price were not considered in the preceding illustrations of methods of evaluation and treatment of taxes. However, price changes may be very important in an investment analysis.

To take price changes into account, it is useful to distinguish between two kinds of price changes. One is a "nominal" price change which results from changes in the purchasing power of the dollar, i.e., inflation or deflation. The amount of this nominal price change is indicated by the change in the general level of prices as measured by price indices.[23]

The other type of price change which can occur is a change in "real" terms. This means that the price of a given good or service rises proportionately more or less than the change in the general purchasing power of the dollar, such that its price changes over time relative to the prices of other goods and services. For example, given that the general price index rose over a 5-yr period from 1.00 to 1.30, the price of a piece of equipment which also rose 30 percent over the same period, would in effect remain constant in terms of its price relative to prices in general; i.e., its price would change in nominal terms but would remain constant in real terms.

An approach often followed in investment analysis is to assume that all costs and revenues inflate at the same general rate, and that they therefore remain constant in real terms. With this assumption, renewal costs and other future expenses and benefits are evaluated at present prices. If there is reason to believe that certain items of costs or benefits will not inflate at the general rate, then their future values are adjusted only for the estimated *real* change, i.e., the effective change after taking into account the change in the general price level. If, for example, oil prices were expected to rise at a rate of 6 percent a year and the general price index to rise at a rate of 3 percent a year, the imputed real rise in oil prices would be 3 percent, i.e., half the rise would be considered due to general inflation, and half due to changing demand and supply conditions for oil relative to other commodities. Alternatively, if the future price of a good or service were, say, fixed at current levels by lease arrangements (without a cost escalation clause) the price would decline in real terms, in the face of inflation. It would be necessary to adjust the future payments by a price index prior to discounting them to present value in order to express them in constant dollars. Employing this procedure, the appropriate discount rate is a real rate, that is, one from which the inflation factor has been removed.

Assuming constant future prices can greatly simplify an analysis and in many applications will result in reasonably accurate results. However, the success of the approach rests not only on the assumption that future receipts and expenditures will respond fully and evenly to inflation, but also on the assumptions that tax considerations and the source of investment funds do not importantly affect the outcome—two assumptions which may be quite unrealistic for private investments.

The real after-tax return to the firm may be substantially changed by inflation, even if pre-tax investment receipts and costs are assumed fully and equally responsive to inflation. Other things equal, inflation will tend to have a detrimental effect on an investment financed principally by equity funds, i.e., nonborrowed funds. Among the reasons are the following: (1) tax deductions for depreciation are unresponsive to inflation; (2) terminal value of equipment *is* responsive to inflation and will be reflected in the capital gains tax; (3) tax deductions for interest on the borrowed portion of capital are unresponsive to inflation, such that the present value of the deductions diminishes overtime; (4) inflation in receipts tends to move a firm into higher tax brackets.[24]

However, in the case of an investment funded by a large proportion of borrowed funds it is possible that the after-tax return to equity may be maintained, or even improved, other things equal, in the face of inflation due to the following factor: The real value of the debt charges (amortized principle and interest) to finance the investment may decline because of the fixed nature of these payments. This gain to the borrower may equal or exceed his loss due to the fixed nature of depreciation and interest deductions.

From a practical standpoint, methods of simplifying the treatment of inflation are desirable

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[23] Price indices, given in relation to a base period of 1967 (i.e., 1967 prices = 100), are reported monthly by the U.S. Bureau of Labor Statistics in the U.S. Department of Labor publication, entitled *The Monthly Labor Review*.

[24] At the same time, the move into higher tax brackets would tend to increase the value of a given deduction from taxable income.

whenever they can be implemented without significantly altering the results. We have tried here to indicate some of the important considerations in treating inflation.[25]

Uncertainty

Evaluation results depend directly on both the data estimates and the assumptions employed in the analysis. Among the key factors affecting the outcome of evaluations of waste heat recovery systems are the following: (1) the cost estimates for planning, purchasing, installing, and operating and maintaining the heat recovery systems; (2) the additional costs imposed by the investment, such as labor downtime and production loss; (3) the future rate of real price escalation in energy sources; (4) the amount of usable energy recovered; (5) the economic lives of system components, the length of intended use of the system, the salvage values at termination; and (6) the discount rate used to convert future costs and benefits to a common time.

There will often be uncertainty as to the correct values to use in evaluating an investment. Uncertainty, which can be defined broadly as disparity between the predicted and the actual, encompasses two specific concepts: "risk," an event whose probability of occurrence can be predicted; and "uncertainty," an event whose expected chance *of occurrence* can not be predicted.

Following are brief descriptions of three general approaches to dealing with uncertainty (used in the broad sense): (1) probability analysis, (2) sensitivity analysis, and (3) break-even analysis.

Probability analysis is generally used for situations for which the probability of an expected occurrence can be estimated; i.e., for evaluating risk. By multiplying the probability that an event will occur by the resulting dollar value if it does occur, it is possible to express costs and benefits as "expected values," rather than simply as "point" estimates.[26]

For example, cost of production downtime during installation may be roughly estimated at, say $2000, but perhaps the following might be a more accurate statement of costs:

Possible Situation	Probabilities	Cost if Situation Develops	Expected Cost
No difficulties in installation	0.20	$1500	$300
No serious difficulty	0.70	3000	2100
Serious difficulty	0.10	7500	750
			$3150

A more accurate measure of cost in this situation would be $3150.

Probability or expected value analysis, therefore, provides a method of incorporating uncertainty into the investment evaluation in a quantitative way. It requires, however, determination of probabilities.[27]

Sensitivity analysis can be used to assess the consequences of assuming alternative values for the significant variables in the analysis. By determining the effect on the outcome of potential variation in a factor, the analyst identifies the degree of importance of that estimate or assumption and can then seek more information about it if desired. For example, the profitability of a furnace recuperator might be tested for sensitivity to the expected utilization rate of the furnace.

Break-even analysis, a third technique for dealing with uncertainty, focuses on a single key variable which is regarded as uncertain, and calculates the minimum (or maximum) value of the variable which is required to achieve a given outcome. For example, one might solve for the rate of escalation in fuel prices required for an investment in a heat exchanger to break even, other things given. To find the break-even escalation rate, an equation is developed which equates capital and maintenance costs with fuel cost savings. The "uncertain variable," i.e., the fuel escalation rate, is entered as an unknown, and the equation is solved for the break-even rate.[28]

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[25] For a more comprehensive treatment of the effect of anticipated inflation on investment analysis, see Gerald Smith, *Engineering Economy*, Appendix G, pp. 545-552.

[26] The expected cost or benefit (B) of an event that may have any one of N outcomes can be calculated as

$$E = \sum_{j=1}^{N} [P(X_j) \bullet V_j]$$

where

$P(X_j) =$ Probability that the outcome will be X_j, and
$V_j =$ Cost (or benefit) of outcome X_j.

[27] For further explanation of expected value analysis, see H. Raiffa, *Decision Analysis*, (Reading, Mass.: Addison-Wesley, 1968). For a discussion of present value calculations when both cash flow amounts and timing are uncertain, see D. Young, L. E. Contreras, et al., "Expected Present Worths of Cash Flows Under Uncertain Timing," *The Engineering Economist*, Vol. 20, No. 4, pp. 257-267 (Summer 1975).

[28] Note that calculation of the payback period is one variation of break-even analysis.

Application of Evaluation Methods to Different Kinds of Decisions

As table 3.8 shows, the manager of a plant may be confronted with different kinds of investment decisions. The nature of the decision will influence which method of evaluation is preferred.

One kind of decision is whether to accept or reject a particular investment project—where acceptance may mean implementation without further analysis or designation of the project as potentially profitable and worthy of further evaluation against other profitable projects. A simple accept/reject decision might be made, for example, if there were only one kind and size of waste heat recovery system feasible for a particular plant application. Any of the comprehensive evaluation methods described in this chapter generally could be used to determine if the expected benefits (or cost savings) from the investment would exceed the expected costs.

Another kind of decision is choosing among mutually exclusive projects, where any of them would be expected to yield positive net benefits. This is the kind of choice which arises when doing one project precludes doing the others. For example, it might be determined that waste heat recovery would be desirable in a particular furnace via either a heat wheel or a recuperator. Since only one method of heat recovery would be used, a choice must be made between them.

Determining the efficient scale or size of investment in a given project is a special case of mutual exclusions, in that choosing one size for a project usually means rejecting the other possible sizes. The choice of time for undertaking a single project, that is, whether to begin it now or to delay it, is another special case of mutual exclusion.[29]

The appropriate approach to evaluating mutually exclusive projects is generally to determine the choice which maximizes net benefits by using either the present value or annual net benefit method of evaluation.[30] Assuming that there is no budget constraint, the project which maximizes net benefits is the most efficient choice.

Yet another kind of decision that the manager may face is the choice among a set of project alternatives which are independent, that is, projects where the undertaking of one bears no influence on the acceptance of the others. In face of budgetary and other constraints, it is usually impossible for a firm to carry out all of its potentially profitable investment opportunities; a ranking mechanism is needed to identify those projects which are most profitable. For example, investment in heat recovery equipment may compete for funding with other cost-saving or revenue-generating investments.

When there is a budget constraint, the ranking of individual independent projects according to

TABLE 3.8. *Kinds of investment decision problems*[a]

Decisions	Examples
To accept or reject a given project	Project A is accepted if its level of profitability meets the minimal acceptable level.
To choose between mutually exclusive projects	If Project A is accepted, Project B is rejected; if size 1 of Project A is accepted, size 2 of Project A is rejected.
To decide priority among independent projects	Project A is preferred to Project B, but both are undertaken if funding is adequate
To determine the desirability of a prerequisite project	Project A yields a low return, but Projects B and C which yield high returns cannot be undertaken until A is done
No decision needed—mandatory project	Project A is undertaken regardless of its expected return in order to meet mandated requirement

[a] Here for simplicity the kinds of investment decisions are classified in an absolute sense. In practice they are usually in relative degree rather than absolute. For example, projects may be imperfect substitutes for one another rather than perfectly mutually exclusive; projects are seldom perfectly independent since capital usually must be rationed and manpower may be overlapping; projects may be imperfect complements in that acceptance of one *encourages* acceptance of another, rather than complete prerequisites where acceptance of one project *requires* acceptance of another.

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[29] For a discussion of the optimal time-phasing of projects, see Dasgupta and Pearce, *Cost-Benefit Analysis: Theory and Practice* (Great Britain: Harper and Row Publishers, Inc., 1972), pp. 162-163.

[30] Either the benefit/cost ratio method or the internal rate of return method can usually also be used to determine the efficient project size, provided these methods are applied to the analysis of the increments in investment rather than to the total investments. If there are no budget constraints, the most efficient scale for a given project is that for which the benefit/cost ratio at the margin is 1.0. If, however, carrying one or more projects to the scale where their marginal benefit/cost ratio is 1.0 means foregoing other projects whose benefit/cost ratios are greater than 1.0, then each of the projects should be scaled so that their benefits at the margin are the same for an additional dollar of expenditure.

their net benefits cannot be relied upon to result in selection of that subset of projects from the group which will yield the overall maximum net benefits. This is because a project that yields higher total net benefits than another project may yield lower benefits per last dollar of investment spent, such that combined benefits for the two projects could be increased by transferring funds from the first project to the second. Choosing among independent projects according to either the benefit/cost ratio or the internal rate of return criterion until the available budget is exhausted will yield the maximum total net benefits for that budget. For example, table 3.9 shows a possible gain of $40,000 by using the benefit/cost ratio method rather than the net benefits criterion to rank Projects 1, 2, and 3, when the budget is fixed at $200,000.

A fourth kind of decision is determining the desirability of a project when it is a prerequisite to the undertaking of other projects. This type of decision may be particularly relevant to the evaluation of waste heat recovery systems, in that investment in waste heat recovery may in some cases be necessary to preserve the overall operation of a plant. This situation might arise in the face of severe curtailment of fuel allocations. Without some means of stretching available fuel supplies, plants might have to reduce production levels and lay off workers, thereby causing substantial loss of revenue, without equal reductions in costs. Alternatively, investment in waste heat recovery might be necessary in order to enable desired expansion of production in the face of short supplies of fuel, or to avoid costly changeover to substitute sources of energy.

If the project in question is a prerequisite for other activities of the firm, the evaluation of the prerequisite project should take into account the overall return expected not only from it, but also from the other activities whose success depends upon it. Even if the project shows a very small or negative rate of return when evaluated in isolation, it may in fact produce a very high rate of return in terms of enabling other profitable activities to be undertaken. If the overall return to investment on the group of interdependent projects is attractive, funds should be allocated first to the prerequisite project.

In most cases, any of the comprehensive evaluation methods would be adequate to assess the importance of a project that is prerequisite to other functions of the firm. The essential factor is that all relevant costs and benefits attributable to the investment be included in the evaluation.

The final entry in table 3.8 does not involve a decision. It is the case in which there is no decision to be made because the investment project must be undertaken by mandate, regardless of its profitability to the firm. Legislated requirements for specified levels and/or types of pollution control investments are examples. An important point to remember in this case is that other investments which are being considered, perhaps in conjunction with the required investment, should not be saddled with its costs and/or benefits. For example, the decision to add a waste heat recovery system to a legally required pollution incinerator should be evaluated on its own merits, given the use of the incinerator; the investment evaluation should not lump the costs and benefits of the two systems together.

In summary, each of the investment evaluation methods has its particular advantages and disadvantages, and will be a useful decision criterion in certain cases. For most decision problems, the net present value or the net annual value method, supplemented by benefit/cost ratios or internal rates of return, will provide adequate measures for economically efficient investment decisions.

TABLE 3.9. *Net benefits and benefit/cost ratio rankings for a set of independent projects*

Project	Benefits (B) ($ thousands)	Costs (C) ($ thousands)	Net benefits ($ thousands)	Net benefits Project ranking	B/C Ratio	B/C Project ranking
1	300	200	100	1	1.50	3
2	200	120	80	2	1.67	2
3	140	80	60	3	1.75	1

Source: Harold E. Marshall and Rosalie T. Ruegg, *Efficient Allocation of Research Funds: Economic Evaluation Methods with Applications in Building Technology*, National Bureau of Standards Interagency Report, 1976 (in Press).

TABLE 3.10. *Compound interest factors* [a]

6% Compound interest factors

Single payment				Uniform series			
n	Compound amount factor *F/P*	Present worth factor *P/F*	Sinking fund factor *A/F*	Capital recovery factor *A/P*	Compound amount factor *F/A*	Present worth factor *P/A*	*n*
1	1.0600	0.9434	1.000 00	1.060 00	1.000	0.943	1
2	1.1236	.8900	0.485 44	0.545 44	2.060	1.833	2
3	1.1910	.8396	.314 11	.374 11	3.184	2.673	3
4	1.2625	.7921	.228 59	.228 59	4.375	3.465	4
5	1.3382	.7473	.177 40	.273 40	5.637	4.212	5
6	1.4185	.7050	.143 36	.203 36	6.975	4.917	6
7	1.5036	.6651	.119 14	.179 14	8.394	5.582	7
8	1.5938	.6274	.101 04	.161 04	9.897	6.210	8
9	1.6895	.5919	.087 02	.147 02	11.491	6.802	9
10	1.7908	.5584	.075 87	.135 87	13.181	7.360	10
11	1.8983	.5268	.066 79	.126 79	14.972	7.887	11
12	2.0122	.4970	.059 28	.119 28	16.870	8.384	12
13	2.1329	.4688	.052 96	.112 96	18.882	8.853	13
14	2.2609	.4423	.047 58	.107 58	21.015	9.295	14
15	2.3966	.4173	.042 96	.102 96	23.276	9.712	15
16	2.5404	.3936	.038 95	.098 95	25.673	10.106	16
17	2.6928	.3714	.035 44	.095 44	28.213	10.477	17
18	2.8543	.3503	.032 36	.092 36	30.906	10.828	18
19	3.0256	.3305	.029 62	.089 62	33.760	11.158	19
20	3.2071	.3118	.027 18	.087 18	36.786	11.470	20
21	3.3996	.2942	.025 00	.085 00	39.993	11.764	21
22	3.6035	.2775	.023 05	.083 05	43.392	12.042	22
23	3.8197	.2618	.021 28	.081 28	46.996	12.303	23
24	4.0489	.2470	.019 68	.079 68	50.816	12.550	24
25	4.2919	.2330	.018 23	.078 23	54.865	12.783	25
26	4.5494	.2198	.016 90	.076 90	59.156	13.003	26
27	4.8223	.2074	.015 70	.075 70	63.706	13.211	27
28	5.1117	.1956	.014 59	.074 59	68.528	13.406	28
29	5.4184	.1846	.013 58	.073 58	73.640	13.591	29
30	5.7435	.1741	.012 65	.072 65	79.058	13.765	30
31	6.0881	.1643	.011 79	.071 79	84.802	13.929	31
32	6.4534	.1550	.011 00	.071 00	90.890	14.084	32
33	6.8406	.1462	.010 27	.070 27	97.343	14.230	33
34	7.2510	.1379	.009 60	.069 60	104.184	14.368	34
35	7.6861	.1301	.008 97	.068 97	111.435	14.498	35
40	10.2857	.0972	.006 46	.066 46	154.762	15.046	40
45	13.7646	.0727	.004 70	.064 70	212.744	15.456	45
50	18.4202	.0543	.003 44	.063 44	290.336	15.762	50
55	24.6503	.0406	.002 54	.062 54	394.172	15.991	55
60	32.9877	.0303	.001 88	.061 88	533.128	16.161	60
65	44.1450	.0227	.001 39	.061 39	719.083	16.289	65
70	59.0759	.0169	.001 03	.061 03	967.932	16.385	70
75	79.0569	.0126	.000 77	.060 77	1 300.949	16.456	75
80	105.7960	.0095	.000 57	.060 57	1 746.600	16.509	80
85	141.5789	.0071	.000 43	.060 43	2 342.982	16.549	85
90	189.4645	.0053	.000 32	.060 32	3 141.075	16.579	90
95	253.5463	.0039	.000 24	.060 24	4 209.104	16.601	95
100	339.3021	.0029	.000 18	.060 18	5 638.368	16.618	100

[a] The letter notations in the column headings, e.g., *F/P*, *P/F*, etc., indicate the operation to be performed with the factors in that column. For example, *F/P* indicates that the single compound amount factors are used to find the future value of a given present amount.

TABLE 3.11

8% Compound interest factors

	Single payment			Uniform series			
n	Compound amount factor *F/P*	Present worth factor *P/F*	Sinking fund factor *A/F*	Capital recovery factor *A/P*	Compound amount factor *F/A*	Present worth factor *P/A*	*n*
1	1.0800	0.9259	1.000 00	1.080 00	1.000	0.926	1
2	1.664	.8573	0.480 77	0.560 77	2.080	1.783	2
3	1.2597	.7938	.308 03	.388 03	3.246	2.577	3
4	1.3605	.7350	.221 92	.301 92	4.506	3.312	4
5	1.4693	.6806	.170 46	.250 46	5.867	3.993	5
6	1.5869	.6302	.136 32	.216 32	7.336	4.623	6
7	1.7138	.5835	.112 07	.192 07	8.923	5.206	7
8	1.8509	.5403	.094 01	.174 01	10.637	5.747	8
9	1.9990	.5002	.080 08	.160 08	12.488	6.247	9
10	2.1589	.4632	.069 03	.149 03	14.487	6.710	10
11	2.3316	.4289	.060 08	.140 08	16.645	7.139	11
12	2.5182	.3971	.052 70	.132 70	18.977	7.536	12
13	2.7196	.3677	.046 52	.126 52	21.495	7.904	13
14	2.9372	.3405	.041 30	.121 30	24.215	8.244	14
15	3.1722	.3152	.036 83	.116 83	27.152	8.559	15
16	3.4259	.2919	.032 98	.112 98	30.324	8.851	16
17	3.7000	.2703	.029 63	.109 63	33.750	9.122	17
18	3.9960	.2502	.026 70	.106 70	37.450	9.372	18
19	4.3157	.2317	.024 13	.104 13	41.446	9.604	19
20	4.6610	.2145	.021 85	.101 85	45.762	9.818	20
21	5.0338	.1987	.019 83	.099 83	50.423	10.017	21
22	5.4365	.1839	.018 03	.098 03	55.457	10.201	22
23	5.8715	.1703	.016 42	.096 42	60.893	10.371	23
24	6.3412	.1577	.014 98	.094 98	66.765	10.529	24
25	6.8485	.1460	.013 68	.093 68	73.106	10.675	25
26	7.3964	.1352	.012 51	.092 51	79.954	10.810	26
27	7.9881	.1252	.011 45	.091 45	87.351	10.935	27
28	8.6271	.1159	.010 49	.090 49	95.339	11.051	28
29	9.3173	.1073	.009 62	.089 62	103.966	11.158	29
30	10.0627	.0994	.008 83	.088 83	113.283	11.258	30
31	10.8677	.0920	.008 11	.088 11	123.346	11.350	31
32	11.7371	.0852	.007 45	.087 45	134.214	11.435	32
33	12.6760	.0789	.006 85	.086 85	145.951	11.514	33
34	13.6901	.0730	.006 30	.086 30	158.627	11.587	34
35	14.7853	.0676	.005 80	.085 80	172.317	11.655	35
40	21.7245	.0460	.003 86	.083 86	259.057	11.925	40
45	31.9204	.0313	.002 59	.082 59	386.506	12.108	45
50	46.9016	.0213	.001 74	.081 74	573.770	12.233	50
55	68.9139	.0145	.001 18	.081 18	848.923	12.319	55
60	101.2571	.0099	.000 80	.080 80	1253.213	12.377	60
65	148.7798	.0067	.000 54	.080 54	1847.248	12.416	65
70	218.6064	.0046	.000 37	.080 37	2720.080	12.443	70
75	321.2045	.0031	.000 25	.080 25	4002.557	12.461	75
80	471.9548	.0021	.000 17	.080 17	5886.935	12.474	80
85	693.4565	.0014	.000 12	.080 12	8655.706	12.482	85
90	1018.9151	.0010	.000 08	.080 08	12 723.939	12.488	90
95	1497.1205	.0007	.000 05	.080 05	18 701.507	12.492	95
100	2199.7613	.0005	.000 04	.080 04	27484.516	12.494	100

Table 3.12

10% Compound interest factors

	Single payment		Uniform series				
n	Compound amount factor *F/P*	Present worth factor *P/F*	Sinking fund factor *A/F*	Capital recovery factor *A/P*	Compound amount factor *F/A*	Present worth factor *P/A*	
1	1.1000	0.9091	1.000 00	1.100 00	1.000	0.909	1
2	1.2100	.8264	0.476 19	0.576 19	2.100	1.736	2
3	1.3310	.7513	.302 11	.402 11	3.310	2.487	3
4	1.4641	.6830	.215 47	.315 47	4.641	3.170	4
5	1.6105	.6209	.163 80	.263 80	6.105	3.791	5
6	1.7716	.5645	.129 61	.229 61	7.716	4.355	6
7	1.9487	.5132	.105 41	.205 41	9.487	4.868	7
8	2.1436	.4665	.087 44	.187 44	11.436	5.335	8
9	2.3579	.4241	.073 64	.173 64	13.579	5.759	9
10	2.5937	.3855	.062 75	.162 75	15.937	6.144	10
11	2.8531	.3505	.053 96	.153 96	18.531	6.495	11
12	3.1384	.3186	.046 76	.146 76	21.384	6.814	12
13	3.4523	.2897	.040 78	.140 78	24.523	7.103	13
14	3.7975	.2633	.035 75	.135 75	27.975	7.367	14
15	4.1772	.2394	.031 47	.131 47	31.772	7.606	15
16	4.5950	.2176	.027 82	.127 82	35.950	7.824	16
17	5.0545	.1978	.024 66	.124 66	40.545	8.022	17
18	5.5599	.1799	.021 93	.121 93	45.599	8.201	18
19	6.1159	.1635	.019 55	.119 55	51.159	8.365	19
20	6.7275	.1486	.017 46	.117 46	57.275	8.514	20
21	7.4002	.1351	.015 62	.115 62	64.002	8.649	21
22	8.1403	.1228	.014 01	.114 01	71.403	8.772	22
23	8.9543	.1117	.012 57	.112 57	79.543	8.883	23
24	9.8497	.1015	.011 30	.111 30	88.497	8.985	24
25	10.8347	.0923	.010 17	.110 17	98.347	9.077	25
26	11.9182	.0839	.009 16	.109 16	109.182	9.161	26
27	13.1100	.0763	.008 26	.108 26	121.100	9.237	27
28	14.4210	.0693	.007 45	.107 45	134.210	9.307	28
29	15.8631	.0630	.006 73	.106 73	148.631	9.370	29
30	17.4494	.0573	.006 08	.106 08	164.494	9.427	30
31	19.1943	.0521	.005 50	.105 50	181.943	9.479	31
32	21.1138	.0474	.004 97	.104 97	201.138	9.526	32
33	23.2252	.0431	.004 50	.104 50	222.252	9.569	33
34	25.5477	.0391	.004 07	.104 07	245.477	9.609	34
35	28.1024	.0356	.033 69	.103 69	271.024	9.644	35
40	45.2593	.0221	.002 26	.102 26	442.593	9.779	40
45	72.8905	.0137	.001 39	.101 39	718.905	9.863	45
50	117.3909	.0085	.000 86	.100 86	1 163.909	9.915	50
55	189.0591	.0053	.000 53	.100 53	1 880.591	9.947	55
60	304.4816	.0033	.000 33	.100 33	3 034.816	9.967	60
65	490.3707	.0020	.000 20	.100 20	4 893,707	9.980	65
70	789.7470	.0013	.000 13	.100 13	7 887.470	9.987	70
75	1 271.8952	.0008	.000 08	.100 08	12 708.954	9.992	75
80	2 048.4002	.0005	.000 05	.100 05	20 474.002	9.995	80
85	3 298.9690	.0003	.000 03	.100 03	32 979.690	9.997	85
90	5 313.0226	.0002	.000 02	.100 02	53 120.226	9.998	90
95	8 556.6760	.0001	.000 01	.100 01	85 556.760	9.999	95
100	13780.6123	.0001	.000 01	.100 01	137 796.123	9.999	100

Table 3.13

12% Compound interest factors

	Single payment		Uniform series				
n	Compound amount factor F/P	Present worth factor P/F	Sinking fund factor A/F	Capital recovery factor A/P	Compound amount factor F/A	Present worth factor P/A	n
1	1.1200	0.8929	1.000 00	1.120 00	1.000	0.893	1
2	1.2544	.7972	0.471 70	0.591 70	2.120	1.690	2
3	1.4049	.7118	.296 35	.416 35	3.374	2.402	3
4	1.5735	.6355	.209 23	.329 23	4.779	3.037	4
5	1.7623	.5674	.157 41	.277 41	6.353	3.605	5
6	1.9738	.5066	.123 23	.243 23	8.115	4.111	6
7	2.2107	.4523	.099 12	.219 12	10.089	4.564	7
8	2.4760	.4039	.081 30	.201 30	12.300	4.968	8
9	2.7731	.3606	.067 68	.187 68	14.776	5.328	9
10	3.1058	.3220	.056 98	.176 98	17.549	5.650	10
11	3.4785	.2875	.048 42	.168 42	20.655	5.938	11
12	3.8960	.2567	.041 44	.161 44	24.133	6.194	12
13	4.3635	.2292	.035 68	.155 68	28.029	6.424	13
14	4.8871	.2046	.030 87	.150 87	32.393	6.628	14
15	5.4736	.1827	.026 82	.146 82	37.280	6.811	15
16	6.1304	.1631	.023 39	.143 39	42.753	6.974	16
17	6.8660	.1456	.020 46	.140 46	48.884	7.120	17
18	7.6900	.1300	.017 94	.137 94	55.750	7.250	18
19	8.6128	.1161	.015 76	.135 76	63.440	7.366	19
20	9.6463	.1037	.013 88	.133 88	72.052	7.469	20
21	10.8038	.0926	.012 24	.132 24	81.699	7.562	21
22	12.1003	.0826	.010 81	.130 81	92.503	7.645	22
23	13.5523	.0738	.009 56	.129 56	104.603	7.718	23
24	15.1786	.0659	.008 46	.128 46	118.155	7.784	24
25	17.0001	.0588	.007 50	.127 50	133.334	7.843	25
26	19.0401	.0525	.006 65	.126 65	150.334	7.896	26
27	21.3249	.0469	.005 90	.125 90	169.374	7.943	27
28	23.8839	.0419	.005 24	.125 24	190.699	7.984	28
29	26.7499	.0374	.004 66	.124 66	214.583	8.022	29
30	29.9599	.0334	.004 14	.124 14	241.333	8.055	30
31	33.5551	.0298	.003 69	.123 69	271.292	8.085	31
32	37.5817	.0266	.003 28	.123 28	304.847	8.112	32
33	42.0915	.0238	.002 92	.122 92	342.429	8.135	33
34	47.1425	.0212	.002 60	.122 60	384.520	8.157	34
35	52.7996	.0189	.002 32	.122 32	431.663	8.176	35
40	93.0510	.0107	.001 30	.121 30	767.091	8.244	40
45	163.9876	.0061	.000 74	.120 74	1 358.230	8.283	45
50	289.0022	.0035	.000 42	.120 42	2 400.018	8.305	50
∞				.120 00		8.333	∞

TABLE 3.14

15% Compound interest factors

Single payment			Uniform series				
n	Compound amount factor *F/P*	Present worth factor *P/F*	Sinking fund factor *A/F*	Capital recovery factor *A/P*	Compound amount factor *F/A*	Present worth factor *P/A*	*n*
1	1.1500	0.8696	1.000 00	1.150 00	1.000	0.870	1
2	1.3225	.7561	.465 12	.615 12	2.150	1.626	2
3	1.5209	.6575	.287 98	.437 98	3.472	2.283	3
4	1.7490	.5718	.200 26	.350 27	4.993	2.855	4
5	2.0114	.4972	.148 32	.298 32	6.742	3.352	5
6	2.3131	.4323	.114 24	.264 24	8.754	3.784	6
7	2.6600	.3759	.090 36	.240 36	11.067	4.160	7
8	3.0590	.3269	.072 85	.222 85	13.727	4.487	8
9	3.5179	.2843	.059 57	.290 57	16.786	4.772	9
10	4.0456	.2472	.049 25	.199 25	20.304	5.019	10
11	4.6524	.2149	.041 07	.191 07	24.349	5.234	11
12	5.3503	.1869	.034 48	.184 48	29.002	5.421	12
13	6.1528	.1625	.029 11	.179 11	34.352	5.583	13
14	7.0757	.1413	.024 69	.174 69	40.505	5.724	14
15	8.1371	.1229	.021 02	.171 02	47.580	5.847	15
16	9.3576	.1069	.017 95	.167 95	55.717	5.954	16
17	10.7613	.0929	.015 37	.165 37	65.075	6.047	17
18	12.3755	.0808	.013 19	.163 19	75.836	6.128	18
19	14.2318	.0703	.011 34	.161 34	88.212	6.198	19
20	16.3665	.0611	.009 76	.159 76	102.444	6.259	20
21	18.8215	.0531	.008 42	.158 42	118.810	6.312	21
22	21.6447	.0462	.007 27	.157 27	137.632	6.359	22
23	24.8915	.0402	.006 28	.156 28	159.276	6.399	23
24	28.6252	.0349	.005 43	.155 43	184.168	6.434	24
25	32.9190	.0304	.004 70	.154 70	212.793	6.464	25
26	37.8568	.0264	.004 07	.154 07	245.712	6.491	26
27	45.5353	.0230	.003 53	.153 53	283.569	6.514	27
28	50.0656	.0200	.003 06	.153 06	327.104	6.534	28
29	57.5755	.0174	.002 65	.152 65	377.170	6.551	29
30	66.2118	.0151	.002 30	.152 30	434.745	6.566	30
31	76.1435	.0131	.002 00	.152 00	500.957	6.579	31
32	87.5651	.0114	.001 73	.151 73	577.100	6.591	32
33	100.6998	.0099	.001 50	.151 50	664.666	6.600	33
34	115.8048	.0086	.001 31	.151 31	765.365	6.609	34
35	133.1755	.0075	.001 13	.151 13	881.170	6.617	35
40	267.8635	.0037	.00 56	.150 56	1 779.090	6.642	40
45	538.7693	.0019	.000 28	.150 28	3 585.128	6.654	45
50	1 083.6574	.0009	.000 14	.150 14	7 217.716	6.661	50
∞				.150 00		6.667	∞

TABLE 3.15

20% Compound interest factors

Single payment			Uniform series				
n	Compound amount factor *F/P*	Present worth factor *P/F*	Sinking fund factor *A/F*	Capital recovery factor *A/P*	Compound amount factor *F/A*	Present worth factor *P/A*	*n*
1	1.2000	0.8333	1.000 00	1.200 00	1.000	0.833	1
2	1.4400	.6944	0.454 55	0.654 55	2.200	1.528	2
3	1.7280	.5787	.274 73	.474 73	3.640	2.106	3
4	2.0736	.4823	.186 29	.386 29	5.368	2.589	4
5	2.4883	.4019	.134 38	.334 38	7.442	2.991	5
6	2.9860	.3349	.100 71	.300 71	9.930	3.326	6
7	3.5832	.2791	.077 42	.277 42	12.916	3.605	7
8	4.2998	.2326	.060 61	.260 61	16.499	3.837	8
9	5.1598	.1938	.048 08	.248 08	20.799	4.031	9
10	6.1917	.1615	.038 52	.238 52	25.959	4.192	10
11	7.4301	.1346	.031 10	.231 10	32.150	4.327	11
12	8.9161	.1122	.025 26	.225 26	39.581	4.439	12
13	10.6993	.0935	.020 62	.220 62	48.497	4.533	13
14	12.8392	.0779	.016 89	.216 89	59.196	4.611	14
15	15.4070	.0649	.013 88	.213 88	72.035	4.675	15
16	18.4884	.0541	.011 44	.211 44	87.442	4.730	16
17	22.1861	.0451	.009 44	.209 44	105.931	4.775	17
18	26.6233	.0376	.007 81	.207 81	128.117	4.812	18
19	31.9480	.0313	.006 46	.206 46	154.740	4.844	19
20	38.3376	.0261	.005 36	.205 36	186.688	4.870	20
21	46.0051	.0217	.004 44	.204 44	225.026	4.891	21
22	55.2061	.0181	.003 69	.203 69	271.031	4.909	22
23	66.2474	.0151	.003 07	.203 07	326.237	4.925	23
24	79.4968	.0126	.002 55	.202 55	392.484	4.937	24
25	95.3962	.0105	.002 12	.202 12	471.981	4.948	25
26	114.4755	.0087	.001 76	.201 76	567.377	4.956	26
27	137.3706	.0073	.001 47	.201 47	681.853	4.964	27
28	164.8447	.0061	.001 22	.201 22	819.223	4.970	28
29	197.8136	.0051	.001 02	.201 02	984.068	4.975	29
30	237.3763	.0042	.000 85	.200 85	1 181.882	4.979	30
31	284.8516	.0035	.000 70	.200 70	1 419.258	4.982	31
32	341.8219	.0029	.000 59	.200 59	1 704.109	4.985	32
33	410.1863	.0024	.000 49	.200 49	2 045.931	4.988	33
34	492.2235	.0020	.000 41	.200 41	2 456.118	4.990	34
35	590.6682	.0017	.000 34	.200 34	2 948.341	4.992	35
40	1 469.7716	.0007	.000 14	.200 14	7 343.858	4.997	40
45	3 657.2620	.0003	.000 05	.200 05	18 281.310	4.999	45
50	9 100.4382	.0001	.000 02	.200 02	45 497.191	4.999	50
∞				.200 00		5.000	∞

TABLE 3.16

25% Compound interest factors							
	Single payment		Uniform series				
n	Compound amount factor *F/P*	Present worth factor *P/F*	Sinking fund factor *A/F*	Capital recovery factor *A/P*	Compound amount factor *F/A*	Present worth factor *P/A*	*n*
1	1.2500	0.8000	1.000 00	1.250 00	1.000	0.800	1
2	1.5625	.6400	.444 44	.694 44	2.250	1.440	2
3	1.9531	.5120	.262 30	.512 30	3.813	1.952	3
4	2.4414	.4096	.173 44	.423 44	5.766	2.362	4
5	3.0518	.3277	.121 85	.371 85	8.207	2.689	5
6	3.8147	.2621	.088 82	.338 82	11.259	2.951	6
7	4.7684	.2097	.066 34	.316 34	15.073	3.161	7
8	5.9605	.1678	.050 40	.300 40	19.842	3.329	8
9	7.4506	.1342	.038 76	.288 76	25.802	2.463	9
10	9.3132	.1074	.030 07	.280 07	33.253	3.571	10
11	11.6415	.0859	.023 49	.273 49	42.566	3.656	11
12	14.5519	.0687	.018 45	.268 45	54.208	3.725	12
13	18.1899	.0550	.014 54	.264 54	68.760	3.780	13
14	22.7374	.0440	.011 50	.261 50	86.949	3.824	14
15	28.4217	.0352	.009 12	.259 12	109.687	3.859	15
16	35.5271	.0281	.007 24	.257 24	138.109	3.887	16
17	44.4089	.0225	.005 76	.255 76	173,636	3,910	17
18	55.5112	.0180	.004 59	.254 59	218.045	3.928	18
19	69.3889	.0144	.003 66	.253 66	273.556	3.942	19
20	86.7362	.0115	.002 92	.252 92	342.945	3.954	20
21	108.4202	.0092	.002 33	.252 33	429.681	3.963	21
22	135.5253	.0074	.001 86	.251 86	538.101	3.970	22
23	169.4066	.0059	.001 48	.251 48	673.626	3.976	23
24	211.7582	.0047	.001 19	.251 19	843.033	3.981	24
25	264.6978	.0038	.000 95	.250 95	1 054.791	3.985	25
26	330.8722	.0030	.000 76	.250 76	1 319.489	3.988	26
27	413.5903	.0024	.000 61	.250 61	1 650.361	3.990	27
28	516.9879	.0019	.000 48	.250 48	2 063.952	3.992	28
29	646.2349	.0015	.000 39	.250 39	2 580.939	3.994	29
30	807.7936	.0012	.000 31	.250 31	3 227.174	3.995	30
31	1 009.7420	.0010	.000 25	.250 25	4 034.968	3.996	31
32	1 262.1774	.0008	.000 20	.250 20	5 044.710	3.997	32
33	1 577.7218	.0006	.000 16	.250 16	6 306.887	3.997	33
34	1 972.1523	.0005	.000 13	.250 13	7 884.609	3.998	34
35	2 465.1903	.0004	.000 10	.250 10	9 856.761	3.998	35
40	7 523.1638	.0001	.000 03	.250 03	30 088.655	3.999	40
45	22 958.8740	.0001	.000 01	.250 01	91 831.496	4.000	45
50	70 064.9232	.0000	.000 00	.250 00	280 255.693	4.000	50
∞				250 00		4.000	∞

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Chapter 4

CASE STUDIES OF SUCCESSFUL SYSTEMS FOR INDUSTRIAL WASTE HEAT RECOVERY

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This chapter provides summaries of several successful industrial installations of waste heat recovery systems. These case studies are selected to represent

(a) Major commercial options in waste heat recovery equipment. Examples are radiation recuperator, economizer, heat wheel.
(b) Various industrial applications of heat recovery equipment. Examples are glass-making, forge furnace, paint and varnish processing.
(c) Various geographical locations throughout the United States.

The case studies are presented without naming the industrial firm where the system was successfully operated. In each case, a check has been made of the data used or recorded by the firm to establish the benefits resulting from the application of the waste heat recovery system.

Some of the case studies are of such recent history that only a limited amount of comparative before-and-after data can be presented; others are much more complete.

The general outline for each case study is as follows:

Title of Case
Background
Description of pertinent pre-existing equipment
Specifications for design of the waste heat management device(s)
Description of final design of equipment (especially of added device(s) and of major changes in pre-existing equipment)
Cost study of (a) added device(s); (b) changes in existing equipment, if appropriate
Difficulties and/or special factors encountered in developing and/or applying the new design
Comparison of normal-operation results before and after changes (emphasizing savings in energy use and operating expense and any significant changes in product quality)
Review of the application
Suggestions (recommendations) for potential users of the application
Appropriate technical data.

CASE STUDY

4.1 Waste Heat Recovery in a Glass Plant (Metallic Radiation Recuperator)

4.1 Performance of a Stack Recuperator on a Day Tank at a Glass Plant

4.1.1 Background

As a part of ordinary maintenance, a glass company rebuilt one of its day tanks, and decided to save energy by using a stack recuperator to preheat combustion air. An independent firm was consulted (in 1971), which recommended a high nickel steel stack recuperator.

The day tank was oval, approximately 6 ft × 8 ft, with a 12 in high metal lining. The holding capacity was 3.5 ton of glass. It was provided with two flat-flame burners, a 16 oz blower, and a zero regulator air-gas mixer.

4.1.2 Design of the Stack Recuperator

The equipment and tank design were unchanged, but the stack operation was changed from downdraft (fig. 4.1.1) to updraft (fig. 4.1.2). The recuperator was set on a refractory stack base approximately 48 in high, and the new stack was built over the top of the existing stack at floor level, leaving the lower part of the old stack to serve as a cleanout and (if needed) as an entrance for cold air. The unit was designed to withstand 2100-2300 F, using 1/8 in Incoloy 800 for the recuperator and 1/16 in stainless steel for the stack.

Two distinct modes of operation were to be determined, one for melting and one for holding. The heat transfer rate for melting was to be 7.71 Btu/h ft^2 F; combustion air was to be supplied at 800-825 F with a pressure loss of about 4.25 in W.C. (water column) on high fire and 0.9 in W.C. on low fire. Fuel input and flue temperatures were estimated from previous operation.

4.1.4 Description of the Stack Recuperator

The center pipe was approximately 7 ft long and 24 inches in diameter, with vertical strips 3 in long and 3/8 in wide welded on the outside. A larger concentric pipe of the same length fitted over the center pipe, leaving an air space of approximately 3/8 in between the pipes. The outer pipe was covered with insulation and the insulation was then covered with a metal shield. Six inch diameter inlet and outlet pipes were fitted to the outer pipe.

4.1.5 Cost Study of the Stack Recuperator

$3175.00	for the recuperator
445.00	for a 13 ft × 12 in diameter stack
220.00	for piping and flanges
680.00	for labor
$4520.00	total cost

The estimated payback period based on normal operation, as calculated in the following section, was approximately 1.5 yr.

4.1.6 Difficulties Encountered in Changeover to Recuperator Use

The use of heated combustion air resulted in a need for flexible piping between burners to allow for thermal expansion, a 2 in cold air line from the blower to the burner housings to cool the burners, changes in air/gas ratio control to allow for variable expansion of air due to variable recuperator heating, a stack damper to protect the recuperator during failures of power for the blower and to allow bleed-in of cool air if the combustion air becomes too hot, and control of the back pressure in the tank.

4.1.7 Comparison of Normal Operation Results Before and After Recuperation

The following were the normal operating parameters for the glass tank in melting and holding operations.

ON MELT	BEFORE	AFTER
Av melt time, h	14.5	15.0
Av melts/week	2.5	2.5
Total h melting/week	36.25	37.5
Av gas use, ft^3/h	3700	2250
Total gas use ft^3/week	134,125	84,375
Pressure drop across blower, oz/in^2	13	11
HOLDING		
Total h holding/week	131.75	130.5
Av gas use, ft^3/h	1500	1250
Total gas use, ft^3/week	197,625	163,125

MELTING AND HOLDING		
Total gas use, ft^3/week	331,750	247,500
Total gas use, ft^3/yr	16,587,500	12,375,000 (25.4% savings)
GLASS QUALITY	less than 1% of poor quality	less than 1% of poor quality

From the preceding tabulation the annual (50 weeks) gas savings for melting and holding = (331,750−247,500) = 4,212,500 ft^3/yr. At a theoretical cost of \$0.70/Mft^3 (1971 prices), the savings (for fuel only) is (4212.5) (\$0.70) − \$2948.75/yr. The simple payback period, based on fuel savings only, is

$$\frac{\$4520 \text{ total cost}}{\$2948.75/\text{yr, savings}} = 1.53 \text{ yr.}$$

4.1.8 Review of the Recuperator Application

Circumstances were ideal for the installation of the recuperator as the day tank was to be rebuilt anyway. By using the existing combustion equipment and the glass company's own people for installation, the expenditures were kept to a minimum. In addition to the recuperator itself, including insulation and base, the only new equipment needed was some extra piping and flanges and the 13 ft × 12 in diameter stack. Although installation of a recuperator generally increases the required blower pressure, in this case the lowering of fuel and air consumption more than compensated for this effect and lowered the blower pressure.

Relatively minor difficulties were encountered in the changeover (as already summarized). Two melts of glass were ruined during adjustment of the air/gas ratio. There were no visible signs of recuperator deterioration at the end of the first year's operation.

4.1.9 Suggestions for Potential Recuperator Customers

Knowledge of the following operating conditions of the furnace are of prime importance:

(1) Type of fuel and quantity used,
(2) Amount of excess air,
(3) Flue gas temperature leaving the furnace,
(4) If the present blower cannot deliver sufficient pressure to overcome the additional pressure drop of the recuperator a new blower will be required.
(5) All other equipment such as burners, air lines, control valves, etc., should be checked for suitability for hot air operation.

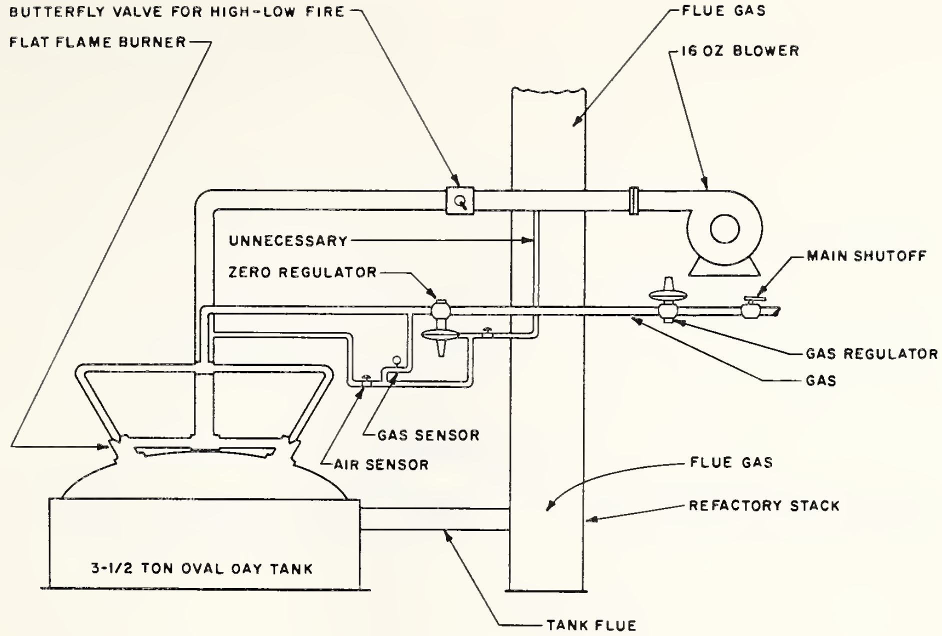

FIGURE 4.1.1 *Day tank and combustion air system before installation.*

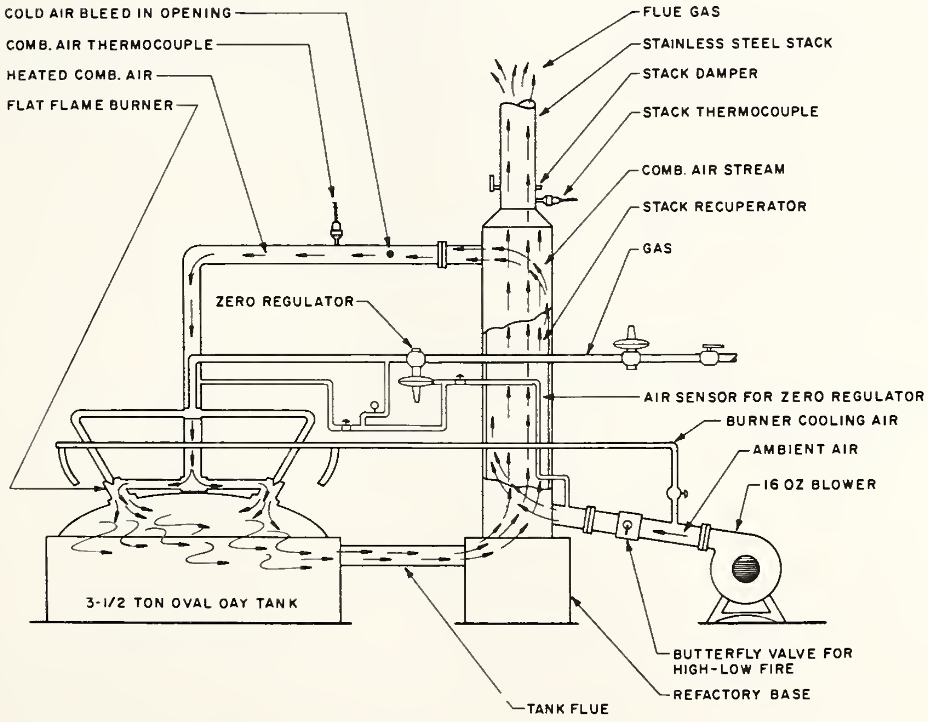

FIGURE 4.1.2 *Day tank and combustion air system after installation.*

CASE STUDY

4.2 Waste Heat Recovery in a Rotary Forge Furnace (Recuperator)

4.2 Recuperator Test Project—Rotary Forge Furnace with Metallic Radiation Recuperator

4.2.1 Background

This case study illustrates waste heat recovery on the scale of a small furnace. A recuperator test program was sponsored by the research department of a natural gas company, working in conjunction with the rotary forge furnace manufacturer, the small industrial firm using the furnace, and the supplier of the recuperator. A second natural gas company coordinated all activities related to the project, thus involving a total of five organizations in the test project. The objective was to preheat combustion air for the rotary forge furnace by removing the waste heat from the flue gas using a metallic radiation recuperator.

4.2.2 Description of the Equipment

The radiation recuperator was of standard size and construction as pictured schematically in figure 4.2.1. The unit size chosen had an advertised rating of 1.6 × 10^6 to 2.6 × 10^6 Btu/h. Other ratings included a maximum waste gas temperature of 2700 F, maximum air preheat to 950 F, and a pressure drop on the air side of approximately 5 inch water column. Additional factors such as ease of installation and maintenance, high corrosion resistance, and short delivery time were also persuasive in selecting this unit.

At the time of the test (1968), the project required about $6400 in actual cash outlay—$2400 for the recuperator and $4000 for installation. It was recognized at the outset that the recuperator's inner shell, costing $1690, had a lifetime of 5-7 yr. The necessary replacement of this shell could accompany a complete furnace rebuild, which traditionally occurs within the same time period. In this way, the additional maintenance cost for the recuperator was minimized.

4.2.3 Difficulties Encountered

Installation of the recuperator coincided with the rebuilding of an old furnace. It was correctly anticipated that some period of adjustment should be expected. In fact, a week of production was lost because unforeseen difficulties prevented complete pre-engineering of the installation. For instance, fixed orifices and unknown flow characteristics of existing bonnets and manifolds created a perplexing situation in predicting total effects of preheating the air.

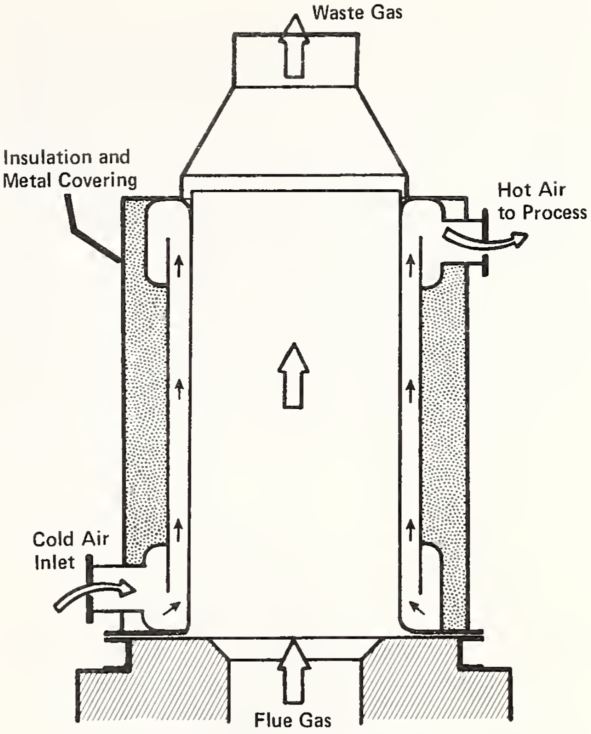

FIGURE 4.2.1 *Radiation recuperator.*

Some typical problems which upset the furnace operation during the testing period reemphasized the importance of good maintenance *with* or *without* recuperation. These included leaking zero governors, a sticking butterfly valve, a slipping linkage, and excess combustion air levels. For example, at one point during the tests, the furnace actually used more fuel with the recuperator than without, because of improper damper controls.

Operation of the rotary forge furnace in question was contingent on work scheduling and physical condition of aging equipment. Because of the job-shop nature of the plant, the product mix was diverse, causing frequent die changes. Thus, intermittent operation led to a pattern of recuperator use different from what may be expected under conditions of continuous operation. In particular, the recuperator must be protected from potential damage due to excess residual heat during idle periods. In this project some cooling air was introduced into the recuperator during such periods.

4.2.4 Before and After Comparison

Preheating combustion air generally results in lower fuel consumption for two reasons: first, less fuel is needed to bring the air up to the flame temperature; second, the air formerly required to burn that increment of fuel need not be provided (or heated) at all. This multiplying effect can increase fuel savings far beyond the simple proportional gain associated with new vs. old incoming air temperatures. Theoretical estimates of fuel savings as a function of air preheat and furnace temperature are shown in figure 4.2.2.

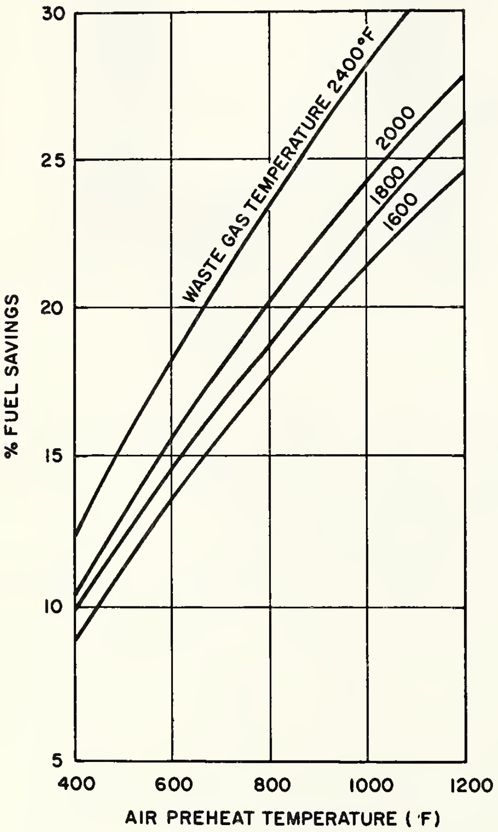

Figure 4.2.2 *Fuel savings using a waste heat recuperator.*

There are further savings due to the more intense heat and faster response of the furnace with recuperation. These factors permitted:

(a) reducing the usual 2400 F operating temperature to 1400 F during work lulls;

(b) weekend idling (previously at 2200 F) at 1200 to 1400 F, thus saving 30,000 ft^3 of gas per weekend;

(c) reduction of furnace firing time for production use after a shutdown from several hours to less than an hour.

Product quality was more stable with recuperation. Scaling is a function of the furnace aeration temperature and soak time. With recuperation, the more stable temperature and reduced excess air diminishes scaling problems.

Table 4.2.1. *Summary of comparative data before and after recuperator installation*

Before recuperation (typical)

Product[a]	Size of stock	Test hours recorded	Total MCF[b]	Average MCF/h[b]
A	$1\frac{1}{4}'' \times 28\frac{1}{2}''$	$14\frac{1}{2}$	44.1	3.0
B	$1'' \times 30''$	4	9.4	2.4
C	$1\frac{1}{4}'' \times 18''$	4	9.6	2.4

After recuperation (typical)

Product[a]	Size of stock	Test hours recorded	Total MCF[b]	Average MCF/h[b]
D	$1\frac{1}{4}'' \times 28\frac{1}{2}''$	4	7.8	2.0
E	$1\frac{1}{8}'' \times 31''$	7	10.6	1.5
F	$1\frac{5}{16}'' \times 16''$	7	13.8	2.0

[a] The products chosen for before-and-after comparison were selected so that approximately similar size of stock could be compared. Cross-section and length of bar stock directly affected hourly fuel consumption. A compares with D, etc.

[b] Thousands of cubic feet.

To establish representative fuel consumption characteristics of the furnace, a check meter was installed. Some typical comparative data before and after recuperation for a variety of products are included in table 4.2.1. These data are a small portion of the total comparative data compiled. After deleting non-typical work interrupted by unusual lulls and breakdowns, the comparative totals before and after recuperation were:

Recuperation	Test Hours Recorded	Total ft^3	Av ft^3/h
Before (total)	117	284,000	2430
After (total)	123	224,900	1830

Considering all the valid test data, the range of fuel input was between 1600 ft^3/h and 2000 ft^3/h with recuperation as compared to a range between 2100 ft^3/h and 3000 ft^3/h without recuperation. Average savings in fuel were about 20 percent.

4.2.5 Cost Savings Analysis

As mentioned in section 4.2.2, initial installation cost amounted to \$6400 in 1968; to this was to be added \$1690 every 5-7 yr for inner shell replacement. The 1968 price of gas was \$0.64/1000 ft^3. The results of the before-after study were converted to an "annual" basis by anticipating an up-time of 3800 h/yr. The more conservative figure of 2000 ft^3/h with recuperation was used in the following calculations. Fifty weekends were assumed, with 30,000 ft^3 saved each weekend by idling at lower temperatures.

	Before	After	Net Savings
Gas input, 100 ft^3/yr (3800 operating h/yr)	9230	7600	\$1630
Weekend differential, 1000 ft^3/yr			1500
Total			3130
Price (av rate per 1000 ft^3)	\$0.64	\$0.64	
Total fuel savings/yr			\$2000

The simple "payback period" (see sec. 6.2) is immediately calculated to be \$6400/2000 = 3.2 yr. Evidently the \$1690 in replacement costs every 5 or 7 yr is fully repaid in the year when it occurs.

Because this was a test program, profitability was not the decisive issue leading to the installation of this recuperator. It is instructive to consider the merit of this investment on the basis of typical 1968 business forecasting parameters, including 8 percent interest rates. The reader who applies the "Uniform Present Worth" method of section 6.3 will find a payback period of 3.8 yr for the figures presented here. If one retrospectively looks at the inflation that has occurred since 1968, using the methods of section 6.5, the investment looks even worse, given the small rise permitted in the price of natural gas during the same time period.

However, if one considers the decrease in profit due to cutting back production in the face of gas allocations, the payback period shortens considerably: Under allocations, a 20 percent savings in fuel translates into greater than 20 percent more up-time (the furnace would have been idling otherwise), with a corresponding increase in output and hence profit. Then \$6400 is fully repaid in much less than a year for a factory of this size.

4.2.6 Suggestions for Potential Users

In operating a furnace with heat recovery, the following should be considered important criteria:

1. Positive pressure must be maintained within the furnace at all times.
2. Air/fuel ratio with minimum excess air is necessary to achieve highest preheat. If burner air and fuel orifices are fixed (as in this project), the highfire on the working range may need to be changed.
3. Furnace control should remain automatic except for normal turndown during work lulls.
4. In sizing the recuperator to the furnaces, it is necessary to know actual volume of fuel used in the furnace under average and maximum conditions. If the recuperator is too small, early failure of the unit can be expected; conversely, if the unit is sized too large, poor heat transfer will result.

CASE STUDY

4.3 Waste Heat Recovery in a Gas Company Facility (Gas to Gas Regenerator)

4.3 Turbine Exhaust Gas Waste Heat Recovery

4.3.1 Background

A gas company was adding natural gas dehydrators to their LNG (Liquified Natural Gas) facilities. The regeneration gas had to be heated for the dehydrators. An existing gas turbine in the plant was used to drive the compressor for the refrigerant liquifying the natural gas. The exhaust gas from the gas turbine was discharged into the atmosphere through a silencer at a minimum temperature of 825 F. Since the amount of regeneration gas to be heated was only about 5 percent of the turbine exhaust gas and the temperature required was lower than that of the turbine exhaust gas, it was possible to use the sensible heat in the turbine exhaust gas for heating regeneration gas.

4.3.2 Description of Pertinent Existing Equipment

A gas turbine with a silencer was the compressor driver. The existing duct on the gas turbine exhaust had inside dimensions of 5 ft 0 in (W) × 8 ft 0 in (L). The opening faced upward. The waste heat recovery device for the regeneration gas could be mounted on the top of the ductwork.

4.3.3 Specification for the Waste Heat Regenerator Exchanger

It was required to design, furnish, fabricate and ship to job site, one waste heat recovery unit to be installed in the gas turbine exhaust system duct of the C-101T and recover the waste heat used to heat the regeneration gas for the natural gas dehydrators at the LNG facilities.

The process specifications were as follows:

	Tubeside	Shell Side (Exhaust ductwork)
Flowing medium	Natural Gas	Turbine exhaust gas
Flow rate lb/h (max)	18,850	363,000
Flow rate lb/h (normal)	17,300	334,000
Flow rate lb/h (min)	0	317,400
Operating temp. (max) outlet	650 F	970 F (inlet)
Operating temp. (min) inlet	40 F	825 F (outlet)
Operating pressure	186 lb/in^{2}g	4-in H_2O (max)
Design pressure	250 lb/in^{2}g	10-in H_2O
Design temperature	1100 F	
Allowable pressure drop	6 lb/in^2	2-in H_2O
Molecular weight lb/mol	16.96	28.9

The system is cyclic. There can be no flow of natural gas except with full pressure through the tubes while the turbine exhaust is at maximum flow rate and temperature. Tube metallurgy must withstand these conditions.

The unit was built and stamped to the latest revision of section VIII, Div. 1, of the ASME code for unfired pressure vessels. The unit was designed with inlet connection to match the bolting and the size of the existing duct opening. The exterior of the unit was sandblasted to SSPC-SP6 finish and painted with one coat of primer to withstand 1100 F.

4.3.4 Description of Regenerator Waste Heat Exchanger Design

Serpentine-type, finned tube coils were designed for the application. The turbine exhaust gas flows continuously across the coils and this coil configuration together with fins around the tubes minimizes the thermal shock during the change from zero to full flow inside the tubes.

SA-335-P11 (1-1/4% Cr 1/2% Mo) alloy steel pipes with 11 percent Cr fins were used for the coils. The material will withstand full turbine exhaust gas temperature without any cooling from inside the coils.

The gas inlet of the unit has a short transition section to match the bolting and size of the existing ductwork from the gas turbine. The design information is summarized in the table 4.3.1. The overall dimensions of the unit are shown on figure 4.3.1.

Table 4.3.1

PERFORMANCE AND DATA

PURCHASER:	DATE:	DES.:
FOR:	TYPE:	PROJ.:
PLANT SITE:	DRAWING NO.:	DRAWN:
PURCHASERS REFERENCE:	NUMBER	EST.:

UNIT							
HEATING GAS SIDE	Gas flow	lb/hr	363,000	334,000		317,400	
	Design pressure/temperature	in/F	4/1100				
	Operating pressure at inlet	in	2.0				
	Operating pressure at outlet	in					
	Pressure drop	in	2.0	1.69		1.52	
	Temperature at inlet	F	970	970		825	
	Temperature at outlet	F	891	892		774.5	
	Temperature drop	F	79	78		50.5	
	Specific heat (ave.)	Btu/lb/F	0.266	0.266		0.266	
	Heat given up	MBtu/hr	7.63	6.93		4.26	
	Efficiency	%	96.3	97.3		97.0	
	Fouling factor	hr, sq ft, F/Btu	0.001	0.001		.001	
	Auxiliery fuel (lhv=)	lb/h					
HEATED SIDE	Fluid flow	lb/hr	18,850	17,300		10,290	
	Design pressure/temperature	psig/F	250				
	Operating pressure at inlet	psig	186	186		186	
	Operating pressure at outlet	psig	183				
	Pressure drop	psi	3				
	Temperature at outlet	F	650	650		650	
	Temperature at inlet	F	40	40		40	
	Temperature rise	F	610	610		610	
	Heat absorbed/lb of steam	Btu/lb					
	Heat absorbed	MBtu/h	7.35	6.74		4.13	
	LMTD	F	542	543		390	
	Overall heat tr. coeff.	Btu/hr, sq ft, F	6.52	6.00		4.97	
	Fouling factor	hr, sq ft, F/Btu	0.001	0.001		.001	
	Steam quality	% or PPM					
	Blow-down water	%					
MECHANICAL DATA	Heating surface	sq ft	2132				
	Tube diameter	in	1.90 O.D. Pipe				
	min. thickness	in	0.127				
	length		10'-0"				
	material		SA-335-P11 (same as SA-213-T11)				
	No. of rows transverse to flow		12				
	Transverse spacing (S_T)	in	4 7/8" STAGG.				
	No. of rows longitudinal to flow		4				
	Longitudinal spacing (S_L)	in	4½"				
	Fin no/in		6				
	height	in	0.5				
	thickness	in	0.05				
	material		11% Cr				
	Drum (header) diameter	in	8				
	length	in					
	material	in	SA-106B				
	Inlet nozzle diameter	in	6				
	Outlet nozzle diameter	in	6				
			Corten (see drawing)				

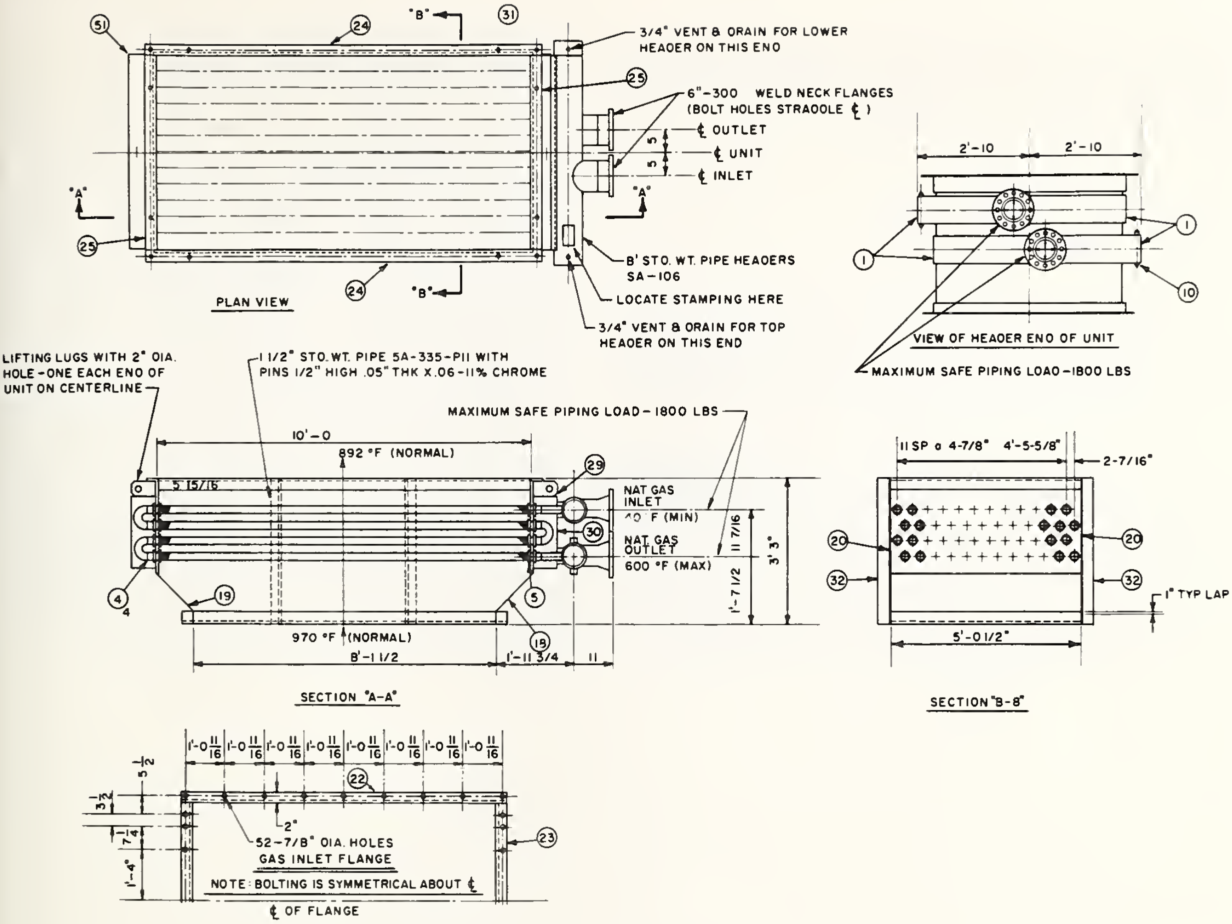

Figure 4.3.1 *Drawing of a turbine waste heat regenerator.*

4.3.5 Cost Evaluation (Based on the Cost Figures in 1972)

Regenerator exchanger:	$16,000
Piping to and from Natural Gas Dehydrators:	10,000
Field installation:	15,000
Total Cost	$41,000

Under normal conditions, 17,300 lb/h of natural gas was to be heated from 90 F to 650 F. This required 6.74 × 10^6 Btu/h. Based on natural gas price of $0.75/MBtu and 300 h/yr of operating time, the annual saving using the waste heat was: 6.75 × 10^6 Btu/h × $0.75/$10^6$ Btu × 3000 h/yr – $15,187.50/yr. An approximate payback period, then, is

$$\frac{41,000}{15,000} = 2.75 \text{ yr}$$

4.3.6 Special Factors Encountered in Developing the Design

The cyclic natural heating of the regeneration gas required particular attention to the design of the metallurgy of the coils. The flowing medium inside the coils was natural gas, which must not leak and mix with the turbine exhaust gas. Nondestructive test methods such as radiography and ultrasonic testing for all the welded joints add extra assurance to the safe operation of the unit.

The existing turbine had a silencer which reduced the exhaust gas pressure before the gas was discharged into the atmosphere. The available draft loss for the regenerator exchanger was therefore limited. The unit had to be designed with the exhaust gas pressure drop to within the specified limits so as not to reduce the efficiency of the gas turbine. The unit was designed to be mounted on the top of the existing ductwork, reducing required modification.

4.3.7 Review of the Application

The amount of natural gas to be heated was small compared to the quantity of the exhaust gas. Only a small portion (12%) of the sensible heat in the exhaust gas was utilized.

The silencer on the existing gas turbine limited the available draft loss for the regenerator exchanger. This, in turn, affected the size of the unit. If more draft loss were available, the unit could have been designed more economically. Additional heat recovery devices also could have been installed behind the regenerator exchanger.

4.3.8 Recommendations for Potential Heat Recovery Users

For a gas turbine of the size described in this example, approximately 59.5 × 10^6 Btu/h of sensible heat is available in the exhaust gas. This heat could have been used to produce approximately 43,000 lb/h of medium pressure steam superheated to 600-700 F. The steam could have been used in a steam turbine to drive an electric generator or compressor. The combined cycle could have improved the thermal efficiency of the system from about 33 percent to the 70-80 percent range. The heat in the exhaust gas also could have been used to preheat the combustion air for the gas turbine. Less fuel would be needed for the turbine to produce the same horsepower. Another possibility was for the waste heat recovery device to be designed to muffle the exhaust gas noise as well as recovering heat. The silencer for the gas turbine could thus be eliminated.

CASE STUDY

4.4 Waste Heat Recovery in a Can Company (Gas to Gas Regenerator-Incinerator)

4.4 Heat Recovery by Gas to Gas Regeneration in a Hydrocarbon Fume Incinerator

4.4.1 Background

A potential air pollution problem was introduced when another production can line was to be added at a large can company's plant. Hydrocarbons were emitted from solvents utilized in the bake process following lithographic printing of coated steel sheets for caps and closures. At times the solvent loadings encountered in the effluent were quite high; at other times they were considerably lower. Therefore, an outside contractor was retained to engineer, fabricate, deliver and install an incineration system which could include heat recovery for economy while being sensitive to the variable nature of the effluent.

4.4.2 Description of Previously Existing Equipment

The new can line was the third one installed at the plant site. The two previously existing lines had fume incineration; but those incinerators did not include provision for any type of heat recovery from the products of incineration. The process can be schematically represented by going directly from the forced draft fan of figure 4.4.1 to the combustor without preheat, followed by direct discharge of combustion exhaust as clean, uncooled gas to the stack.

4.4.3 Specifications for Design of the Waste Heat Recovery System

The primary specifications for the waste heat and incineration system included:

1. complete destruction of hydrocarbons in the gases from the bake ovens;
2. direct-flame incineration;
3. operational economy.

Specifically, the first of these objectives was recognized to be the most restrictive requirement, because the high solvent loadings encountered at times in the effluent had a correspondingly high heat of combustion. At times negligible heat release results from the actual incineration of the fumes, but at other times more heat may be released than can be handled without special provisions in the system.

4.4.4 Description of the Fume Incineration System

A schematic of the fume incineration system is shown as figure 4.4.1. Oven exhaust fumes were directed from the process gas forced draft fan through the inside of the tubes of a recuperative type heat exchanger where they were preheated before entering the combustor. This heat exchanger was two-pass cross-counterflow system with gases from the bake ovens entering at up to 325 F and leaving at up to 927 F to enter the combustor.

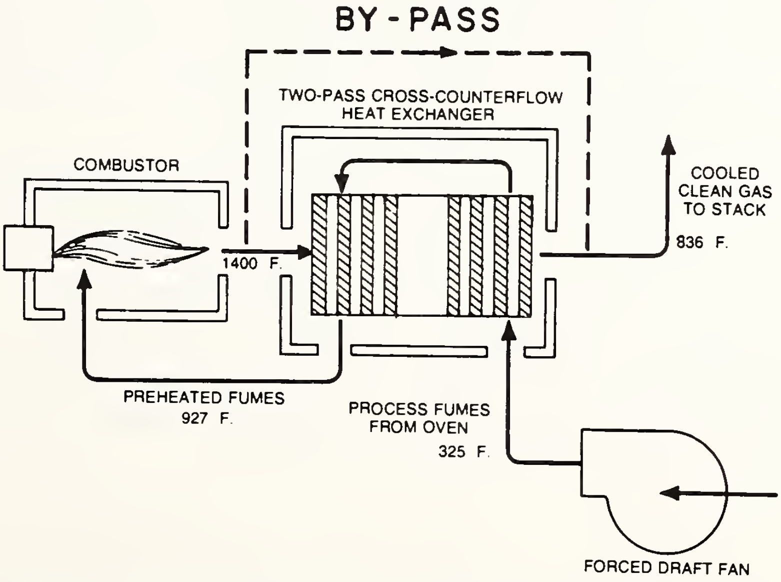

FIGURE 4.4.1 *Schematic diagram of fume incineration system used in lithographic printing of coated steel sheets. Heat exchanger is used to recover heat and reduce fuel cost in incineration.*

In the combustion chamber, the fumes were exposed under proper conditions to a direct flame which oxidized the contaminants to carbon dioxide and water vapor.

The incineration chamber was designed so that the incinerated gases remain in the chamber for a minimum residence time of 0.3 s to insure the complete destruction of the hydrocarbons in the gases.

The now-incinerated hot gases (about 1400 F) left the direct-flame combustion chamber and passed over the outside of the heat exchanger tubes, going to the stack as cleaned, cooled gases at about 836 F. Of course, this cooling was accomplished by heat energy transfer from the clean gases to the contaminated incoming oven exhaust fumes.

The system was made up of modular components for a wide range of performance capability and arrangement flexibility. The heat exchanger had a hot gas bypass control (discussed more later), an incineration chamber complete with burner, safeguard controls and instrumentation, a forced-draft process gas fan, platform and support steel for rooftop mounting, ductwork, dampers, expansion joints and a stainless steel stack with rain cap protection.

4.4.5 Cost Study of the Recovery System

The 1970 total cost of the incineration-heat recovery system, including installation and research work, was $100,000. (The fuel cost per year for a process flow rate of 8500 SCFM with heat recovery was estimated at $17,000 against $37,000 without recovery (54% savings), when operating at an incineration temperature of 1400 F.) This was based on 6000 h/yr of operation and a fuel cost of $0.50/MBtu (1969-1970 cost base). The calorific value of the effluent was not considered because of the widely varying solvent loadings.

A simple payback period calculation can now be made. Based on the savings in fuel cost per year of $20,000 and total first cost of $100,000, one readily finds the payback period to be 5 yr. More sophisticated economic analysis would, of course, alter this figure somewhat; but, this somewhat longer than desired payback period is acceptable because of the pollution-control aspect of the project.

4.4.6 Difficulties Encountered in Design and Special Features in the System

To accommodate the very high solvent loadings encountered in the effluent, a hot gas side heat exchanger bypass was provided (fig. 4.4.1). This bypass automatically opens when solvent heat release by combustion is unusually high, thereby lowering the temperature of the "preheated" fumes entering the combustor. Thus, this damper functioned to maintain a minimum burner fuel requirement, restrict the maximum operating temperature, and achieve good incineration efficiency.

A raw gas burner was provided because of its operational economy. This type of burner requires no primary combustion air, but utilizes available oxygen from the air in the effluent stream to support combustion.

4.4.7 Comparison of Normal-Operation Results Before and After the Recuperation-Incineration Installation

A major benefit from the installation of the recuperation-incineration system as compared to operation of the bake ovens without either incineration or recuperation was clean exhaust gases. While incineration without recuperation (as for the two previously existing can lines) also would have led to reduction of a potential odor and pollution problem, it would have cost more in fuel consumption.

One significant later addition to the system was the installation of baffle plates in a dead corner of the burner section to better direct the air flow. Under consideration is the addition of a catalytic cone-type converter between the burner and heat exchanger. It is anticipated that this would cut the entrance temperature to the heat exchanger from the present 1400 F to about 700 F, which could result in 40-45 percent additional reduction in fuel consumption.

CASE STUDY

4.5 Waste Heat Recovery in a Rubber Plant (Heat Pipes in Series with Heat Exchangers)

4.5 Heat Recovery Using a Heat-Pipe Regenerator to Preheat Make-Up Air in a Rubber Vulcanizing Operation

4.5.1 Background

One rubber company produces the majority of linesmen's protective gloves and arm covers used by the electric industry in this country. The gloves and covers are manufactured in a variety of specialized sizes and types to provide protection from 10,000 to 40,000 V, depending on glove thickness. This rubber company normally produces around 200,000 pairs of gloves per year. Acceptable gloves retail for between $15 and $22 and rejects for $2, so there is great incentive to produce a quality product.

Although the operation normally runs 24 h a day, 7 days a week, constantly increasing demand for gloves pushed manufacturing capability to the limit during 1972. Additional boiler capacity was needed to expand vulcanization and other steam process facilities, but when the company's management consulted with the local gas company concerning increased energy requirements, they discovered the gas company was unable to supply any additional natural gas to either existing or new industrial customers. Since oil and propane were becoming increasingly unavailable in the area, the rubber company was faced with the possibility that their expansion plans would have to be cancelled due to the unavailability of energy. During discussions with the rubber company's management, the gas company's industrial sales personnel pointed out the possibility that heat recovery equipment could save enough energy to handle the proposed expansion.

4.5.2 Description of Previously Existing Equipment

Two 30-yr-old 30-hp boilers of a vertical type were in operation when the proposed expansion was being considered. An antiquated and inefficient steam heated preheat coil was also in operation. This preheat coil was used to temper fresh outside air when the outside air entered its intake duct at less than 50 F. Following preheating, a reheat coil then supplied necessary heat to take the supply air to 94 F, 35 percent relative humidity, which was the condition needed in the dipping room. The air condition had to be maintained very carefully, and it was changed completely every 2-1/2 min in order to control naphtha fumes.

One condition that made the selected recovery unit particularly suitable for the application was the proximity of the intake and exhaust ducting for the makeup and exhaust air at the glove dipping operation. The intake and exhaust ducting were side by side, and this not only simplified construction, but also held down expenses.

4.5.3 Specifications for Design of the Waste Heat Recovery Unit

The heat recovery installation was designed for 12,000 ft^3/min of exhaust air at 94 F, 35 percent rh, and 15,000 ft^3/min of supply airflow at no lower than −5 F. The original preheat was to 50 F and the reheat to 94 F.

4.5.4 Description of Final Design of Replacement Equipment, Including the Heat Recovery Unit

The rubber company's energy problem was resolved as follows:

(1) The thermal recovery unit, as installed to replace the old steam heated preheat coil, recovered 3.2 billion Btu annually from exhaust air. This recovery was based on a typical local heating season of 6196 degree days. The equivalent of 3.2 Mft^3 of gas per year was thus freed for use in the new boiler.

(2) A new 125 hp boiler was installed to provide higher pressure and additional steam to meet the additional processing requirements.

(3) The two 30-yr-old 30 hp vertical upright boilers were retired and the natural gas previously used by this equipment was applied to the new boiler.

The new thermal recovery unit was a standard packaged unit best described as two heat exchangers connected with a series of heat pipes. Each heat pipe was a self-contained closed system capable of transporting large quantities of heat from a source to a sink with very little temperature drop. Its actual operation was dependent on two well-known physical principles—vapor heat transfer and capillary action, as explained in chapter 5. The choices of container (usually a tube, as in the present case), wick material and capillary fluid depend on the operation and design criteria of the heat pipe application. Thermal energy transfer was very efficient since it was essentially an isothermal process—that is, there was no significant temperature difference within the heat pipe from one end to the other. An assembly of heat pipes combined with a source of heat and an area to dissipate the heat, provided a counterflow heat exchanger. (See fig. 5.10 of chap. 5.)

The materials used in this thermal recovery unit consisted of 5/8 in o.d. round seamless aluminum tubing for the pipes and aluminum for the fins. Coil casing material was 16 gage galvanized steel positioned to separate the supply and exhaust air streams and sealed to prevent cross contamination. End covers were of 20 gage galvanized steel. On lower temperature models, copper tubing is available and, where corrosive atmospheres are encountered, protective coatings are also available. The manufacturer of this thermal recovery unit normally uses one of the following class one refrigerants as the working fluid: R 12, R 113, or R 114.

Using the specified exhaust and supply air volume flow rates, and the approximate 550 SFPM face velocities, the required thermal recovery unit was sized as:

$$\text{Area} = \frac{\text{Av ft}^3\text{/min of flow}}{\text{Approx. SFPM of velocity}}$$

$$= \frac{12{,}000 + 15{,}000}{550} = 49.2 \text{ ft}^2$$

The unit selected was a six-row deep 14 fin per inch unit which measured 48 in × 144 in and provided 48 ft^2 of surface. It was installed horizontally at a tilt angle of 10°.

4.5.5 Cost Study of the Thermal Recovery Unit and Installation

Although in the present instance a new boiler replaced two wornout boilers, the economics of that change is not directly related to the cost study of the thermal recovery unit and its installation. At least in theory, the antiquated preheat coil might have been retained for use in conjunction with the new boiler. On the other hand, if the old boilers had been retained, a thermal recovery unit might have replaced the preheat coil without, of course, full satisfaction of the needs of the proposed glove-production expansion. Thus the incremental cost of the thermal recovery unit and installation modifications was more economical than rebuilding and modernizing the original system. The calculation of the payback period involves proprietary information; however, the payback period is between 2.6 and 3.1 yr.

The critical point to be remembered in this and similar applications is that in addition to monetary savings in fuel, the installation made heat recovered available to be used elsewhere.

4.5.6 Difficulties and Special Conditions Encountered

A condition existed here which was fortunate, though not typical—the proximity of the intake and exhaust air ducting. This contributed to the profitability of the venture.

The only real difficulty encountered was caused by condensation that continually occurred on the coil at outside air temperatures up to 49 F. This increased the static pressure drop across the coil. Fortunately, the exhaust and intake fan motors were slightly oversized and could handle the job. However, the drain facilities in the original installation were not adequate to handle as much condensate as the unit produced. Due to the fact that the dipping operation ran 24 h a day, 7 days a week, dipping schedules had to be rearranged so that drain changes could be made.

4.5.7 Comparison of Normal-Operation Results Before and After Heat Recovery Installation

The thermal recovery unit was installed along with a larger, new gas-fired boiler. The unit itself, however, replaced only the old steam-heated preheat coil; therefore, it is strictly the comparison of the old and new preheating operations which is pertinent.

The typical heating season at the rubber plant location is 6196 degree days; that is $X(65 - T_{av}) = 6196$ where X is the number of days of heating, and T_{av} is the average outdoor temperature for the X days of heating, F. The number of days during which the average outside air temperature is less than 50 F will be fewer than the number of days during which it is below 65 F. Further, the preheat coil; therefore, it is strictly the comparison needed at all times when the outside temperature is below 50 F.

Before the thermal recovery unit was installed, the old preheat steam coil had to supply essentially the same amount of preheat to the incoming air for ventilation as was later obtained from waste heat recovery. Analysis of the unit, as installed, has shown that (based on the 6196 degree day heating season) it recovered 3.2 billion Btu/yr from the exhaust stream with an efficiency of 62 percent. Efficiency, or recovery factor as the thermal recovery unit manufacturer refers to it, is a measure of the unit's ability to recover heat and transfer it to the incoming stream. This factor is a function of the fin surface area, the number of rows of heat pipes, the velocity of the two air streams. Because of this, efficiency can vary from installation to installation.

Instantaneous heating rate to preheat incoming air to 50 F is given by

$q = m\ c_p(50 - T)$, Btu/h, where
m is flow rate in lb/h,
c_p is constant pressure specific heat, approximately 0.24 Btu/lb•F, and
T is actual outside air temperature, F, which is less than 50 F.

Although it is not a precise approximation, the assumption that the total heating for the year can be taken as based directly on the degree days can be used to show that the quoted recovery is realistic. Thus,

$$q, \text{Btu/yr} = 15{,}000\ \text{ft}^3/\text{min}\ (0.075\ \text{lb/ft}^3)\ (1440\ \text{min/day})(0.24\ \text{Btu/lb•F})\ (6196\ \text{F day/yr})$$

$$q = 2.4 \times 10^9\ \text{Btu/yr}$$

Dividing this by the unit's recovery factor, one has

$$\frac{2.4(10^9)}{0.62} = 3.8 \times 10^9\ \text{Btu/yr},$$

a figure which compares quite well with the quoted 3.2 billion Btu/yr.

Consider again the operation before the thermal heat recovery unit was installed. The preheat supply actually going into the incoming ventilation air (based on the assumption that it is the same as after the recovery unit was operating and that the quoted figures were correct) is about 3.2 × 0.62 billion Btu/yr; that is, 1.984×10^9 Btu/yr. This was not obtained from waste heat, but from combustion of fuel. Allowing for the inefficiency of the old preheat coil and the old boilers which supplied steam to the coil, the combined efficiency of the coil and boiler operation was, no doubt, roughly the same as that of the thermal unit (about 0.62, say 0.8 for boiler and 0.8 for coil). Thus, the 3.2 billion Btu/yr which was previously paid for to do the preheating was now supplied free by the thermal recovery unit from heat which was previously thrown away as waste heat. At about 1000 Btu/ft^3 of gas, this amounts to a fuel saving of 3200 Mft^3/yr.

4.5.8 Review of the Application

An annual energy savings of 3.2 billion Btu (or 3.2 Mft^3 of natural gas) which permitted a plant expansion in the face of fuel limitations were the results of the installation of a thermal recovery unit at a rubber glove manufacturing plant. Installation was completed and the unit has been in operation since July 1973. So far it has been remarkably trouble free.

It should be emphasized that this case was a low temperature application, the exhaust air from which heat was recovered being at 94 F.

4.5.9 Suggestions for Potential Users of the Application

There are many areas in modern industrial plants where heat recovery units such as the thermal recovery unit used here can be employed to recover energy. In addition to recovering building exhaust, the units can be used in drying operations, air preheat, curing operations and numerous other areas where process heat is necessary. The manufacturer of the subject unit has, subsequently, developed systems for use at up to 800 F, and is presently aiming for even higher temperature levels. For temperatures beyond present capabilities, new fluids for use in the heat pipes must be found. Other high temperature heat recovery devices also exist, and should be considered as alternatives.

CASE STUDY

4.6 Waste Heat Recovery in a Dairy (Space Heating Using an Economizer)

4.6 Heat Recovery System for a Dairy

4.6.1 Background

A dairy was building a new warehouse (1971) and conventional gas heating was not available due to a gas shortage. The alternative solutions considered were: (1) electric heating, and (2) recovery of heat from boiler flue gases which were discharged to the atmosphere.

Electric heating was ruled out because operating costs would have been excessive. Furthermore, the use of heat recovery from the high temperature flue gases was attractive because the boiler plant was located adjacent to the new warehouse.

At first it was thought that a duct system could be fabricated utilizing an exhaust fan to draw the hot gases through the building and then vent them to the atmosphere. This idea was discarded because of the probability that the hot gases would condense part way down the 135 ft length of the building.

About 6,500,000,000 Btu/yr was escaping to the atmosphere during normal boiler operation. The proposed heat recovery system could recover 75 percent (4,875,000,000 Btu/yr) of this loss. The use of the heat recovery system therefore served the following two functions: (1) provide heat for the new warehouse (1,300,000,000 Btu/yr), and (2) provide heat for a boiler feedwater preheater (1,580,000,000 Btu/yr).

This left 1,900,000,000 Btu/yr available for future use, however, that excess heat was available only for a warm weather application as all recoverable heat was required by the system in cold weather.

Since a large portion of the boiler output was consumed as live steam and was not returned in the form of condensate, the problem of preheating the boiler makeup feedwater was serious.

4.6.2 Specifications for the Design of the Heat Recovery System

The general design requirements were:

(1) Installation of a finned hot water coil in the stack breeching.

(2) Installation of a pump to circulate water to the fan coils in the warehouse.

(3) A conventional hot water system to be used in the warehouse.

(4) Water from the finned coil to be automatically diverted to the new boiler feedwater preheater when warehouse heating requirements were satisfied.

(5) Provision to be made to automatically divert flue gases to the atmosphere when more heat was available than could be used.

4.6.3 Description of Waste Heat Recovery System

The general arrangement of the coil and damper system is shown in figure 4.6.1. Figure 4.6.2 is a diagram of the flow system showing various control locations.

Hot water temperature control element T-1 modulated the bypass dampers to maintain the desired temperature in the heating system (200-210 F).

Whenever the outside temperature was above 55 F, the 3-way valve maintained the desired temperature in the warehouse.

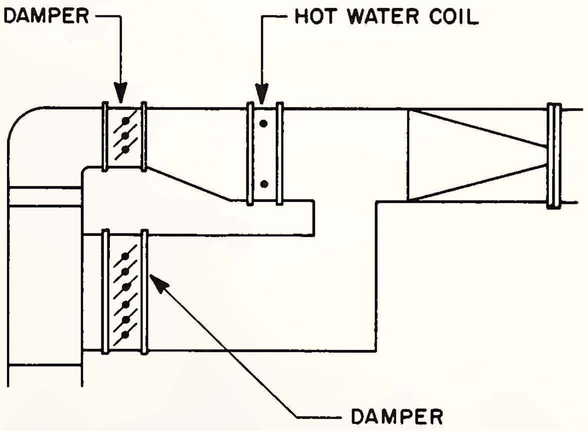

Figure 4.6.1 *Coil and by-pass installation in boiler stack breeching.*

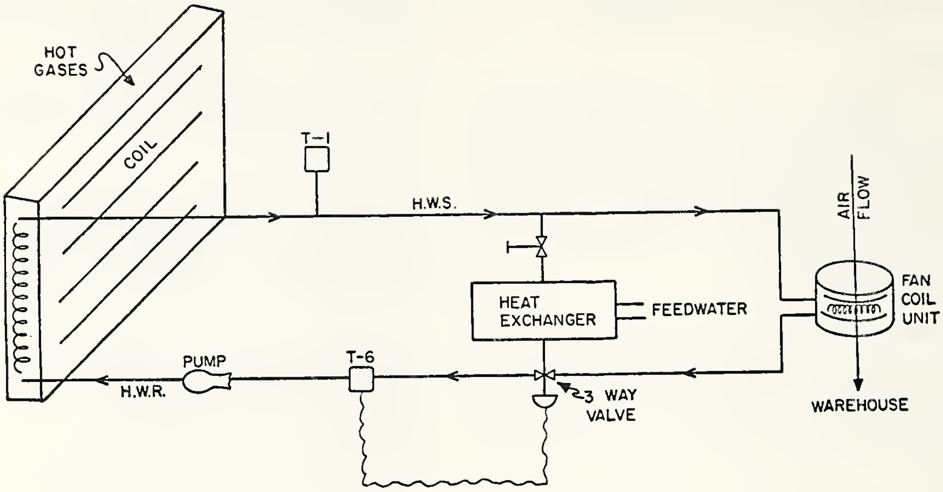

FIGURE 4.6.2 *Control of hot water system.*

4.6.4 Cost Evaluation

A summary of costs for payout purposes was as follows:

Payout

	System Cost
Recovery System*	$17,340
Electric Heat	8,300
Incremental Cost	$ 9,040

*Includes a feedwater preheater

	Annual Operating Cost
Electric	$4,820
Recovery System**	0
Operating Savings**	$4,820

**Does not include pump motor operating cost

$\frac{\$9{,}040}{\$4{,}820} = 1.9$ yr payout

From section 4.6.6 the annual savings in gas if a feedwater preheater was used was 1528 ft^3. The annual savings based on a cost of $0.892/Mft^3 would be $1409.

The basis for elimination of electrical heating as an option for heating the warehouse was as follows:

Cost of electric heating:
Rate 41 General Service – Large
Additional energy rate 0.87/kWh
Additional demand charge $1.40/kWh

Energy cost:

$$\frac{\frac{371{,}000\text{ Btu}}{\text{h}} \times \frac{24\text{ h}}{\text{day}} \times \frac{6000\text{ degree days}}{\text{season}}}{\frac{3414\text{ Btu}}{\text{kWh}} \times 100\%\text{ eff.} \times 50\text{ F}}$$

$$= \frac{312{,}970\text{ kWh}}{\text{season}}$$

$$\frac{312{,}970\text{ kWh}}{\text{season}} \times \$0.0087 = \frac{\$2722}{\text{season}} -$$

(371,000 Btu/h is the warehouse heat loss; see sec. 4.6.6).

Demand charge:

$$\frac{371{,}000\text{ Btu}}{\text{h}} \times \frac{\text{kWh}}{3414\text{ Btu}} = 109\text{ kW Demand}$$

$$109\text{ kW} \times \$1.40 - \$153/\text{mo}$$

Using this value for Dec., Jan., and Feb.: $153 × 3 = $459

Using 1/2 this value for Oct., Nov.. and Mar.: $153 ÷ 2 × 3 = 230

Annual Demand Charge = $689

Annual Energy Cost	$2722
Annual Demand Charge	689
Annual Electric Heat Cost	$3411

4.6.5 Difficulties and Special Conditions Encountered in Adding and Operating the Heat Recovery System

The worst problem encountered was in the damper control equipment. The controls installed were basically specified as residential type. Malfunctions and breakdowns occurred until heavy duty industrial controls were installed.

A second problem was condensation of water vapor on the coil causing "rain" in the boiler stack when the return water to the coil was below 90 F. This usually happened at start-up. It was not a problem once the system reached normal operating conditions.

Another initial control problem occurred when the boiler shifted to high fire, raising the temperature and producing steam in the system, thereby popping the relief valve. This was corrected by delaying the move to high fire until the damper could close ahead of the coil.

4.6.6 Comparison of Normal Operating Results Before and After Adding the New System

The heat available from flue gas was:

$$\frac{50{,}000\ \text{Mft}^3\ \text{gas}}{\text{yr}} \times \frac{10^6\ \text{Btu}}{\text{Mft}^3} \times 13\%\ \text{(heat in flue gases at 550 F and 30\% excess air)}$$

$$= \frac{6{,}500{,}000{,}000\ \text{Btu loss}}{\text{yr}}$$

Recovery of 75 percent of annual heat loss from the boiler provides

$$\frac{6{,}500{,}000{,}000\ \text{Btu}}{\text{yr}} \times 75\%$$

$$= \frac{4{,}875{,}000{,}000\ \text{Btu recoverable}}{\text{yr}}$$

Calculation of the heat loss from the warehouse was as follows:

Heat Loss from Warehouse Q = UAdT

Walls

Lower portion, brick and block,
U = 0.33 Btu/h•ft^2•F, 8 ft high, 371 ft exposed perimeter.
0.33 × 371 × 8 × 50 F = 48,972 Btu/h

Upper portion, Glaros Products Inc.,
U = 0.25 Btu/h•ft^2•F, 12 ft high,
1-1/2 in thick, 371 ft exposed perimeter.
0.25 × 371 × 12 × 50 F = 55,650 Btu/h

Roof

Flat, insulated 1-1/2 in celotex plus tar and gravel, U = 0.22 Btu/h•sq ft•F,
135 ft × 101 ft, 0.22 × 135 × 101 × 50 F
= 143,167 Btu/h

Air Changes

$$\frac{0.24\ \text{Btu}}{\text{lb F}} \times \frac{0.075\ \text{lb}}{\text{ft}^3\ \text{air}} \times \frac{50\ \text{F}}{1} \times 272{,}700\ \text{ft}^3\ \text{volume}$$

$$\times \frac{0.5\ \text{air change}}{\text{h}} = 122{,}715\ \text{Btu/h}$$

Total heat loss = 370,504 Btu/h

For the warehouse a heat loss of 371,000 Btu/h was then used in all calculations.

For the feedwater preheater

Water to boiler:

$$\frac{50{,}000\ \text{Mft}^3\ \text{gas to boiler}}{\text{yr}} \times \frac{75\%\ \text{boiler eff.}}{1}$$

$$\times \frac{10^6\ \text{Btu}}{\text{Mft}^3} \times \frac{1\ \text{lb steam}}{1185\ \text{Btu}} \times \frac{1\ \text{lb water}}{1\ \text{lb steam}}$$

$$= \frac{31{,}650{,}000\ \text{lb water}}{\text{yr}}$$

Annual savings in gas if feedwater preheater were used:

$$\frac{31{,}640{,}000\ \text{lb water}}{\text{yr}} \times 50\ \text{F temp. rise}$$

$$\times \frac{1\ \text{Btu}}{\text{F}\cdot\text{lb H}_2\text{O}} \times \frac{1\ \text{Mft}^3}{10^6\ \text{Btu}} = 1528\ \text{Mft}^3\ \text{of gas}$$

4.6.7 Review of the Application

The restriction by the gas company on any new industrial loads required consideration of either electrical heat or use of the boiler flue gas heat for the warehouse heating system. The primary need was heating the warehouse but additional benefits were obtained by incorporating a feedwater preheater in the system.

The recovery system with the feedwater preheater included was more expensive than an electrical heating system but the incremental cost of $9040 was recovered in 1.9 yr because of the estimated annual saving of $4820. The total cost of $17,340 could be recovered in about 3-1/2 yr.

The system described provided sufficient heat for the warehouse and the feedwater with enough heat left over for other applications. Whether there has been an actual lowering of gas consumption has not been determined although there may have been since the feedwater preheater should reduce boiler fuel consumption.

4.6.8 Suggestions for Potential Heat Recovery Users

Use of a different heat transfer fluid would have been worth consideration. This would have eliminated the need for damper controls as the coil could then "run wild" and remove as much heat as possible without going into the vapor phase.

If the use of a heat transfer fluid other than water is not practical, then the use of pneumatic controls should be considered for faster damper control operations.

Applications of this type of system should be carefully considered. Not all boiler applications lend themselves to heat recovery. Heat recovery is normally feasible only if there is a year around load on the boiler, the flue gas temperatures are high, or live steam is a fairly large percentage of the boiler load.

CASE STUDY

4.7 Waste Heat Recovery in a Glass Plant (Economizer)

4.7 Performance of an Economizer (Feedwater Heater) for a Boiler at a Glass Plant

4.7.1 Background

Anticipating a possible fuel shortage, a glass plant began exploring fuel conservation in April 1972. For efficient use of fuel, effective heat transfer appeared to be an obvious need, so their small packaged boiler was retrofitted with an economizer to heat the boiler feedwater.

4.7.2 Description of the Packaged Boiler

The small packaged boiler was a two-pass gas fired boiler as shown schematically in figure 4.7.1. It had a typical design, operating at 120 lb/in^{2}g saturated (350 F) and producing both heat and process steam. It had 2000 ft^2 of heating surface and an original rating of 200 hp. However, applying the currently used factor of 5 ft^2/hp, its present rating is equivalent to 400 hp.

The total steam produced during the heating season, approximately 6 mo of the year, differed from that for the remainder of the year. On the average, of the total steam produced, 20 percent was used for building heat; the remaining 80 percent was used by process equipment and for heating oil. Heaviest demand was during an 8-h period in the early part of each day—about 3 a.m. to 11 a.m.—with the load tapering off by noon.

During the high demand period, the boiler operated at approximately 18,000 lb/h of steam (or about 90% capacity). After this period, usage was about 13,000 lb/h. On weekends the boiler was on minimum fire.

Approximately 98 percent of all steam generated was recovered as condensate. About 2 percent was lost through boiler blowdown, injection of live steam into process, cleaning of parts, and continuous exhaust from the feedwater tank to get rid of the oxygen expelled from the make-up water.

4.7.3 Specification for Design of the Economizer (Feedwater Heater)

It is difficult to retrofit an existing piece of equipment with new auxiliary equipment, and in this case a custom design was needed for the economizer application. A corporation which supplies such custom-designed economizers, using computer aided analysis to meet specific load parameters, was asked to furnish the needed design.

Information furnished by the glass plant to the economizer manufacturer was based on normal operation at the time, taking into consideration the maximum and minimum limits experienced. Pre-economizer conditions included:

35-40 gpm maximum of feedwater through the boiler
2000-3000 gal (or more) make-up water per day
4160 ft^3/min combustion air blower capacity
4600 ft^3/min induced draft air blower capacity; 3 hp @ 800 rpm
Stack temperature downstream of i.d. fan 400-425 F
Temperature from feedwater tank, 212-218 F

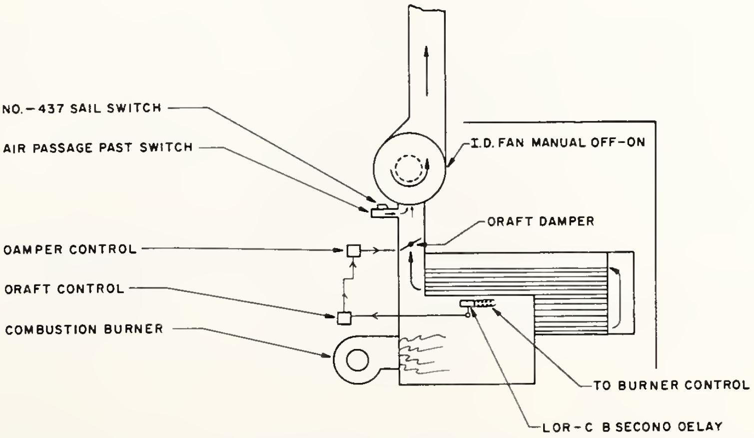

FIGURE 4.7.1 *Two-pass gas fired boiler. Steam produced at 120 lb/in^{2}g (350 °F.).* 2000 ft^2 *heating surface.*

The area available for an economizer was measured, taking into consideration the space needed for new piping.

The following general operating conditions and installation requirements were to be met:

(1) relative simple installation and startup;

(2) normal maintenance to occur once yearly at the time of normal boiler shutdown (this assumed natural gas only as fuel, a condition which has not been met);

(3) installation in the boiler's duct section;

(4) connection to the condenser system and use of the heat recovered (from formerly rejected heat) to warm the feedwater as it passed back to the radiation section, thereby reducing necessary heat from fuel combustion; and

(5) satisfaction of specific load parameters, in particular heating and process steam needs.

4.7.4 Description and Operation of the Economizer

With the specifications that were provided, the economizer manufacturer initially sized the unit to have an expected 640,000 Btu/h heat pickup. After a reasonable time in use it was determined that the pickup averaged more than a million Btu/h.

Figure 4.7.2 shows the relation of the economizer to the boiler. Because of space limitations, the draft damper was located ahead of the economizer as installed. This was not desirable because the air passage to the economizer was always restricted, except on high fire. The damper should be on the downstream side to give 100 percent utilization of the transfer area. The breeching duct was (as in most installations) smaller than the opening to the transfer area, so when the draft damper was partially closed during operation the flue gas would use less than the full transfer area.

It was necessary to increase the horsepower and speed of the induced draft air fan, because the economizer restricted the discharge air breeching. After this change, the boiler had its original capacity, but with less fuel consumption and better efficiency.

Installation and startup of such economizer was relatively simple and required only a few days. If natural gas was burned, normal maintenance was required only once yearly and occurs simultaneously with normal boiler shutdown. To maintain a constant steam supply through the winter months, a four-pass serpentine design was used for the economizer.

Boiler feedwater entered the heat transfer system at slightly different temperature and with considerably different flow rate with the economizer than without.

Economizer-in-use conditions include:

15-20 gpm of feedwater through the boiler
800-1200 gal make-up water per day
4160 ft^3/min combustion air blower capacity
4600 ft^3/min induced air blower capacity, 7-1/2 hp@1250 rpm
Stack temperature downstream of i.d. fan, 300 F
Temperature from feedwater tank, 200 F
Temperature from economizer to boiler, 300 F

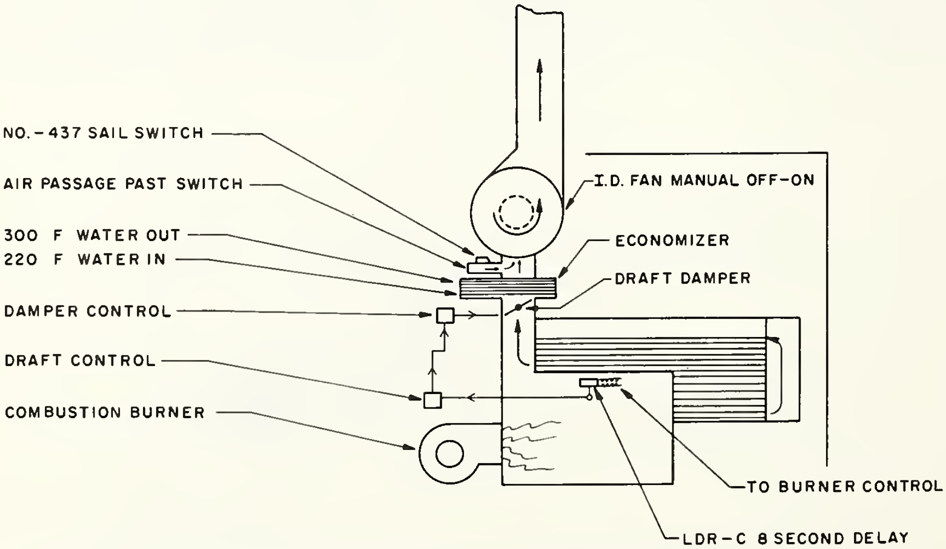

FIGURE 4.7.2 *Two-pass gas fired boiler. Steam delivered at 120 lb/in^{2}g (350 °F); 2000 ft^2 heating surface (with economizer installed).*

Since the water still evaporated to steam at 120 lb/in^{2}g (350 F), the boiler now needed to raise water temperature only 50 degrees to achieve a saturated liquid condition. Without the economizer the boiler had to raise the water temperature 132 degrees before beginning evaporation.

4.7.5 Cost Study of the Economizer

As stated before, the economizer unit was used the year round. A first unit was installed in 1972, and had savings of $5000 for the first year based on 1972 costs. The installation cost figure for this first economizer unit was not available, but after a period of evaluation a second economizer unit was installed (on a second boiler) at a cost of $6046. The two boilers were very seldom used at the same time. One unit would carry the steam demand during normal operation, except in very cold weather with heavy demand from production.

The current yearly savings (1975) were at a rate of $10,000/yr for natural gas and $20,000/yr for No. 5 fuel oil, which has been used all year (or nearly so) because of the gas curtailment in effect. Thus the payback period was well under a year.

The quoted savings per year from the units were based on fuel only. No savings on water (which will be discussed later), labor, other equipment efficiency, etc., were included.

It can readily be demonstrated that the quoted fuel cost savings are correct. A typical average rate of process plus heating demand was 12,500 lb/h of steam. Since this was the feedwater supply rate, assuming no losses at the boiler, and the enthalpy increase of feedwater through the economizer equal to $c_p(T_{out} - T_{in})$, we have Btu/h recovered by economizer = 12,500 lb/h (1.0 Btu/lb-F) (300 − 200) F = 1.25 × 10^6 Btu/h as earlier quoted. If one assumes a 5 day/week operation, 50 week/yr, and 24 h/day, 6 × 10^9 Btu/yr is recovered by the economizer.

Ignoring any weekend savings, and assuming boiler efficiency from 80 to 90 percent, the actual energy saved by the use of the economizer was (at 80% boiler efficiency) between 6.5 × 10^9 and 6.67 × 10^9 in 1 yr of operation.

At a cost of $1.20/MBtu for gas and $2.40/MBtu for oil, the money savings are approximately $9000/yr. Similarly one obtains:

Savings of $8000/yr using gas at 90 percent boiler efficiency

Savings of $18,000/yr using oil at 80 percent boiler efficiency

Savings of $16,000/yr using oil at 70 percent boiler efficiency

Fuel savings were recognized through actual gas meter readings and the position of the burner during a normal operation. Combustion efficiency for oil and gas was checked periodically to insure good results from equipment. The glass plant quoted an efficiency of 79-80 percent, using either No. 5 oil or natural gas.

The only additional equipment need for the conversion besides the economizer itself was the 7-1/2 hp-460 V motor which replaced the 3 hp-460 V motor on the induced draft fan. The difference

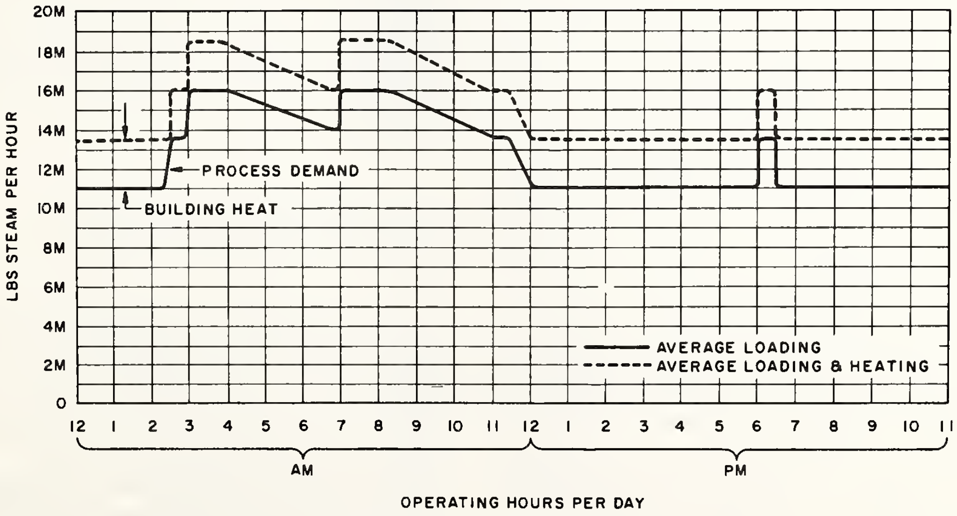

Figure 4.7.3 *Steam demand profile.*

in operational costs of the two was minimal compared to the cost of producing the steam. As far as could be determined, the increase in the induced draft fan horsepower did not add to efficiency.

The manpower operating cost of the boiler operation has not changed. Because of the increased efficiency of the boiler, demands on the operator were reduced, giving him time for other duties.

Comparison of the make-up water rates before and after installation of the economizer also indicated some savings. This was not, included in cost analysis since such items were a bonus, along with the improved quality of the steam now available to the users. The latter has improved the efficiency of process equipment.

4.7.6 Special Conditions Associated with Development and Operation of the Economizer (Feedwater Heater)

There was nothing special about the economizer except that it had been adapted to a small boiler. As in any retrofit, inlet and outlet headers had to be designed specifically for the operation.

Heat transfer techniques for recovering and reusing heat to improve heating efficiency have long been a part of larger production boiler designs. Small boilers have recently become the objects of intensified energy saving studies, because the expense of incorporating the new equipment has offset the savings in use of readily available, inexpensive fuels. With the shortage and increased cost of fuels, the situation has changed. Technology formerly applied only to large boilers is now being used to conserve energy in smaller boilers.

The gain in economy by use of the economizer feedwater heater was by way of reducing prime energy input into the product (steam). In this sense it was much different from the use of a recuperator, which improves combustion efficiency by preheating combustion air.

The present installation required less than 1 week, including installation of auxiliary piping by plant craftsmen. Customized couplings designed to fit existing boiler feedwater lines facilitated installation. The couplings eliminated the need to cut into the heat transfer piping, thereby preserving the integrity of equipment design and related heat transfer and pressure ratings.

4.7.7 Comparison of Normal Operation Before and After Installation

Some secondary savings which are of significance result from the change in handling of the feedwater. Before installation of the economizer, during a heavy demand period, the velocity of steam leaving the boiler actually siphoned water with the steam. Because feedwater was entering the boiler from the feedwater tank at approximately 218 F, the boiler did not have the ability to recover fast enough to prevent the siphoning. The feedwater pump had to work double duty to maintain water level in the boiler, resulting in water usage of 3000-4000 gal/day. The condensate return system could not handle this volume of return so the excess went down the overflow pipe.

After the economizer was installed, feedwater temperature entering the boiler was raised to 300 F, stack temperature decreased from 400 F to 300 F, make-up water reduced to 800-1200 gal/day, no condensate was lost to waste, and natural gas and oil consumption were reduced. Handling less water gave more time of exposure in the heat transfer areas.

The combustion air blower capacity was not changed. Damper control was reset to lower operating pressure in the combustion chamber, and the horsepower of the induced draft air fan was increased from 3 hp to 7-1/2 hp. The speed was increased from 800 rpm to 1200 rpm. Installation of the economizer in the flue gas breeching caused a restriction in flow of air through the boiler, so volume was increased to compensate for the drop. Lack of information about the original air flow rate and flown characteristic curves made it necessary to resolve the air flow problem by trial and knowledge of past experience.

4.7.8 Suggestions for Potential Users of an Economizer for Feedwater Heating

It is not necessary to have an induced draft fan to operate with an economizer, so each system must be studied to determine the need for such an installation. If an induced fan is used and there is sufficient room, the economizer should be installed ahead of the draft damper.

Economizers are essentially maintenance free if good water treatment is used and natural gas is the burner fuel. If heavy oil is used for a fuel, it is necessary to shut down occasionally to blow any accumulated particulate matter off the fins in the unit, since such an accumulation will cause a drop in draft pressure and in efficiency. The conditions can be determined by a qualified operator.

The economizer used here was designed to use either gas or oil. If oil is used for any length of time, its use should be included in the specification requirements. The spacing of the fins is determined by the primary fuel.

The heat transfer equipment has been easy to maintain. There has been no tube clogging in using normally treated feedwater. Currently, the application presented is operating on No. 5 oil 100 percent of the time, so it is necessary to blow the unit out with compressed air every 3-4 weeks. Since boiler tubes require periodic cleaning no special shutdown is made to blow out the economizer. The operator takes care of this, so no extra cost has been added to the operation.

CASE STUDY

4.8 Waste Heat Recovery in the Citrus Industry (Evaporator)

4.8 Waste Heat Evaporator Used in Citrus Industry

4.8.1 Background

The following case study describes waste heat recovery in Florida citrus processing plants. Florida is unique among citrus growing areas of the world in that more than 90 percent of the 10 million tons annually produced is processed. This requires about 100 citrus processing plants, producing canned juices and concentrates as food products, and dried pulp and peel for animal feed. Annual production of dried citrus animal feed exceeds 700,00 tons.

A simplified flow diagram of a citrus plant is shown in figure 4.8.1. This figure shows that washed fruit is fed to juice extractors from which the juice is sent to concentrators. These concentrators, which operate on plant steam, are called "TASTE" evaporators (Temperature Accelerated Short-Time Evaporator).

Pulp from the juice extractors is pressed to remove liquid and sent to the feed mill. Peel is sent directly to the feed mill. In the feed mill, peel and pulp are mixed with lime, passed through a shredder called a pug mill and into a screw conveyor reaction vessel to hydrolyze the peel.

From the reaction vessel, the shredded reaction mass is dried to about 8 percent moisture in a "fire dryer." The dryer uses hot gases from a furnace to dry the peel by direct contact of the gases with the peel. In some plants the peel is pressed to reduce moisture content prior to entering the dryer. The press liquor is concentrated in a separate evaporator to produce molasses which is either sold or added to the animal feed. This scheme is shown in figure 4.8.1. After cooling, the dried mixture is stored and sold for animal feed.

Prior to 1964, almost all fire dryers were directly vented to the atmosphere. Hot gases leaving the dryer were exhausted via large stacks. In 1964, primarily because of air pollution complaints, attempts were made to control the particulate emissions from fire dryers. One method investigated was to use the hot gas leaving the dryer to operate an evaporator in the plant. During this operation, particulate matter in the gas was to have been reduced by condensation of water vapor as the gas temperature decreased. An ideal location for use of this "waste heat" was in the molasses evaporator which was concentrating the peel press liquor. Thus, elimination of a pollution problem and energy conservation could be accomplished at the same time. Figure 4.8.2 shows such a plant.

In 1964, the first "waste heat" molasses evaporator was installed in a Florida citrus plant. One problem encountered was fouling of the hot gas side of the evaporator by condensed particulate matter. This was eliminated by

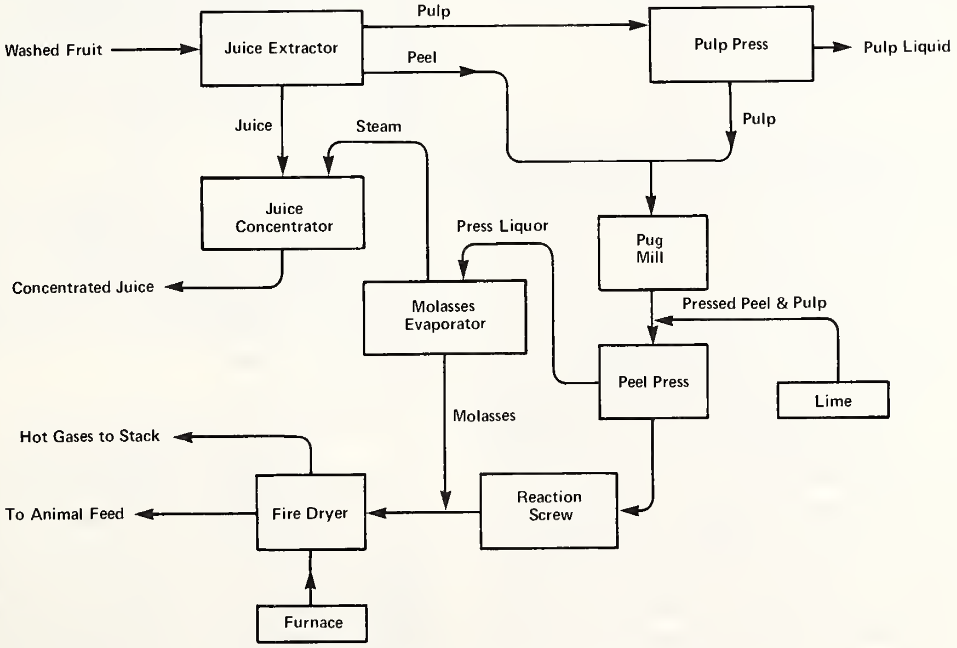

FIGURE 4.8.1 *Schematic diagram of a citrus processing plant.*

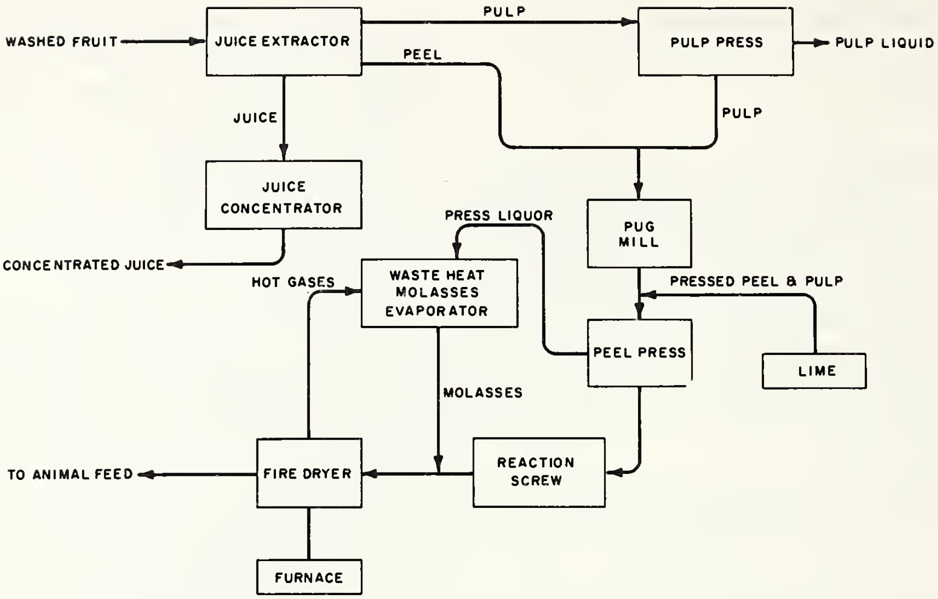

FIGURE 4.8.2 *Schematic diagram of a citrus plant with a waste heat evaporator installed.*

installing particulate separators upstream of the waste heat evaporator. Also, the fire dryers had to be modified to recirculate the hot gases by decreasing gas exit temperature and increasing moisture content to improve condensation in the waste heat evaporator. Waste heat evaporators were installed in many Florida citrus plants during the 1964-1972 period.

4.8.2 Description of Existing Equipment

The citrus processing plant selected for this study had a waste heat evaporator installed in 1973. Prior to installation of waste heat evaporation, the plant used a molasses evaporator to concentrate peel press liquor as shown in figure 4.8.1.

The rotary dryer in use was equipped with hot gas recirculation which facilitated installation of the waste heat evaporator. This type of dryer was installed in 1967 because its operating characteristics produced a higher yield of dried peel (approximately one more pound of dried peel per 90 lb of fresh citrus). Figure 4.8.3 shows a sketch of the dryer along with the existing pug mill and peel presses. Also shown in figure 4.8.3 are the waste heat evaporator, storage tanks and cooling tower which were put in as part of the waste heat evaporator installation.

4.8.3 Specifications for Design of the Waste Heat Evaporator

Figure 4.8.4 shows the proposed material balance and process flows agreed on by the evaporator contractor and the citrus plant management. The waste heat evaporator was to be designed to evaporate 120,000 lb/h of water even though only 60,800 lb of press liquor would be available. The remainder of the evaporative load, 40,000 to 60,000 lb, would be supplied from plant effluent or waste water which is concentrated in the evaporator to eliminate waste disposal problems.

Another feature of the specifications was that particulate emissions were to be 25 lb/h or less. State pollution control regulations for this plant required 32 lb/h or less. Fuel consumption for 30,000 lb/h of dried peel was projected at 93,000 ft^3/h of natural gas.

4.8.4 Final Design of Equipment

The major change in design was in elimination of one of two 5000 gal caustic tanks initially proposed. (Caustic was used for evaporator cleaning.) The caustic tank was replaced with a 20,000 gal tank for holding condensate from the evaporator, as shown in figure 4.8.3. Condensate was withdrawn from this tank for makeup when

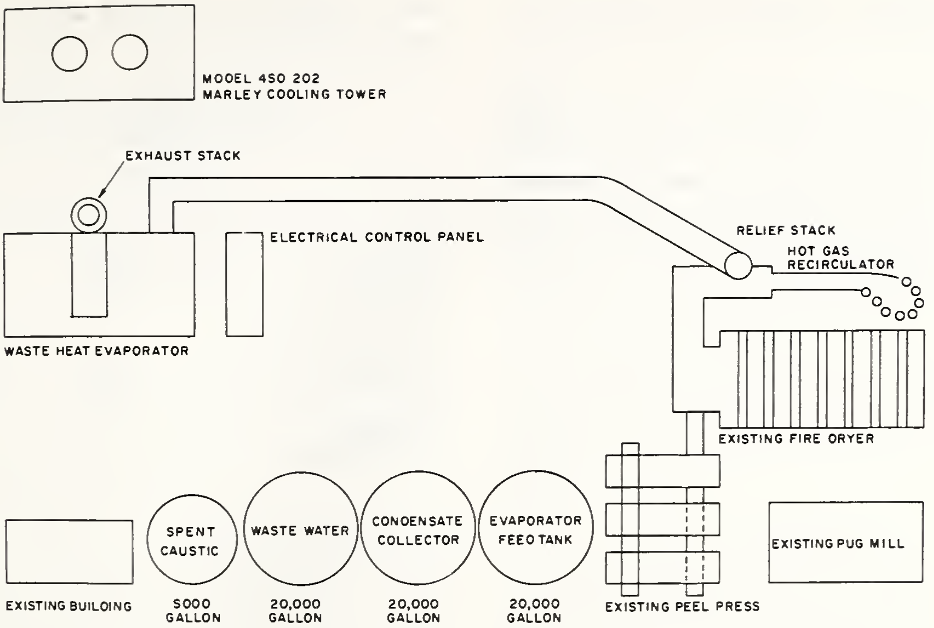

FIGURE 4.8.3 *Diagram of fire dryer and pug mill with tank system, cooling tower, and waste heat evaporator installed.*

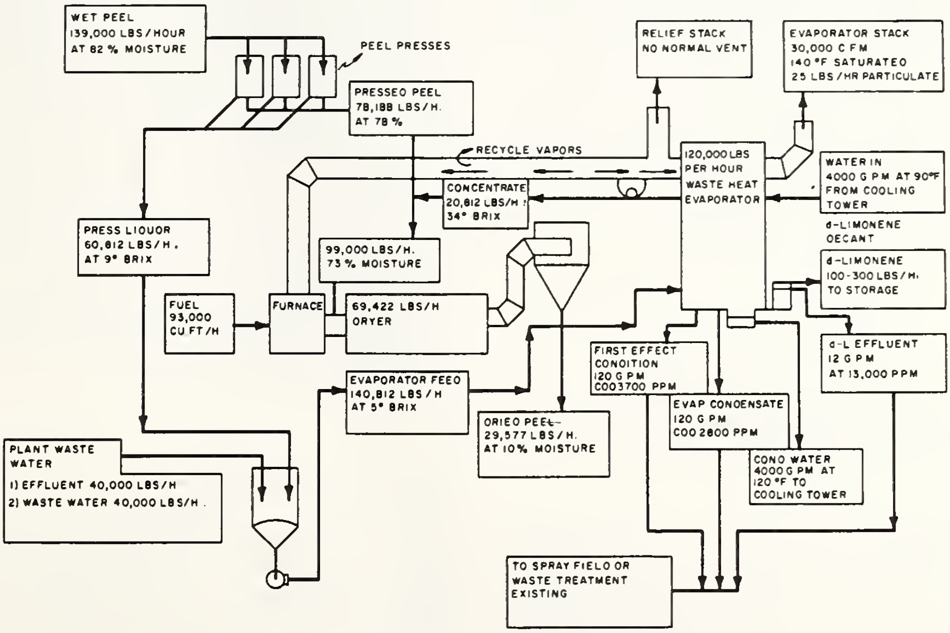

FIGURE 4.8.4 *Materials balance for waste heat evaporator*

needed to maintain a relatively constant load on the evaporator.

In this operation, 60,000 lb/h of press liquor along with 40,000 lb of waste water or condensate was fed to the evaporator. Twenty thousand pounds of additional evaporative capacity was left over to accommodate future increases of press liquor or waste water.

4.8.5 Cost Study

Table 4.8.1 is the quotation on the waste heat evaporator supplied to the citrus plant in November 1972. The principal items were the waste heat evaporator—$525,000, and the cooling tower—$44,000. The third 20,000 gal tank for condensate storage was not included in this quotation. The quotation total was $640,125. An additional $45,000 was subsequently added to the quotation for electrical panels, switches, conduit, and other electrical work. In addition, an additional $100,000 was spent on electrical equipment such as transformers, installation labor, plumbing and piping. Thus, the total capital cost was about $780,000 in late 1972 and 1973 dollars. This total cost was within budget for the project.

Estimated payback based on normal operation, 1975 fuel costs and no discounting of cash flow (see sec. 4.8.7 for details), was 5.2 yr, since fuel savings at that time were $150,000/yr.

4.8.6 Difficulties Encountered in Construction and Changeover to Waste Heat Evaporator

An unexpectedly large amount of labor was required to clean the static screen filters (downstream of peel press). One rotary filter was installed and has partially alleviated this problem. Also the 100 hp blower used to circulate hot air through the waste water heat evaporator caused problems due to the high noise level.

4.8.7 Comparison of Results with Conventional and Waste Heat Evaporation

TABLE 4.8.1 Comparison of Results with Conventional and Waste Heat Evaporator

	Conventional Evaporation	Waste Heat Evaporator
Plant Steam Use, lb 150 lb/in^2 equivalent/h	25,000	—
Electricity Use	255 hp	750 hp
Particulate Emissions, lb/h	30	2

Calculation of dollar savings was based on 1975 fuel costs to produce steam (typical plant mix of Bunker C oil and natural gas was used). Cost of extra electricity for operation of waste heat evaporator was substracted from fuel savings. Total yearly dollar savings calculation (based on operation 6 mo/yr) was $150,000/yr.

Natural gas flow rate to the fire dryer was measured at 91,000 ft^3/h for 30,000 lb of dry peel, very close to the contractor's projection of a 93,000 ft^3/h (see fig. 4.8.4). Measurements were not conducted continuously, however, since the plant had only one gas meter. Thus, gas flow to the dryer could only be measured when the remainder of the plant was using oil for fuel.

The color of the molasses product from the waste heat evaporator was slightly darker than with the original evaporator. Also, the moisture content of molasses leaving the waste heat evaporator was higher than from the original evaporator. Thus, the product from the waste heat evaporator cannot be easily sold in the merchant molasses market. However, sales for human consumption showed no financial advantages over sales for animal feed. Accordingly, these were not considered significant problems.

4.8.8 Review of Application

The waste heat evaporator was installed and brought on stream within budget although delays were encountered, due primarily to delivery problems.

Energy cost savings through use of the waste heat evaporator (at 1975 energy prices) were about $150,000/yr. Additional savings in waste water concentration were realized in addition to the savings cited above. Based solely on energy savings, and 1975 energy costs, a payback time of 5.2 yr is estimated. Naturally, payback time would be longer if cash flow were discounted.

Excellent control of particulate emissions was obtained. Two lb/h of particulates was measured in 1975 versus the 25 lb/h minimum guaranteed by the contractor. This large reduction in emissions is partly due to operating the evaporator at below design capacity.

Installation of waste heat molasses evaporators in the citrus industry is becoming standard practice. The installation reported on in this case study was improved by the contractor's experience obtained at other citrus plants.

A significant advantage obtained with this installation was that the fire dryer had a recirculating hot air system. This system was an integral part of the dryer design.

Installation of waste heat evaporators in citrus plants having burners without hot gas recirculation was also briefly studied. In general, it costs about $40,000 to convert a dryer to hot gas recirculation. Problems in adjusting air flows and dusting were encountered but eventually solved.

The current (1974) cost of a 60,000 lb/h fire dryer is approximately $410,000. This dryer comes equipped with connections for installation of a waste heat evaporator. A quotation is attached as exhibit 4.8.2.

CASE STUDY

4.9 Waste Heat Recovery in an Asphalt Roofing Plant (Boiler)

4.9 Waste Heat Recovery in an Asphalt Roofing Plant

4.9.1 Background

A plant which produced asphalt for impregnated felt roofing, was required, for pollution control purposes, to dispose properly of fumes from the asphalt-flowing still and the felt saturator. Two methods were considered. One was the use of a scrubber; the other was the use of a fume incinerator to eliminate the odor. Since incineration is a high temperature oxidation process and the heating value of the fumes was low and variable, some fuel was needed to maintain the required temperature in the incinerator.

The plant also used process steam, supplied by four boilers which were about 20 to 40 yr old. The plant manager realized that the energy in the fumes together with that in the natural gas used for the incinerator could be recovered by the addition of a heat recovery boiler at the end of the incinerator, thus reducing the amount of fuel used for process steam generation.

4.9.2 Specifications for Design of the Waste Heat Recovery Device

One waste heat boiler was to be designed to take heat from the flue gas from an incinerator burning fumes. The unit was to be a skid-mounted, shop-assembled, watertube boiler to continuously generate 15,000 lb/h of steam at 150 lb/in^{2}g pressure.

The following was the process information for the boiler:

(1) Waste Gases	
Volume	16,000 SCFM
Inlet temperature	1400 F
Outlet temperature	600 F approx.
Pressure drop	6 in of water max
Fouling factor	0.001 h•ft^2•F/Btu
(2) Boiler Feedwater and Steam	
Feedwater pressure	175 lb/in^{2}g
Feedwater temperature	200 F
Steam quantity (saturated)	15,000 lb/h
Steam side fouling factor	0.001 h•ft^2•F/Btu

The unit was to be strictly designed and stamped to section I of the ASME Boiler and Pressure Vessel Code. The boiler was to be of watertube, natural circulation design. Extended surfaces, such as finned tubes, were acceptable. Carbon steel tubes (SA-178A) and shell plates (SA-515-70) were to be used. The following accessories were to be included:

(a) Standard boiler trim, such as safety valves, water column with gage glasses and try cocks, steam pressure gage, feed and check valves, continuous blowdown valves and intermittent blow-off valves.

(b) A single-element feedwater regulator with by-pass valve.

(c) A steam stop-check valve.

The unit should have horizontal gas inlet and vertical gas outlet.

The manufacturer was to provide, with his proposal, information on the heat surface and overall heat transfer coefficient, the calculated gas pressure drop, the gas outlet temperature, and the steam purity in ppm of solid carryover in the steam.

4.9.3 Description of Waste Heat Boiler Design

One water tube waste heat boiler was designed for this application. The unit generated 15,000 lb/h of saturated steam at 150 lb/in^{2}g. The thermal and mechanical data are shown in table 4.9.1. The general outline of the unit is shown on figure 4.9.1.

The heating surface of the boiler consisted of bare tubes and finned tubes. The first two rows of the first pass were bare tubes and were radiant shield tubes for the boiler, since the incinerator was to be hooked up to the boiler and the tubes were exposed to the flames. The remaining tubes were finned.

The unit was completely shop-assembled. It was designed, built and stamped according to section I of the ASME Boiler and Pressure Vessel Code. The gas flow arrangement had a horizontal inlet and vertical outlet. The short stack was mounted on the top of the boiler.

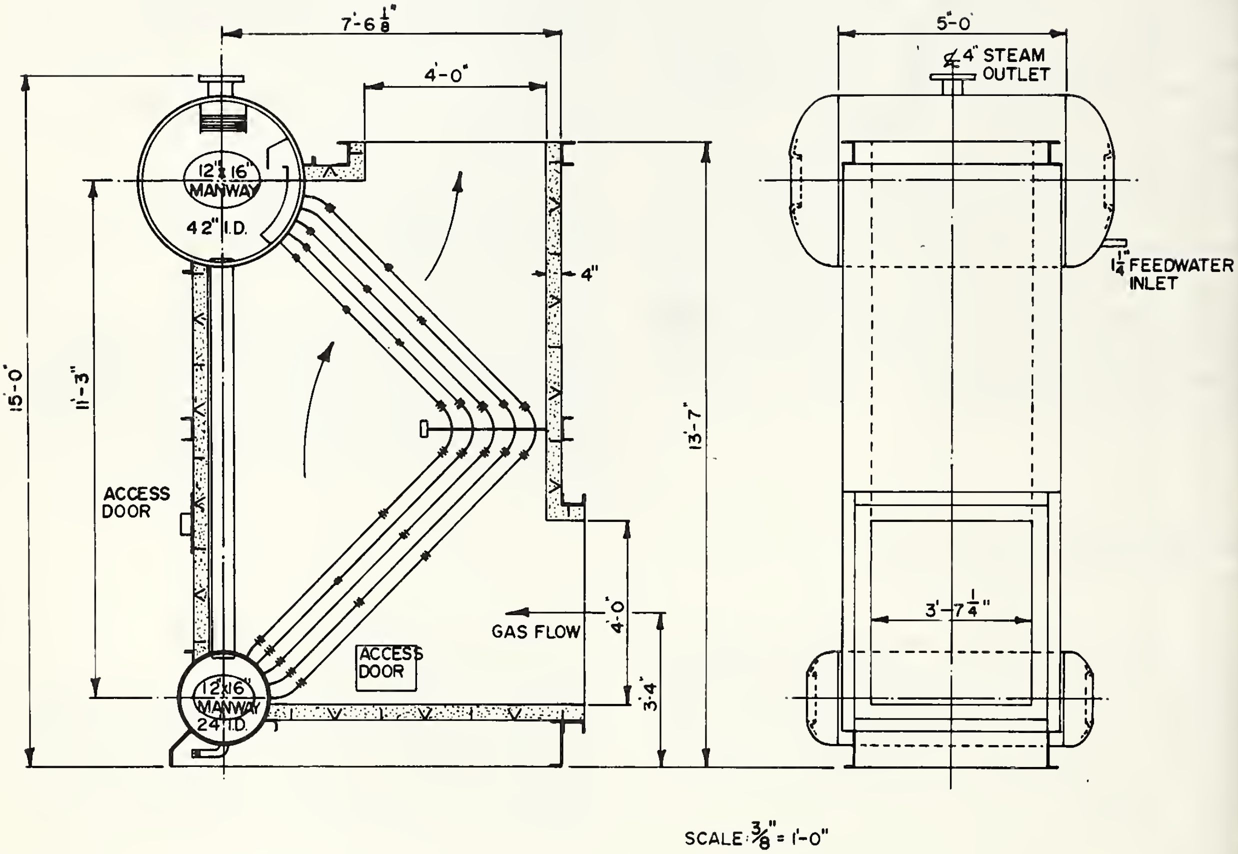

Figure 4.9.1 *Waste heat boiler.*

Table 4.9.1

PERFORMANCE AND DATA

PURCHASER:	DATE:	DES.:
FOR:	TYPE:	PROJ.:
PLANT SITE:	DRAWING NO.:	DRAWN:
PURCHASERS REFERENCE:	NUMBER	EST.:

UNIT							
HEATING GAS SIDE	Gas flow	lb/h		73,300			
	Design pressura/temparature	in/F		20			
	Operating pressure at inlet	in					
	Operating pressure at outlet	in					
	Pressure drop	in		3.5			
	Temperature at inlet	F		1,400			
	Temperature at outlet	F		628			
	Tamperature drop	F		772			
	Specific heat (ave.)	Btu/lb/F		.28			
	Heat given up	MBtu/h		15.85			
	Efficiency	%		98			
	Fouling factor	hr, sq ft, F/Btu		.001			
	Auxiliery fuel (lhv=)	lb/hr					
HEATED SIDE	Fluid flow	lb/h		15,000			
	Design pressure/tamparature	psig/F		200			
	Operating pressure at inlet	psig		150			
	Operating pressure at outlet	psig		150			
	Pressure drop	psi		-			
	Temperatura at outlet	F		366			
	Tamperature at inlet	F		200			
	Temperature rise	F		166			
	Heat absorbed/lb of steam	Btu/lb		1,036			
	Heat absorbed	MBtu/hr		15.52			
	LMTD	F		562			
	Overall heat tr. coeff.	Btu/hr, sq ft, F		9.63			
	Fouling factor	hr, sq ft, F/Btu		.001			
	Steam quality	% or PPM		3			
	Blow-down water	%		.5			
MECHANICAL DATA	Heating surface	sq ft		2,970			
	Tuba diameter	in		2			
	min. thickness	in		.105			
	length						
	material			SA-178A			
	No. of rows transverse to flow			28, 10F			
	Transverse spacing (S_T)	in		3 3/4			
	No. of rows longitudinal to flow			5 (2 half rows bare)			
	Longitudinal spacing (S_L)	in		4			
	Fin no/in			6			
	height	in		5/8			
	thickness	in		.05			
	material			CS			
	Drum (header) diameter	in		42.24			
	length	in					
	material	in		SA-515-70			
	Inlet nozzla diamater	in		1¼			
	Outlet nozzla diametar	in		4			

4.9.4 Cost evaluation of the Pollution Control and Waste Heat Recovery System

One way of analyzing the cost of this investment is as follows:

Fume collecting hood, ductwork, and forced draft fan	$ 9,000
Incinerator	25,000
Waste heat boiler	32,000
Stacks	2,000
Piping for steam line	2,000
Field installation	30,000
Total Cost	$100,000

This corresponds to a production of about 1500 lb/h of steam from the fumes. Since steam was at this time estimated to cost $1.00/1000 lb, this represents a savings of ($1.50) (24 h/day) (356 operating days/yr) (0.65 boiler efficiency) = $6,600/yr.

A direct analysis of payback period is then given by dividing $100,000 by $6600; the result is a payback period in excess of 15 yr, which would indicate that the investment should not be made.

However, this is not a realistic estimate of the payback period on the waste heat equipment. As discussed in chapter 3, it is erroneous to burden the waste heat installation with the cost of environmental equipment. The environmental protection measures were required to correct the pollution problem and thus a realistic payback analysis for the waste heat installation (in this case, the steam production from the waste heat boiler) should include only the additional expenditures which were chargeable to the components of that part of the system.

In this case, the total benefit from the waste heat boiler was the provision of 15,000 lb/h of 150 lb/in^{2}g steam which formerly had to be purchased at a cost of $1.00/1000 lb. This is equivalent to $45,000/yr. The additional cost of installing the waste heat boiler was approximately $37,000 which gives a payback period of approximately 10 mo.

4.9.7 Review of the Application

The incinerator was installed primarily because of pollution control regulations. However, process steam was needed in the plant and it was advantageous to add the waste heat recovery boiler to the incinerator. Although the waste heat boiler had about the same efficiency as the fired boiler, less fuel was used to generate the same amount of steam because of the energy recovered from the fumes. The waste heat boiler was designed to satisfy the steam requirement in the specification, not for the maximum heat recovery. Therefore, there was additional sensible heat remaining in the stack flue gas which could have been recovered by an economizer on the top of the boiler if more steam had been needed.

CASE STUDY

4.10 Waste Heat Recovery in a Food Processing Plant (Hot Water Heating via Air Conditioning Cycle)

4.10 Hot Water Heating Via Air Conditioning Cycle in a Food Processing Plant

4.10.1 Background

As early as 1955, a Florida electric company recognized that large users of hot water could capture waste heat from building heating and air conditioning systems. In that year it purchased a heat pump water heater to evaluate hot water heating with a standard refrigeration compressor. In this system, building cooling was a by-product. The excellent performance obtained with this system over a 6-yr period suggested further experiments would be of value.

In 1961, the electric company decided to investigate hot water heating by use of rejected heat from a standard air conditioning cycle.

The following parameters were chosen as criteria for selection of the test facility:

(1) The air conditioning system should serve a facility which requires cooling through most of the year.

(2) Hot water usage should be large so that the facility would generate significant electricity savings.

(3) The facility should operate around the clock so that performance could be evaluated in a continuous mode.

Since the selected test facility was a restaurant located in Florida, where the air conditioning season is about 8 mo, criterion (1) was satisfied. Selection of a restaurant (and its associated food processing facility), where customer comfort is vital, served to ensure that year-round air conditioning would be practiced. Also, food processing facilities use large quantities of hot water, which satisfied the second criterion. Finally, a restaurant which was to operate 24 h a day was selected, thus satisfying the third criterion.

4.10.2 Description of Pertinent Existing Equipment

The facility selected for installation of the waste heat recovery unit was under construction at the time of its selection. Equipment room space had been fixed and all air conditioning and water heating equipment had been ordered.

The central air conditioning equipment consisted of 10-ton and 15-ton air to air refrigeration units. The original water heating system was a 211-gal water heater having a 50 kW resistance heating element and a 550-gal hot water storage tank. The storage tank was connected to the heater by a piping loop which had a circulating pump installed in it. This arrangement was dictated by space limitations.

4.10.3 Specifications for Design of Waste Heat Management Device

Since the waste heat recovery unit was built partly for evaluation purposes, to determine how much hot water could be produced by rejected heat from the existing cooling system, preconceived requirements on output were not imposed.

4.10.4 Description of Final Design of Equipment

The 15-ton compressor was selected as the waste heat source. A Halsted-Mitchell tube in tube heat exchanger was chosen to transfer the heat of compression from the hot refrigerant gas to the water. The hot gas was circulated through the outer tubes and the water through the inner tubes of the exchanger. The water side of the heat exchanger was connected by a piping loop to a 176-gal preheat tank. Water was circulated through the loop by a pump (in addition to the pump mentioned previously). This preheat tank was connected to the electric water heater inlet. Figure 4.10.1 shows the equipment, piping and instrumentation.

Operation of the equipment proceeded as follows: The appropriate pump circulated water between the heater and the storage tank. This maintained the temperature of the water in the storage tank the same as that of the water in the heater. Simultaneously, the other pump circulated water between the heat exchanger and the preheat tank to maintain high temperature in the preheat tank. When hot water was drawn off from the storage tank for use, hot water from the water heater refilled the storage tank. Cold water from the city main replaced the hot water after it left the hot water system. When valves A and B were opened, the waste heat recovery system became part of the water heating system. Thus, instead of cold makeup entering the water heater, heated water from the preheat tank was supplied. This water was heated by heat transferred from the heat exchanger at low cost (pump electricity for

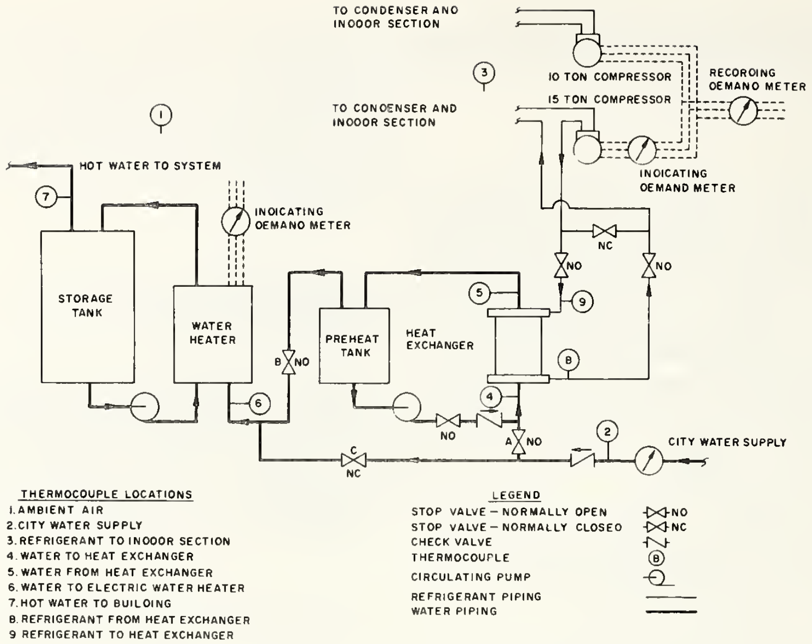

FIGURE 4.10.1 *System schematic diagram.*

water circulation being the only purchased power needed) since it would have been released to the atmosphere otherwise.

4.10.5 Cost Study of Waste Heat Recovery Unit

Total cost of the waste heat recovery unit was $2370. This included capital cost, labor, and installation.

Calculations showed (see app.) that 49 percent of the heat required for hot water during the test year of September 1961 to September 1962 was supplied by the waste heat recovery unit. This translated to a yearly saving of $61,900 kWh for the year of the study. At 1961 electricity costs of $0.013/kWh this was a saving of $805 for that year.

A simple pay-back time based in 1961 electricity costs is

$$\text{Pay-back Time} = \frac{\$2370}{\$805/\text{yr}} = 2.94 \text{ yr}$$

Pay-back time would be longer if cash flow had been discounted, but on the other hand, using higher 1975 fuel costs would result in a shorter pay-back period.

A more detailed analysis of energy savings is given later.

4.10.6 Difficulties Encountered in Applying the New Design

The major problem encountered with this equipment was in the constraints imposed by the lack of adequate space for efficient equipment installation. For example, the preheat tank size was not optimum because of space considerations.

4.10.7 Review of the Application

Conditions were not ideal for the installation of this unit. Space limitations had previously been imposed, making it necessary to use a nonoptimum tank size. Also, two small compressors had been chosen instead of one large unit which would have resulted in greater energy savings. However, despite these drawbacks, impressive energy and dollar savings were still obtained. The savings from the waste heat installation paid for its original cost many times over during the years it has been in service.

This waste heat recovery application is now under scrutiny by many organizations in Florida, including large food processing plants, supermarkets, and office buildings. Several fast food chains are currently installing trial units. The electric company has assisted in installation of waste heat water heaters on several other restaurants and motels. Also, many similar (but smaller) waste heat recovery units have been installed on home air conditioners in Florida.

4.10.8 Recommendations[1]

(1) Determine sizing and number of air conditioning units on the *combined* requirements of air conditioning and water heating.

(2) Provide sufficient space to permit selection of a tank size that provides optimum hot water volume.

(3) Use a storage water heater containing the electric heating element to avoid using separate units described in this study. This would eliminate one circulating pump and some piping.

---

[1] Taken in part from a report by John A. Bedingfield and Associates, Inc.

APPENDIX

ONE-YEAR SUMMARY

(September 8, 1961 through September 8, 1962)

Outside average temperature	72 F
Total hot water used	813,196 gal
Bonus hot water from heat exchanger	396,270 gal
% hot water from heat exchanger	49%
Average tap water temperature (Weighted yearly average)	75 F
Average building water temperature	140 F
Degree F temperature rise	65 F
Total energy needed for water heating	450,471,345 Btu
Energy obtained from heat exchanger	211,110,676 Btu
% energy from heat exchanger	47%
Total kWh used in heating water	132,026 kWh
kWh from electric heater	76,640 kWh
kWh from heat exchanger	61,873 kWh
% kWh from heat exchanger	47%

SAMPLE CALCULATIONS

(September 8, 1961 through September 8, 1962)

1. Gal of water = (ft^3 of water) × (7.48 gal/ft^3)
   Example—(3057 ft^3) × (7.48 gal/ft^3) = 22,866 gal
2. Total Btu = (8.3 lb/gal) × (gal hot water) × (total temp. diff.) × (1 Btu/lb F)
   Example—(8.3 lb/gal) × (68,951 gal) × (60 F) (1 Btu/lb F) = 34,337,598 Btu
3. Total kWh for heating water (no losses included) = Total Btu/(3412 Btu/kWh)
   Example—34,337,598 Btu/ (3412 Btu/kWh) = 10,064 kWh
4. kWh Input (losses included) = Total kWh for heating water (no losses included + hot water tank losses)
   Example—10,064 kWh + 551 kWh = 10,615 kWh where hot water tank losses = \$60,636 Btu/day) × (31 days) = 551 kWh
5. kWh from heat exchanger (no losses included) = kWh input (losses included)—Total kWh from electric heater (losses included)
   Example—10,615 kWh − 3,680 kWh = 6,935 kWh
6. Btu supplied by heat exchanger = (kWh from heat exchanger) × (3412 Btu/kWh)
   Example—(6935 kWh) × (3412 Btu/kWh) = 23,662,220 Btu
7. Temperature difference (preheat to tap water) = (Btu from heat exchanger)/(total gal) × (8.3 lb/gal) × (1 Btu/lb F)
   Example—23,662,220 Btu/(68,951 gal) × (8.3 lb/gal) × (1 Btu/lb F) = 41.3 F
8. Bonus hot water = temp. diff. (preheat to tap)/temp. diff. (building to tap) × (total gal H_2O)

   $$\text{Example—}\ \frac{41.3\ \text{F}}{60\ \text{F}}\ (68{,}951\ \text{gal}) = 47{,}459\ \text{gal}$$

9. Paid for hot water = total hot water − bonus hot water
   Example—68,951 − 47,459 = 21,492 gal
10. Preheat water temp. = tap water temp. + temp. diff. (preheat to tap)
    Example—80 F + 41.3 F = 121.3 F

SYSTEM TANK LOSSES

Hot Water Storage Tank

(761 gal) × (8.3 lb/gal) = 6.316 lb
From charts—0.4 F drop/h
(6316 lb) × (0.4 F/h) (24 h) (1 Btu/lb F) - 60,636 Btu/day loss

Preheat Tank

(176 gal) × (8.3 lb/gal) = 1461 lb
From charts—1 F drop/h (1461 lb) × (24 h) × (1 F/h) × (1 Btu/lb F)
= 34,059 Btu/day loss

CASE STUDY

4.11 Waste Heat Recovery in an Electrical Firm (Hot Water Generation via Electric Heat Pump)

4.11 Hot Water Generation Using an Electric Heat Pump

4.11.1 Background

A major industrial electrical firm began a long-term energy study in one of its subdivisions in 1972 and 1973 which led to development of the following heat pump application for waste-heat recovery. That study concluded that in addition to satisfying the current need of gas and oil, a long-term solution would have to combine conservation measures with a shift to energy sources that are plentiful—mostly uranium and coal. Since, for most applications, uranium and coal have to be converted to electricity, the study proceeded to identify end uses of energy where electricity could be substituted for gas and oil.

A feasibility analysis indicated that coefficients of performance (COP) in the 2.5 to 3.5 range could be expected in industrial size heat pumps employing the same basic principle as a residential heat pump. (Coefficient of Performance is defined as the ratio of the heat delivered to the compressor work required). By using a much larger compressor and working at higher temperature levels than the residential heat pumps, such an industrial heat pump could be applied not only to produce process steam but also to produce hot water, if preferred.

Some suitable sources of hot waste heat were brought to the study group's attention—extruder cooling water, injection molding cooling water, refrigeration equipment cooling water and, in one of their own plants, arc welder cooling water, for examples. The last-mentioned source (arc welder cooling water) is the basis for this case history.

The case being summarized here is considered as a demonstration installation. It was installed in May 1975, and will be available for inspection by company people and representatives of other interested industrial firms and utilities.

4.11.2 Description of Equipment Which Was Replaced

Before installation of the demonstration unit,[1] a gas boiler supplied 300 F (50 lb/in^{2}g) steam to two processes. The first, a spray degreaser, had to be maintained at 140 F and required 500,000 Btu/h. The second, a prepainting treatment for steel radiators, had to be maintained at 170 F and required 600,000 Btu/h. Three problems were evident:

(1) The 12-yr-old gas fired boiler frequently broke down. Plant personnel had to decide whether to replace it or to supply heat to the two processes another way.

(2) Plate type heat exchangers were used at the process tanks to transfer heat from the steam into the tank solutions. The combination of high heat exchanger temperature due to the 300 F steam and chemicals in the process solutions required frequent and costly heat exchanger replacement.

(3) Plant personnel were concerned about increasing natural gas prices and possible production curtailment because of allocations.

4.11.3 Requirements for Design of the Demonstration Unit (Electric Heat Pump Unit)

The demonstration installation was to replace the gas boiler and thereby reduce plant natural gas consumption. Performance experience to be noted under normal in-service conditions. The unit was to be supplemented with heat storage capability, permitting it to draw its electric power during the utility's off peak period.[2] The off-peak utilization experience gained is of great importance, both in terms of increasing future electric utility load factors and in reducing industrial electric demand charges, but it was not a necessary feature of a heat pump installation. This off-peak utilization, while not energy conserving *per se*, represents a substantial cost saving.

Allowing for a safe capacity margin, small heat transmission losses, and reasonable cold start-up time, it was determined that the heat pump installed could be capable of delivering 1,400,000 Btu/h to the combined degreasing and prepainting processes which previously received their heat from 300 F (50 lb/in^{2}g) steam.

4.11.4 Description of Final Design of the Demonstration Unit

The final configuration of the demonstration unit is shown in figure 4.11.1. The waste heat source was primarily electric seam welders which

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[1] The actual trade name cannot be included here, but rather it will be referred to as either "the demonstration unit" or, for similar proposed units, as "an industrial electric heat pump."

[2] Since heat storage capability was not an essential part of the demonstration heat pump installation, features peculiar to the heat storage have not been included here.

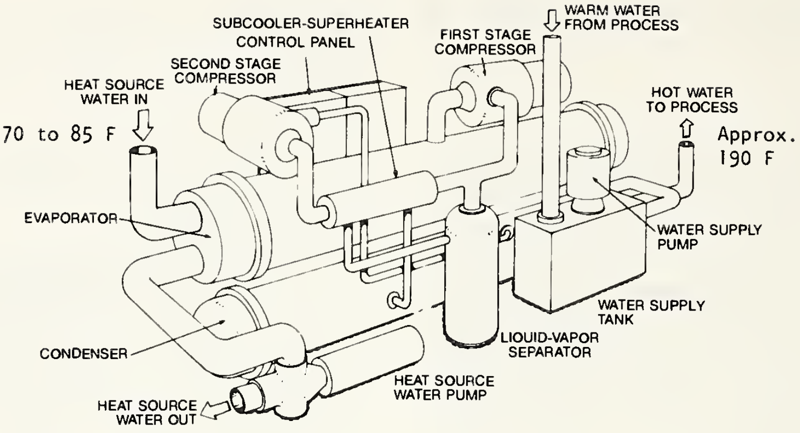

FIGURE 4.11.1 *Demonstration unit.*

rejected about 1,200,000 Btu/h into the plant cooling water and heated it to about 85 F. Since the demonstration unit must operate even though the welders are shut down, it had to be capable of producing its rated capacity using supply water down to 70 F. To maximize the coefficient of performance (COP), the unit was designed to deliver hot water at 190 F rather than steam. Backup resistance heaters were provided to permit compressor inspection without interruption of production.

Figure 4.11.2 is a schematic flow diagram of an industrial electric heat pump designed to deliver 180 F hot water using, in this case, free heat available from 90 F water. This particular design though showing slightly different temperature conditions, is typical of the flow diagram for the demonstration unit.

Figure 4.11.3 shows the temperature control scheme at the process tanks. Automatic temperature control was achieved by individually actuating the heat exchanger circulating pumps when solution temperatures dropped to their set points. The unit controlled its 190 F hot water output temperature automatically by modulating its output down to about half its rated capacity. At lower loads, the unit shut down and automatically restarted when necessary.

As stated earlier, the unit was designed for a capacity of 1,400,000 Btu/h heat delivery to the combined degreasing and prepainting processes. It required 180 compressor horsepower, had a COP just over 3 with a source temperature of 70 F (welders inoperative for a considerable period), annual average COP of 3.3, and an expected COP of 3.6 with a source temperature of 85 F (welders running steadily).

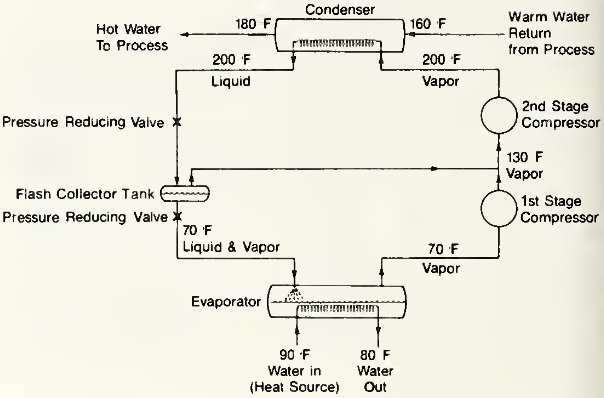

FIGURE 4.11.2 *Heat pump schematic diagram, two stage compression, 90 °F heat source 180 °F hot water delivered.*

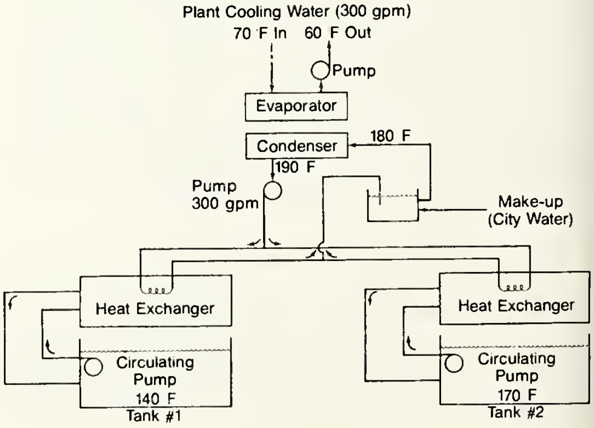

FIGURE 4.11.3 *Demonstration unit water flow.*

4.11.5 Cost Study of the Demonstration Unit

The firm has stated that even for subsequent installations, the installed cost exceeds the installed cost of a gas- or oil-fired boiler. Favorable industrial electric heat pump economics, therefore, depend on electrical energy costs when using the heat pump being lower than the cost of purchased fuel for the boiler.

Using the industrial electric heat pump as treated here, energy savings in cents/MBtu delivered to the process are given by the following formula:

$$ES = \frac{C_F}{N_F} - \frac{C_E(10^3)}{\text{EER}}$$

where

ES = industrial electric heat pump energy savings in ¢/10^6 Btu delivered to the process
C_F = fossil-fuel energy cost in ¢/10^6 Btu
N_F = fossil-fuel boiler system efficiency
C_E = electric cost including demand charges in ¢/kWh
EER = Energy Efficiency Ratio in Btu/W•h = 3.413 (COP)
3.413 = number of Btu/h equal to a watt
COP = Coefficient of Performance

A representative of the electric firm has listed operating costs of various steam generators as given below (presumably typical 1975 costs).

Operating Costs of Various Steam Generators
1 MBtu/h

Boiler Type	Cost/h	Assumptions
Electric	$5.86	Electric Boiler—100% efficient 2¢/kWh
Gas	1.60	Gas Boiler—75% efficient 12¢/therm
Oil	3.60	Oil Boiler—70% efficient 35¢/gal
Coal	1.67	Coal Boiler—60% efficient $25/ton

By way of illustration consider the following two examples (not provided with the original case study).

The first example compares a gas boiler and an electric heat pump.

Using (a) C_E at 2¢/kWh
(b) C_F at 12¢/therm = 12×10^{-5}¢/Btu = 120¢/MBtu
(c) N_F at 75% for gas
(d) EER = 10.6 = 3.413 × 3.1, $\frac{\text{Btu}}{\text{W}\cdot\text{h}}$ using COP of 3.1 as typical for the industrial electric heat pump.

$$ES = \frac{120}{0.75} - \frac{2(10^3)}{10.6} = 160 - 188.7$$
$$= -28.7¢/10^6 \text{ Btu}$$

That is, the electric heat pump would cost 28.7¢/10^6 Btu *more* for operation than the gas boiler. Considering that the price of gas is currently being kept at an artificially low level compared to other fossil fuels, the competitive position of the heat pump will improve as natural gas prices are permitted to increase.

As a second illustration, compare an oil boiler to an electric heat pump.

Using (a) C_E at 2¢/kWh
(b) C_F at 35¢/gal
(c) N_F at 70%
(d) EER = 10.6

$$ES = 360 - 188.7 = 171.3¢/10^6 \text{ Btu}$$

That is, the electric heat pump would cost \$1.71/10^6 Btu *less* for operation than the oil burner.

The essential point in heat pump savings is that the user is taking advantage of the highly developed and very efficient boiler and generator system of a public utility, if he were to use his own boiler, his boiler energy conversion efficiency would in general be much lower.

4.11.6 Limitations in Applying the Industrial Electric Heat Pump

An industrial electric heat pump, such as the demonstration unit being discussed here, must have a heat source which can be considered "free" if it is to be successfully used as a waste heat recovery device. The average operating temperatures of this waste heat source and of the higher temperature heat sink to which the unit delivers thermal energy for some process operation are very important limitations. Thus, "ideal" matching of the waste heat source and the process sink is essential to efficient use of the heat pump.

In real world heat pumping, several factors prevent achievement of the theoretical COP. For example, the need for adequate heat transfer rates restricts the selection of evaporator and condenser temperatures. In actual vapor compressors, there are losses due to fluid friction, valves, heat transfer to the compressor walls, diffuser losses, etc.

Figure 4.11.4 shows a typical industrial heat pump COP taking into account the factors mentioned above (and others) for an installation having a 90 F heat source. The lower solid-line curve reflects the performance achieved. For example, a COP of about 3.0 will be achieved if 190 F hot water is delivered. If steam at 230 F is delivered, the COP drops to about 2.5 and keeps falling as delivery temperature is increased. Resistance heating (shown dotted) may be said to have a COP of 1.0 based on the nonexistence of a heat source; i.e., $T_C = 0$ R.

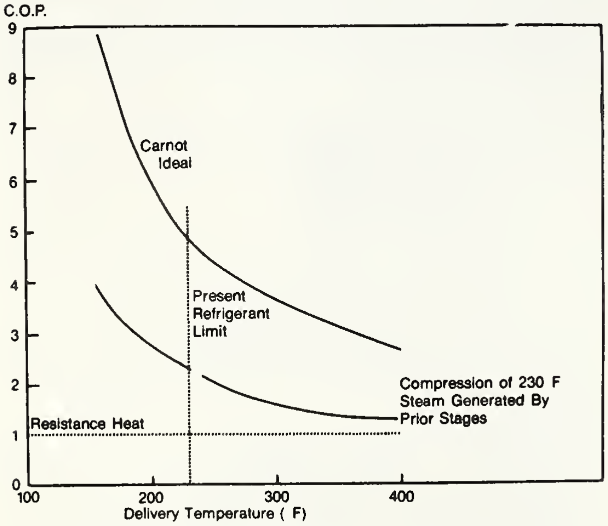

FIGURE 4.11.4 *Typical heat pump performance. Two stages, 90 °F sources, 20 °F differential in evaporator and condenser.*

Several key points should be noted:

(1) It should not be assumed that delivery of, say 300 F steam means a marginally attractive COP. The COP depends on the temperature of the heat source and if figure 4.11.4 were redrawn for an available 200 F heat source, the result would be a much higher COP.

(2) The bottom solid curve illustrates the vital importance of using a heat pump to deliver heat at the lowest temperature consistent with the needs of the process.

(3) It should not be inferred from the preceding discussion that the same heat pump unit is capable of delivering hot water or steam. Rather, for optimal performance, the configuration is determined for the specific application considering the heat source temperature, the delivery temperature required, and the heat delivery capacity needed.

(4) High temperature waste heat cannot be supplied directly to the heat pump because of refrigerant limitations.

4.11.7 Review of the Application and Changes It Produced in Energy Use and Costs

An industrial heat pump application in which waste heat is being recovered to aid in processing elsewhere in the plant has been presented. Specifically, waste heat from electric seam welders, which has been transferred to by cooling water, is then recovered from that cooling water to be used as a source by the heat pump. Adding the energy input needed to drive the heat pump to this recovered waste heat, one obtains the energy delivered by way of hot water for process use. Previously, a gas boiler produced all the energy needed for process use by way of higher temperature steam.

Conceptually, *two major energy-saving changes* are then evident—(1) the hot water is supplied to the processing at a temperature closer to that actually needed than the previously-supplied steam (190 F water vs 300 F steam); and (2) the recovery of waste heat reduces the primary energy needed which is directly chargeable to the processing costs.

Costwise the application of the actual demonstration unit appears to be inferior *at the present time* to the unit it replaced. As stated earlier, installed cost of the demonstration heat pump unit exceeded that which would have been needed for installation of a replacement gas boiler. Further, operating cost of the new unit appears to exceed that of the gas boiler by about 30¢/10^6 Btu delivered to processing needs. Had it replaced an oil-fired boiler, operating cost may have been *reduced* by about \$1.70/10^6 Btu delivered to process needs. In-service data, which are not yet available, will permit more precise comparisons.

Such considerations as comparative maintenance, comparative life, and possible future changes in comparative costs of electricity and gas have not been examined in any detail. As stated earlier, maintenance of the gas boiler and auxiliary equipment and concern over potentially large gas price increases were prime factors in choosing the site for the demonstration unit.

The demonstration unit was installed about the middle of May 1975, and has been operating satisfactorily. Prior to installation it had been on test at another company site since the first of January 1975, where it also performed well. Throughout the test period, a constant check was kept on temperatures, pressures and flow rates in all parts of the cycle. That information, plus the kilowatt/hours used, permitted calculation of the Energy Efficiency Ratio (EER) which ran consistently at 10.6, or a Coefficient of Performance (COP) of 3.1, as used in the cost study earlier.

4.11.8 Suggestions for Potential Users of Industrial Heat Pumps for Waste Heat Recovery

Certain basic conditions must exist for heat pump operations, whether used in conjunction with waste heat recovery or not. If these conditions are met and a waste heat source exists, the stage is set for investigation of the practicality of installing a heat pump. The higher the temperature level of waste heat sources, the greater is the potentially-recoverable energy content, but *high-temperature waste energy above 230 F cannot be directly supplied to the heat pump* because of the limit imposed by present compressor and refrigerant technology. Typical free (waste) heat sources for industrial plants include:

Overhead Vapors from Distillation Processes
Warm Water Effluent from Plant Processes
Refrigeration Equipment Cooling Water
Air Compressor Cooling Water
Electric Welder Cooling
Extruder Cooling Water
Injection Welder Cooling Water
Cooling Tower/Pond Water
Flue Gases
Condenser Cooling Water from Power Plants
Bodies of Water—River, Lake, Ocean
Air

High-temperature flue gases, for example, are not a likely source of waste energy for direct use with a heat pump unit. A radiation recuperator can be readily used with such gases to preheat combustion supply air for effective waste heat recovery, whereas to use a flue gas to liquid water exchanger with subsequent water supply to a heat pump for process energy supply seems completely out of consideration.

Research and development of new refrigerants may raise the limit of heat pump use to as high as 400 F. In any case, it should continue to be considered a low-temperature heat recovery device and as such may have application in the following major uses:

Washing, blanching, sterilizing, and clean-up operations in food processing
Grain drying
Metal cleaning and treating processes
Recycling heat in distillation operations in the food and petrochemical industries, and
Industrial space heating.

CASE STUDY

4.12 Waste Heat Recovery on a Dental Products Plant (Heat Wheel)

4.12 Use of a Heat Wheel to Reduce Energy Costs in a Dental Products Plant

4.12.1 Background

A dental products plant contemplated addition of a paint shop to its already existing plant in early 1972. State law required a complete air change in its paint shop at 2-min intervals; at the same time, air movement at the paint spray station had to have a minimum velocity of 120 ft/min/ft^2 of booth opening; and maintenance of a 75 F temperature in the paint shop during the winter was desired.

The plant engineer, working closely with the engineering staff of the corporation which owned the plant, conducted an extensive analysis of the situation and ascertained that 90 percent of the wasted sensible heat in the exhausted air could be recovered and used to preheat incoming air for the paint shop.

As a result of the investigation, the plant decided to install a heat wheel as the energy recovery system. The wheel was manufactured by a well-known company which specializes in this type of system.

4.12.2 Specifications for and Description of Final Design of Heat Recovery Equipment

The major specifications that the recovery equipment had to satisfy were those ventilation requirements imposed by state law and the comfort temperature requirement imposed by the plant supervisors. Energy recovery had to be treated as a bonus if it could be achieved while fulfilling the more pressing environmental control problems.

Exhaust air from the paint shop was collected into a common duct system. The sources of the exhaust air were paint spray booths, fume incinerators on bake ovens, a washer vent and a drying oven.

The final design was an energy recovery package mounted on the plant roof with an 8-ft diameter heat wheel (as discussed in chap. 5) as the heart of the system. The wheel was rotated by a small electric motor. The blended exhaust air from the paint shop entered at 225 F through one side of the wheel and outside air at ambient temperature passed through the other side. The sensible heat exchanged was used to provide the 75 F primary heating requirement. Exhaust air in excess of that needed for the heat exchange was by-passed around the wheel. Figures 5.7 and 5.8, chapter 5, show a typical heat-wheel arrangement. The principal components of the packaged assembly are listed below followed by a general description of possible operating cycles.

Rooftop Energy Recovery Package

Energy recovery wheel
Centrifugal supply fan assembly
Centrifugal exhaust fan assembly
Cleanable supply air filters
Cleanable exhaust air filters
Motorized supply air damper
Gravity exhaust air backdrop damper
Motorized recirculation damper
Inlet louver and screen
Exhaust air inlet
Supply air inlet
Preheat coil (optional)
Reheat coil (optional)
Separable structural steel roof curb
Weatherproof disconnect switch
Prewired control panel
Insulation and weatherproofing
Remote control station
Cooling coil (optional)
Drain connections (total of 2)
Lifting lugs (2 each side—total of 4)

The packaged rooftop energy recovery units as used here have three standard cycles: the energy recovery cycle, the recirculation cycle, and the ventilation cycle. Each cycle is selected through a 3-way switch remotely located.

When operating in the energy recovery cycle, both fans are on, the outside (supply) and exhaust dampers are open and the recirculating damper is closed. The temperature of the supply air is maintained through automatic wheel speed control.

When operating in the recirculation cycle, the supply fan is on, the exhaust fan is off, the outside and exhaust dampers are closed and the recirculating dampers are open. The wheel does not rotate.

The ventilation cycle is used when energy recovery is not required. Both fans are on, the outside and exhaust dampers are open and the recirculating damper is closed. The wheel rotates at constant minimum speed and energy recovery is negligible.

4.12.3 Cost Study of the Heat Wheel

Since this installation is not a retrofit case, no before and after cost study can be made in a literal sense. However, the ventilation requirements had to be met; if these had been met without the heat recovery bonus provided by the wheel, the firm would have incurred almost $14,000/yr in additional heating costs. This is based on the fuel savings of 55,380 gal of propane per year at 1974 cost in the neighborhood of 25¢/gal.

4.12.4 Review of the Application

Use of the packaged heat wheel recovery unit has enabled the dental plant to comply with the state law for a complete air change at 2-min intervals in the paint shop. Air movement at the paint spray booth exceeds the state's minimum requirements, being between 125 and 140 ft/min/ft^2 of booth opening.

The heat exchange effectiveness exceeded the plant engineer's expectations. In addition, employees are experiencing comfortable working conditions. The heat wheel, in operation since the paint shop was added to the existing plant in early 1972, has been maintenance free except for occasional cleaning of the surface of the wheel.

4.12.5 Suggestions for Potential Heat Recovery System Users

Heat wheels are not used as frequently for waste heat recovery as they might be because (1) they are relatively high in initial cost as compared to some other recovery systems and (2) lack of understanding of their capability often leads potential users to think, instead, of more common systems. In the present case, more than one alternative system was considered (as should always be done). For this application the alternate use of a hot water system was rejected because it would have been prohibitively expensive as compared to the heat wheel, would have been very complex to maintain, and (most importantly), would have presented difficulties in meeting the state's air change requirements. The plant engineer estimated that the hot water system would have been four or five times as expensive as the recovery wheel system.

In many central HVAC systems (heating, ventilating and air conditioning), a great amount of costly energy is wasted because of ventilation requirements. The key to preventing such energy waste is, of course, an efficient waste energy recovery system. When considering the use of energy recovery wheels, one must allow for the fact that the wheel itself is simply the mechanism whereby energy from one gas stream is transferred to an adjacent gas stream. Required collecting and ducting of streams must be included and justified as a part of the total system cost.

CASE STUDY

4.13 Waste Heat Recovery in an Incineration Application (Heat Wheel)

4.13 Use of a Heat Wheel in an Incineration Application

4.13.1 Background

A large industrial firm is located in a state which does not require the use of thermal oxidizers to control pollutants and odors from exhaust fumes. However, because there were some complaints about odors, an environmental group in the firm did some studies on thermal oxidizers. They decided to use a ceramic high-temperature recovery (heat) wheel designed specifically for application with incineration.

At lower temperatures metallic wheels are often used; these are cheaper, less fragile, and used in greater numbers than ceramic wheels. Effective uses of such metallic wheels are often overlooked simply because the potential users assume that they are less practical than more familiar heat exchange devices.

A well-known commercial manufacturer of ceramic heat wheels (both with and without incineration application), aware of the need to promote its products, offered to supply a wheel at less than its normal market price to encourage the test project. Some test results are included here, but the testing is incomplete, a 1-yr warranty on the wheel being due to expire in early October 1975.

4.13.2 Requirements and Final Design of the Heat Recovery System

The heat recovery system was required to have the ability to handle the capacity (10,000 SCFM) of the large furnace involved (furnace size about 20 ft by 60 ft). Exhaust fumes were to enter the wheel at about 1400 F in the burner of the integrated heat recovery unit (fig. 4.13.1) before being sent through the other side of the wheel and out to the stack.

The heart of this recovery system is a slowly rotating ceramic wheel that uses heat from the exhaust flow to preheat solvent fumes, which then pass through a modulating flame front to reach their required incineration temperature.

Incineration temperature is precisely maintained through a three mode control system that regulates the burner, the wheel speed and a volume control damper. The thermal stability of the ceramic wheel permits close tolerance of all seals, and the ceramic construction is also resistant to corrosion and thermal shock. The system provides thermal mass construction as well as an air coiled bearing system intended to give long service life with minimum maintenance requirements.

4.13.3 Test Results on the Recovery System

Tests on the wheel, which has a manufacturer's temperature limit rating of 1600 F, are being run at the 1200-1600 F range. Despite the manufacturer's rating, there is evidence that these wheels may perform satisfactorily for short periods at temperatures as high as 2000 F. In this application typical temperatures (fig. 4.13.1) are: 400 F for air entering into wheel, 1100 F for air hitting the burner, and 1400 F for air leaving the burner before passing back through the wheel.

Good performance is being maintained after about 1500 h of operation. At the start of the test, leakage was somewhere between 2 and 3 percent, and has only slightly increased above 3 percent. Test results indicate 72 percent recovery efficiency.

An observer from another potential high-temperature heat wheel user (that is, from a company which was neither the manufacturer nor purchaser of the test case wheel) saw the wheel when it had been removed after about 1500 h of operation. His observations were that the wheel: a) was a bit discolored, but looked in good condition otherwise; b) had held up well in spite of the thermal cycling and the physical beating from particulate matter in the exhaust gases.

4.13.4 Cost Study of the High-Temperature Wheel

The manufacturer of the ceramic wheel-incineration heat recovery system states that the system was sold to the user at $35,000, but actually cost the manufacturer $60,000 to supply. Because of the high recovery efficiency (about 70%) and the high cost of today's fuel, the unit was estimated to require only an 18-mo payback period based on the actual cost to the manufacturer. Units comparable to the tested unit, with a capacity of 10,000 SCFM at 70 percent rated efficiency, are expected on the market in the fall of 1975. Used for incineration and thermal recovery, such a unit should handle 8000-9000 ft^3/min of exhaust fumes and sell for $85,000 to $90,000. As a simple heat exchanger (no

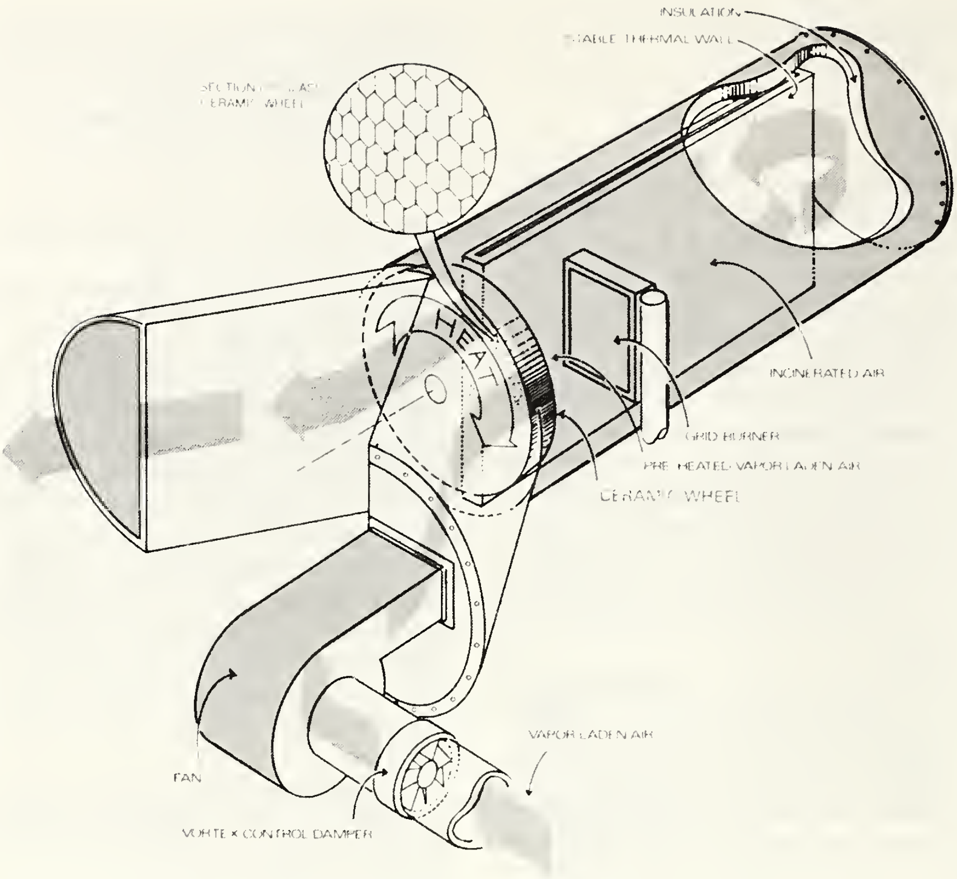

Figure 4.13.1 *Ceramic heat wheel assembly.*

incineration unit), it would be available for about \$60,000.

Estimates of the recovery efficiency and of the energy and cost savings (and, therefore, of the payback period) can be made as follows: Assume pure air enters the wheel at 400 F and is heated by the wheel to 1100 F before being heated to 1400 F by the actual burner operation. The heat recovery efficiency is 70 percent. Energy savings, based on 5000 h/yr operation are approximately

$$10{,}000 \text{ SCFM} \left[0.076 \ \frac{\text{lbm}}{\text{ft}^3}\right] \left[0.24 \ \frac{\text{Btu}}{\text{lbm F}}\right]$$

$$[1100 - 400] \text{ F}\left(60 \ \frac{\text{min}}{\text{h}}\right)\left(5000 \ \frac{\text{h}}{\text{yr}}\right) = 38.3 \ (10^9) \ \frac{\text{Btu}}{\text{yr}}$$

Assuming \$1.20/MBtu as typical fuel cost, the cost savings are

$$\frac{\$1.20}{10^6 \text{ Btu}} \quad (38.3) \quad (10^9) \quad \frac{\text{Btu}}{\text{yr}} \quad = \quad \$46{,}000/\text{yr}.$$

Based on the cost, including installation, varying between \$60,000 to \$90,000, the payback period is between 1.3 and 1.9 yr, and is, therefore, in agreement with the previously stated estimate of 18 mo.

4.13.5 Review of the Application and Suggestions for Potential Customers

Several things may be of special interest to the readers when contemplating this heat recovery application.

(1) A manufacturer of innovative heat recovery equipment and a customer for that equipment have both benefitted from a cooperative testing program.

(2) The customer decided to do something about an odor problem before being required to do so by state regulation.

(3) The ceramic high-temperature heat wheel in the present case has been shown to be cost effective, to require minimum maintenance and to be sufficiently durable; of particular note is the wearability in the face of bombardment by particulates and thermal cycling.

This test has not established that a ceramic heat wheel would be the answer to all high-temperature heat recovery problems. It is more costly initially than a radiation-type stack recuperator, but it is usually easier to install. Furthermore, replacement of a radiation-type stack recuperator would normally require shutting down the exhaust producing process. Proper dampering in the packaged unit treated here will permit continuation of the process (without heat recovery) while a wheel is being replaced.

The manufacturer of the packaged unit discussed here has indicated that research on air to air heat exchangers up to 4000 F might well be considered if an adequate industrial market exists.

Figure 4.13.2 shows another application of the ceramic wheel for primary heat recovery in conjunction with secondary heat recovery via two low-temperature metallic wheels. This represents an actual application by another company and in a different state than the subject case history.

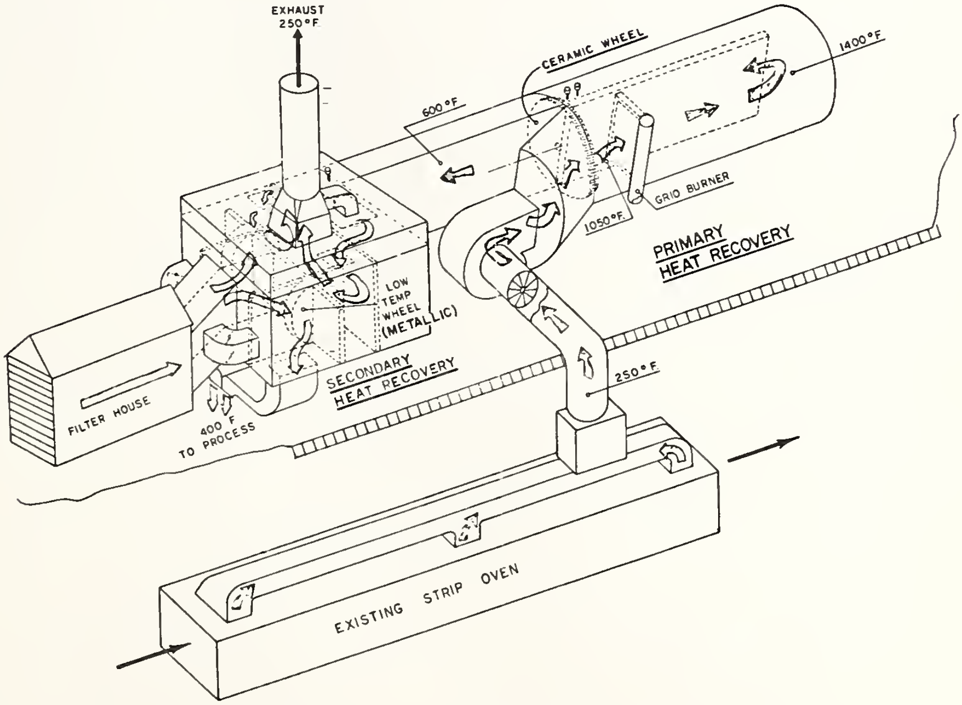

FIGURE 4.13.2 *Two-stage waste heat recovery using heat wheels.*

CASE STUDY

4.14 Waste Heat Recovery in an Automatic Transmission Plant (Radiation Recuperator)

4.14 Waste Heat Recovery in an Automatic Transmission Plant (Radiation Recuperator)

4.14.1 Background

A manufacturer of production heat treating equipment recently conducted tests on its exhaust stack recuperator. The tests were conducted on two identical bent-tube batch furnaces (which it had also manufactured) in an automatic transmission plant. One furnace remained standard and the other was equipped with recuperators. Gas meters measured total volume and rates of gas consumption for each of the two furnaces.

4.14.2 Testing Particulars and Results

The recuperator extracts energy from hot flue gases and transfers this recovered energy to incoming cold air as shown in figure 5.6, chapter 5. By comparing identical production cycles and loads at the same furnace temperatures, the effect of the recuperators in the process could be measured directly.

In all tests on both furnaces, exhausts of radiant tubes were adjusted to show 2 to 2.5 percent O_2, which corresponds to approximately 10 percent excess air. Other test particulars include:

Set point temperature, 1525 F
  Load, approximately 500 lb
Heating cycle (set point temperature recovery),
  33 min for the standard furnace
  34 min for the recuperator furnace
Soaking time,
  59 min for standard furnace
  58 min for recuperator furnace

Table 4.14.1 presents the volume of gas consumed (in SCF) during the tests for the different conditions and shows that the pilots account for a substantial amount of total fuel used. The recuperator saves 29 percent of the fuel when the furnace is on full fire, and 14 percent when it is idling with the pilots off in both cases.

Using the values in table 4.14.1, operating with pilots, 6 days/week, twelve 1-1/2 h cycles per day, and all other times at idle, the gas consumed by the standard batch furnace would be 4.5 Mft^3/yr. Recuperation would save 860,000 ft^3 or 19 percent fuel, in 1 yr. Without pilots, the recuperated furnace saves about 914,160 ft^3 of gas per yr.

In general, the maximum preheating temperature, F, developed by this recuperator will be about 50 percent of the temperature, F, of the exhaust gases.

TABLE 4.14.1. Recuperator tests on batch furnaces

	Gas Consumption, ft^3 Standard Furnace	Gas Consumption, ft^3 Recuperative Furnace	Fuel Saved, ft^3 (%)
Heating:			
Pilots on	515	380	135 (26)
Pilots off	460	325	135 (29)
Soaking:			
Pilots on	430	350	80 (19)
Pilots off	320	240	80 (25)
Total:			
Pilots on	945	730	215 (23)
Pilots off	780	565	215 (28)
Idle:			
Pilots on	365	330	35 (10)
Pilots off	255	220	35 (14)

4.14.3 Cost Analysis of the Recuperator as Installed

At the time the batch furnace operation was tested, the recuperators cost $4620.00. Early 1975 cost would be $4950.00. These figures apply to a new furnace where the recuperators can be installed as the furnace is built. The cost of retrofitting recuperators on existing furnaces would be greater depending on the particular furnace involved, the design of its heating system, and the difficulty of physically locating the recuperators on the furnace.

Consider the operation with pilots, when recuperation would save 860,000 ft^3 of natural gas in 1 yr as shown by the tests. Assuming $1.25/MBtu (or, essentially $1.25/1000 ft^3), this means a savings of 860,000 ft^3 ($1.25/1000 ft^3) = $1075.00/yr. An approximate payback period for the recuperator as installed on a new furnace would then be about $4950/$1075 or 4.6 yr.

4.14.4 Review of the Application and Suggestions for Potential Customers

Some potential customers may consider a payback period of 4.6 years to be too long to justify the inclusion of the recuperation. However, they would normally not expect to amortize the new "standard" furnace; that is, without the recuperator, in as little as 5 yr and by considering recuperative devices as standard equipment, rather than optional, the economic advantages would be more readily apparent.

CASE STUDY SUMMARY CHART

Number	Industry	Installation	Temperature	Capital Investment per Installation
1	Glass	Recuperator	High	$ 4,520
2	Forging	Recuperator	High	6,400
3	Natural Gas	Gas-to-Gas Recuperator	Medium	41,000
4	Can Production	Gas-to-Gas Regenerator	Medium-High	100,000
5	Rubber	Heat Pipes	Low	(proprietary)
6	Dairy	Economizer	Medium	9,040
7	Glass	Economizer	Low	6,046
8	Citrus Processing	Waste Heat Boiler	High	780,000
9	Asphalt Roofing	Waste Heat Boiler	High	100,000
10	Food Processing	Hot Water Heating by A/C Reject Heat	Low	2,370
11	Electrical Equipment	Heat Pump	Low	–
12	Dental Products	Heat Wheel (HVAC)	–	–
13		Heat Wheel	High	60,000
14	Automotive Parts	Radiation Recuperator	High	4,950

Chapter 5

COMMERCIAL OPTIONS IN WASTE HEAT RECOVERY EQUIPMENT

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Introduction

Industrial heat exchangers have many pseudonyms. They are sometimes called recuperators, regenerators, waste heat steam generators, condensers, heat wheels, temperature and moisture exchangers, etc. Whatever name they may have, they all perform one basic function; the transfer of heat.

Heat exchangers are characterized as single or multipass gas to gas, liquid to gas, liquid to liquid, evaporator, condenser, parallel flow, counter flow, or cross flow. The terms single or multipass refer to the heating or cooling media passing over the heat transfer surface once or a number of times. Multipass flow involves the use of internal baffles. The next three terms refer to the two fluids between which heat is transferred in the heat exchanger, and imply that no phase changes occur in those fluids. Here the term "fluid" is used in the most general sense. Thus, we can say that these terms apply to nonevaporator and noncondensing heat exchangers. The term evaporator applies to a heat exchanger in which heat is transferred to an evaporating (boiling) liquid, while a condenser is a heat exchanger in which heat is removed from a condensing vapor. A parallel flow heat exchanger is one in which both fluids flow in approximately the same direction whereas in counterflow the two fluids move in opposite directions. When the two fluids move at right angles to each other, the heat exchanger is considered to be of the crossflow type.

The principal methods of reclaiming waste heat in industrial plants make use of heat exchangers. The heat exchanger is a system which separates the stream containing waste heat and the medium which is to absorb it, but allows the flow of heat across the separation boundaries. The reasons for separating the two streams may be any of the following:

(1) A pressure difference may exist between the two streams of fluid. The rigid boundaries of the heat exchanger can be designed to withstand the pressure difference.

(2) In many, if not most, cases the one stream would contaminate the other, if they were permitted to mix. The heat exchanger prevents mixing.

(3) Heat exchangers permit the use of an intermediate fluid better suited than either of the principal exchange media for transporting waste heat through long distances. The secondary fluid is often steam, but another substance may be selected for special properties.

(4) Certain types of heat exchangers, specifically the heat wheel, are capable of transferring liquids as well as heat. Vapors being cooled in the gases are condensed in the wheel and later re-evaporated into the gas being heated. This can result in improved humidity and/or process control, abatement of atmospheric air pollution, and conservation of valuable resources.

The various names or designations applied to heat exchangers are partly an attempt to describe their function and partly the result of tradition within certain industries. For example, a recuperator is a heat exchanger which recovers waste heat from the exhaust gases of a furnace to heat the incoming air for combustion. This is the name used in both the steel and the glass making industries. The heat exchanger performing the same function in the steam generator of an electric power plant is termed an air preheater, and in the case of a gas turbine plant, a regenerator.

However, in the glass and steel industries the word regenerator refers to two chambers of brick checkerwork which alternately absorb heat from the exhaust gases and then give up part of that heat to the incoming air. The flows of flue gas and of air are periodically reversed by valves so that one chamber of the regenerator is being heated by the products of combustion while the other is being cooled by the incoming air. Regenerators are often more expensive to buy and more expensive to maintain than are recuperators, and their application is primarily in glass melt tanks and in open hearth steel furnaces.

It must be pointed out, however, that although their functions are similar, the three heat exchangers mentioned above may be structurally quite different as well as different in their principal modes of heat transfer. A more complete description of the various industrial heat exchangers follows later in this chapter and details of their differences will be clarified.

The specification of an industrial heat exchanger must include the heat exchange capacity, the temperatures of the fluids, the allowable pressure drop in each fluid path, and the properties and volumetric flow of the fluids entering the exchanger. These specifications will determine construction parameters and thus the cost of the heat exchanger. The final design will be a compromise between pressure drop, heat exchanger effectiveness, and cost. Decisions leading to that final design will balance out the cost of maintenance and operation of the overall system against the fixed costs in such a way as to minimize the total. Advice on selection and design of heat exchangers is available from vendors.

The essential parameters which should be known in order to make an optimum choice of waste heat recovery devices are:

- Temperature of waste heat fluid
- Flow rate of waste heat fluid
- Chemical composition of waste heat fluid
- Minimum allowable temperature of waste heat fluid
- Temperature of heated fluid
- Chemical composition of heated fluid
- Maximum allowable temperature of heated fluid
- Control temperature, if control required

In the rest of this chapter, some common types of waste heat recovery devices are discussed in some detail.

Gas to Gas Heat Exchangers

Recuperators

The simplest configuration for a heat exchanger is the metallic radiation recuperator which consists of two concentric lengths of metal tubing as shown in figure 5.1 below.

The inner tube carries the hot exhaust gases while the external annulus carries the combustion air from the atmosphere to the air inlets of the furnace burners. The hot gases are cooled by the incoming combustion air which now carries additional energy into the combustion chamber. This is energy which does not have to be supplied by the fuel; consequently, less fuel is burned for a given furnace loading. The saving in fuel also means a decrease in combustion air and therefore, stack losses are decreased not only by lowering the stack gas temperatures, but also by discharging smaller quantities of exhaust gas. This particular recuperator gets its name from the fact that a substantial portion of the heat transfer from the hot gases to the surface of the inner tube take place by radiative heat transfer. The cold air in the annulus, however, is almost transparent to infrared radiation so that only convection heat

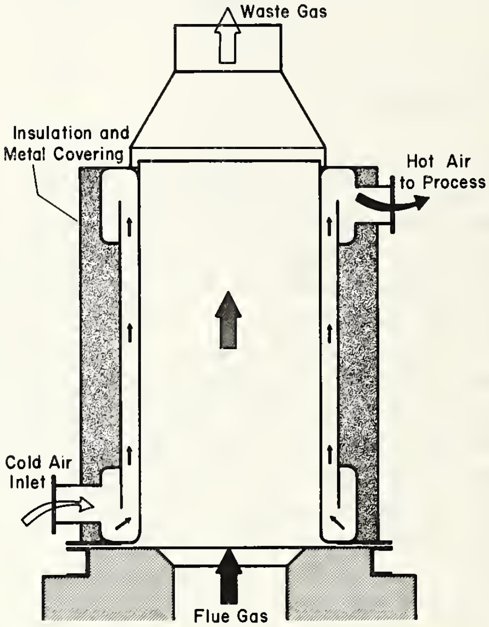


FIGURE 5.1 *Diagram of metallic radiation recuperator.*

transfer takes place to the incoming air. As shown in the diagram, the two gas flows are usually parallel, although the configuration would be simpler and the heat transfer more efficient if the flows were opposed in direction (or counterflow). The reason for the use of parallel flow is that recuperators frequently serve the additional function of cooling the duct carrying away the exhaust gases, and consequently extending its service life.

The inner tube is often fabricated from high temperature materials such as stainless steels of high nickel content. The large temperature differential at the inlet causes differential expansion, since the outer shell is usually of a different and less expensive material. The mechanical design must take this effect into account. More elaborate designs of radiation recuperators incorporate two sections; the bottom operating in parallel flow and the upper section using the more efficient counterflow arrangement. Because of the large axial expansions experienced and the stress conditions at the bottom of the recuperator, the unit is often supported at the top by a free standing suppport frame with an expansion joint between the furnace and recuperator.

A second common configuration for recuperators is called the tube type or convective recuperator. As seen in the schematic diagram of figure 5.2, the hot gases are carried through a number of parallel small diameter tubes, while the incoming air to be heated enters a shell surrounding the tubes and passes over the hot tubes one or more times in a direction normal to their axes.

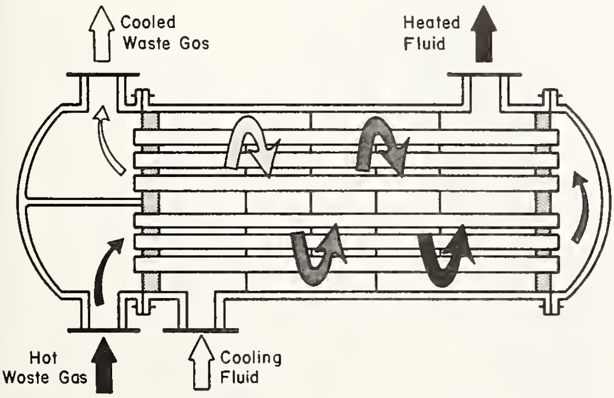


FIGURE 5.2 *Diagram of convective-type recuperator.*

If the tubes are baffled to allow the gas to pass over them twice, the heat exchanger is termed a two-pass recuperator; if two baffles are used, a three-pass recuperator, etc. Although baffling increases both the cost of the exchanger and the pressure drop in the combustion air path, it increases the effectiveness of heat exchange. Shell-and tube-type recuperators are generally more compact and have a higher effectiveness than radiation recuperators, because of the larger heat transfer area made possible through the use of multiple tubes and multiple passes of the gases.

The principal limitation on the heat recovery of metal recuperators is the reduced life of the liner at inlet temperatures exceeding 2000 F. At this temperature, it is necessary to use the less efficient arrangement of parallel flows of exhaust gas and coolant in order to maintain sufficient cooling of the inner shell. In addition, when furnace combustion air flow is dropped back because of reduced load, the heat transfer rate from hot waste gases to preheat combustion air becomes excessive, causing rapid surface deterioration. Then, it is usually necessary to provide an ambient air by-pass to cool the exhaust gases.

In order to overcome the temperature limitations of metal recuperators, ceramic tube recuperators have been developed, whose materials allow operation on the gas side to 2800 F and on the preheated air side to 2200 F on an experimental basis, and to 1500 F on a more or less practical basis. Early ceramic recuperators were built of tile and joined with furnace cement, and thermal cycling caused cracking of joints and rapid deterioration of the tubes. Later developments introduced various kinds of short silicon carbide tubes which can be joined by flexible seals located in the air headers. This kind of patented design illustrated in figure 5.3 maintains the seals at comparatively low temperatures and has reduced the seal leakage rates to a few percent.

Earlier designs had experienced leakage rates from 8 to 60 percent. The new designs are reported to last 2 yr with air preheat temperatures as high as 1300 F, with much lower leakage rates.

An alternative arrangement for the convective type recuperator, in which the cold combustion air is heated in a bank of parallel vertical tubes which extend into the flue gas stream, is shown schematically in figure 5.4. The advantage claimed for this arrangement is the ease of replacing

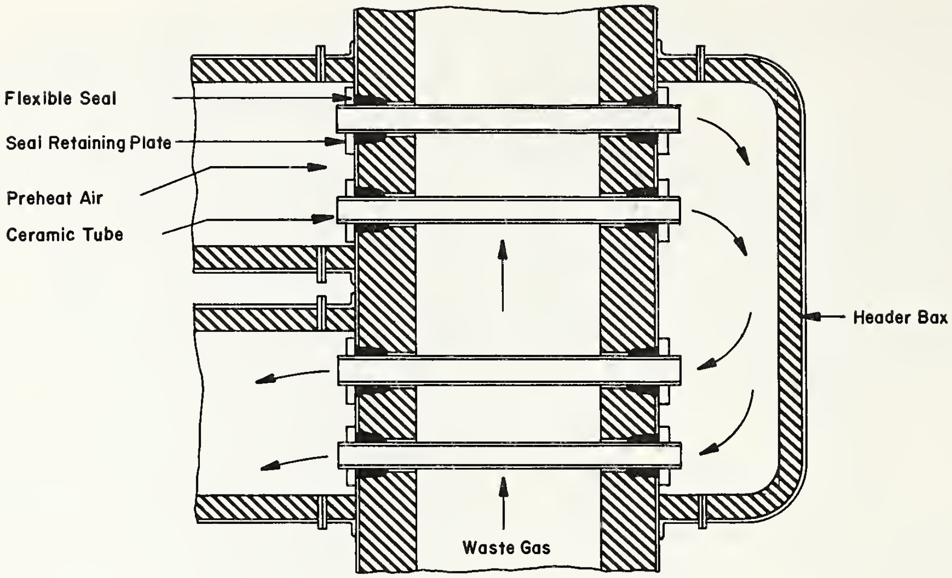


FIGURE 5.3 *Ceramic recuperator.*

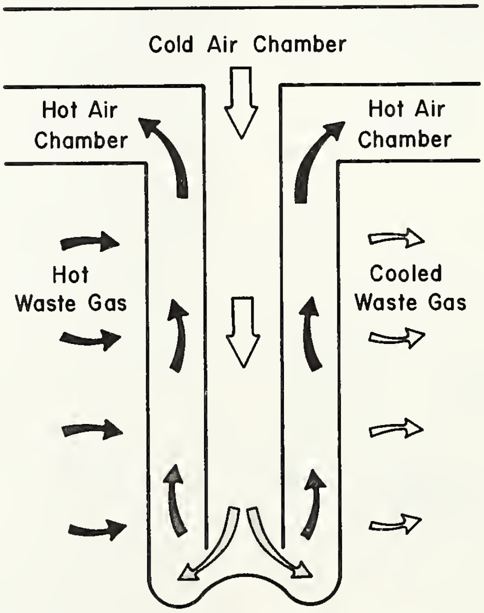


FIGURE 5.4 *Diagram of vertical tube-within-tube recuperator.*

individual tubes, which can be done during full capacity furnace operation. This minimizes the cost, the inconvenience, and possible furnace damage due to a shutdown forced by recuperator failure.

For maximum effectiveness of heat transfer, combinations of radiation type and convective type recuperators are used, with the convective type always following the high temperature radiation recuperator. A schematic diagram of this arrangement is seen in figure 5.5.

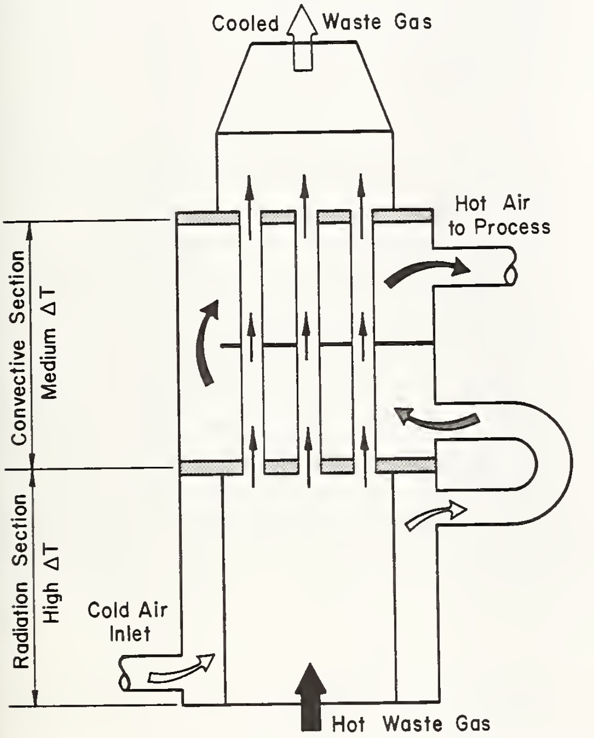


FIGURE 5.5 *Diagram of combined radiation and convective type recuperator.*

Although the use of recuperators conserves fuel in industrial furnaces, and although their original cost is relatively modest, the purchase of the unit is often just the beginning of a somewhat more extensive capital improvement program. The use of a recuperator, which raises the temperature of the incoming combustion air, may require purchase of high temperature burners, larger diameter air lines with flexible fittings to allow for expansion, cold air lines for cooling the burners, modified combustion controls to maintain the required air/fuel ratio despite variable recuperator heating, stack dampers, cold air bleeds, controls to protect the recuperator during blower failure or power failures, and larger fans to overcome the additional pressure drop in the recuperator. It is vitally important to protect the recuperator against damage due to excessive temperatures, since the cost of rebuilding a damaged recuperator may be as high as 90 percent of the initial cost of manufacture and the drop in efficiency of a damaged recuperator may easily increase fuel costs by 10 to 15 percent.

Figure 5.6 shows a schematic diagram of one radiant tube burner fitted with a radiation recuperator. With such a short stack, it is necessary to use two annuli for the incoming air to achieve reasonable heat exchange efficiencies.

Recuperators are used for recovering heat from exhaust gases to heat other gases in the medium to high temperature range. Some typical applications are in soaking ovens, annealing ovens, melting furnaces, afterburners and gas incinerators, radiant-tube burners, reheat furnaces, and other gas to gas waste heat recovery applications in the medium to high temperature range.

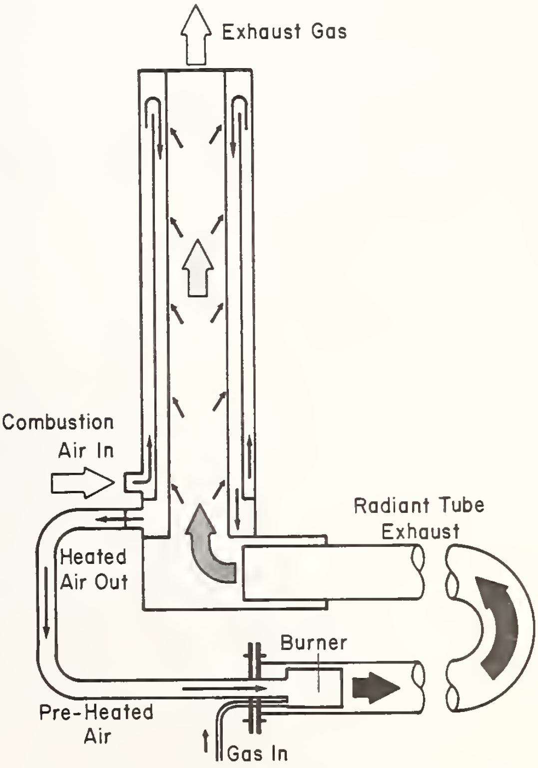


FIGURE 5.6 *Diagram of a small radiation-type recuperator fitted to a radiant tube burner.*

Heat Wheels

A rotary regenerator (also called an air preheater or a heat wheel) is finding increasing applications in low to medium temperature waste heat recovery. Figure 5.7 is a sketch illustrating the application of a heat wheel. It is a sizable porous disk, fabricated from some material having a fairly high heat capacity, which rotates between two side-by-side ducts; one a cold gas duct, the other a hot gas duct. The axis of the disk is located parallel to, and on the partition between the two ducts. As the disk slowly rotates, sensible heat (and in some cases, moisture containing latent heat) is transferred to the disk by the hot air and as the disk rotates, from the disk to the cold air. The overall efficiency of sensible heat transfer for this kind of regenerator can be as high as 85 percent. Heat wheels have been built as large as 70 ft in diameter with air capacities up to 40,000 ft^3/min. Multiple units can be used in parallel. This may help to prevent a mismatch between capacity requirements and the limited number of sizes available in packaged units. In very large installations such as those required for preheating combustion air in fixed station electrical generating stations, the units are custom designed.

The limitation on temperature range for the heat wheel is primarily due to mechanical difficulties introduced by uneven expansion of the rotating wheel when the temperature differences mean large differential expansion, causing excessive deformations of the wheel and thus difficulties in maintaining adequate air seals between duct and wheel.

Heat wheels are available in four types. The first consists of a metal frame packed with a core of knitted mesh stainless steel or aluminum wire, resembling that found in the common metallic kitchen pot scraper; the second, called a laminar wheel, is fabricated from corrugated metal and is composed of many parallel flow passages; the third variety is also a laminar wheel but is constructed from a ceramic matrix of honeycomb configuration. This type is used for higher temperature applications with a present-day limit of about 1600 F. The fourth variety is of laminar construction but the flow passages are coated with a hygroscopic material so that latent heat may be recovered. The packing material of the hygroscopic wheel may be any of a number of materials. The hygroscopic material is often termed a dessicant.

Most industrial stack gases contain water vapor, since water vapor is a product of the combustion of all hydrocarbon fuels and since water is introduced into many industrial processes, and part of the process water evaporates as it is exposed to the hot gas stream. Each pound of water requires approximately 1000 Btu for its evaporation at atmospheric pressure, thus each pound of water vapor leaving in the exit stream will carry 1000 Btu of energy with it. This latent heat may be a substantial fraction of the sensible

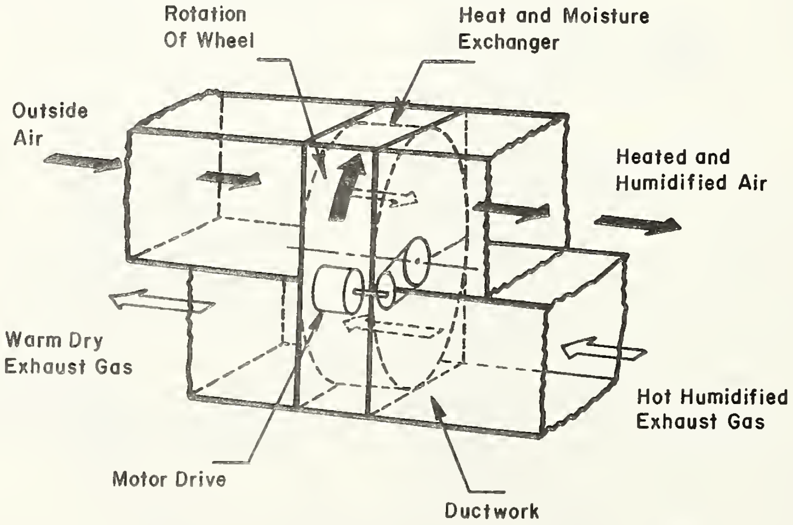


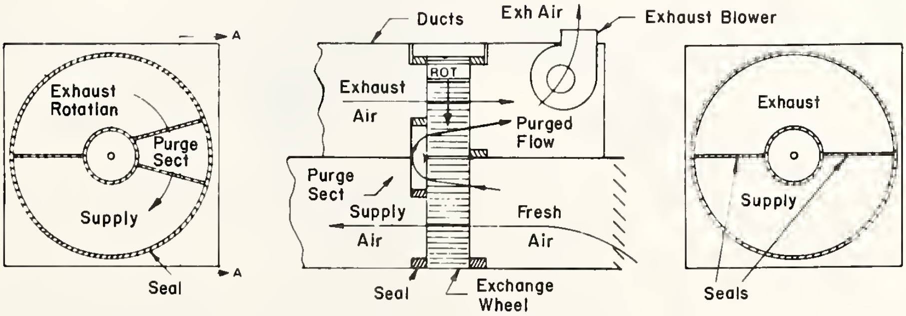
FIGURE 5.7 *Heat and moisture recovery using a heat wheel type regenerator.*

energy in the exit gas stream. A hygroscopic material is one such as lithium chloride (LiCl) which readily absorbs water vapor. Lithium chloride is a solid which absorbs water to form a hydrate, $LiCl \cdot H_2O$, in which one molecule of lithium chloride combines with one molecule of water. Thus, the ratio of water to lithium chloride in $LiCl \cdot H_2O$ is 3/7 by weight. In a hygroscopic heat wheel, the hot gas stream gives up part of its water vapor to the coating; the cool gases which enter the wheel to be heated are drier than those in the inlet duct and part of the absorbed water is given up to the incoming gas stream. The latent heat of the water adds directly to the total quantity of recovered waste heat. The efficiency of recovery of water vapor can be as high as 50 percent.

Since the pores of heat wheels carry a small amount of gas from the exhaust to the intake duct, cross contamination can result. If this contamination is undesirable, the carryover of exhaust gas can be partially eliminated by the addition of a purge section where a small amount of clean air is blown through the wheel and then exhausted to the atmosphere, thereby clearing the passages of exhaust gas. Figure 5.8 illustrates the features of an installation using a purge section. Note that additional seals are required to separate the purge ducts. Common practice is to use about six air changes of clean air for purging. This limits gas contamination to as little as 0.04 percent and particle contamination to less than 0.2 percent in laminar wheels, and cross contamination to less than 1 percent in packed wheels. If inlet gas temperature is to be held constant, regardless of heating loads and exhaust gas temperatures, then the heat wheel must be driven at variable speed. This requires a variable speed drive and a speed control system using an inlet air temperature sensor as the control element. This feature, however, adds considerably to the cost and complexity of the system. When operating with outside air in periods of high humidity and sub-zero temperatures, heat wheels may require preheat systems to prevent frost formation. When handling gases which contain water-soluble, greasy or adhesive contaminants or large concentrations of process dust, air filters may be required in the exhaust system upstream from the heat wheel.

One application of heat wheels is in space heating situations where unusually large quantities of ventilation air are required for health or safety reasons. As many as 20 or 30 air changes per h may be required to remove toxic gases or to prevent the accumulation of explosive mixtures. Comfort heating for that quantity of ventilation air is frequently expensive enough to make the use of heat wheels economical. In the summer season the heat wheel can be used to cool the incoming air from the cold exhaust air, reducing the air conditioning load by as much as 50 percent. It should be pointed out that in many circumstances where large ventilating requirements are mandatory, a better solution than the installation of heat wheels may be the use of local ventilation systems to reduce the hazards and/or the use of infrared comfort heating at principal work areas.

Heat wheels are finding increasing use for process heat recovery in low and moderate temperature environments. Typical applications would be curing or drying ovens and air preheaters in all sizes for industrial and utility boilers.



Building Side (Inside Face)
Upstream Exhaust
(Downstream Supply)

Vertical Cross Section (A-A)
Wheel, Seals and
Purge Section

Weather Side (Outside Face)
Downstream Exhaust
(Upstream Supply)

FIGURE 5.8 *Heat wheel equipped with purge section to clear contaminants from the heat transfer surface.*

Air Preheaters

Passive gas to gas regenerators, sometimes called air preheaters, are available for applications which cannot tolerate any cross contamination. They are constructed of alternate channels (see fig. 5.9) which put the flows of the heating and the heated gases in close contact with each other, separated only by a thin wall of conductive metal. They occupy more volume and are more expensive to construct than are heat wheels, since a much greater heat transfer surface area is required for the same efficiency. An advantage, besides the absence of cross-contamination, is the decreased mechanical complexity since no drive mechanism is required. However, it becomes more difficult to achieve temperature control with the passive regeneration and, if this is a requirement, some of the advantages of its basic simplicity are lost.

Gas-to-gas regenerators are used for recovering heat from exhaust gases to heat other gases in the low to medium temperature range. A list of typical applications follows:

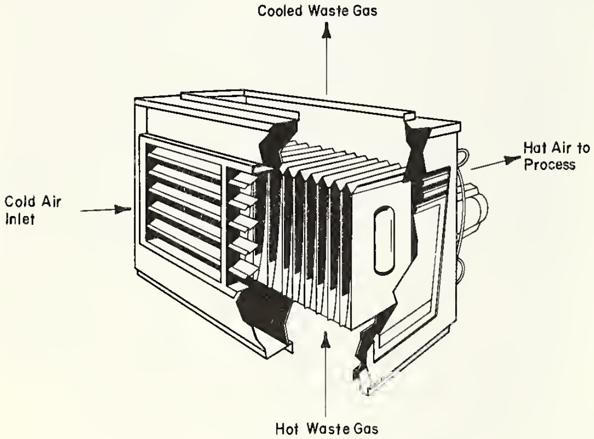


Figure 5.9 *A passive gas to gas regenerator.*

- ° Heat and moisture recovery from building heating and ventilation systems
- ° Heat and moisture recovery from moist rooms and swimming pools
- ° Reduction of building air conditioner loads
- ° Recovery of heat and water from wet industrial processes
- ° Heat recovery from steam boiler exhaust gases
- ° Heat recovery from gas and vapor incinerators
- ° Heat recovery from baking, drying, and curing ovens
- ° Heat recovery from gas turbine exhausts
- ° Heat recovery from other gas-to-gas applications in the low through high temperature range

Heat-Pipe Exchangers

The heat pipe is a heat transfer element that has only recently become commercial, but it shows promise as an industrial waste heat recovery option because of its high efficiency and compact size. In use, it operates as a passive gas-to-gas finned-tube regenerator. As can be seen in figure 5.10, the elements form a bundle of heat pipes which extend through the exhaust and inlet ducts in a pattern that resembles the structured finned coil heat exchangers. Each pipe, however, is a separate sealed element consisting of an annular wick on the inside of the full length of the tube, in which an appropriate heat transfer fluid is entrained. Figure 5.11 shows how the heat absorbed from hot exhaust gases evaporates the entrained fluid, causing the vapor to collect in the center core. The latent heat of vaporization is carried in the vapor to the cold end of the heat pipe located in the cold gas duct. Here the vapor condenses giving up its latent heat. The condensed liquid is then carried by capillary (and/or gravity) action back to the hot end where it is recycled. The heat pipe is compact and efficient because: (1) the finned-tube bundle is inherently a good configuration for convective heat transfer in both gas ducts, and (2) the evaporative-condensing cycle within the heat tubes is a highly efficient way of transferring the heat internally. It is also free from cross contamination. Possible applications include:

- ° Drying, curing and baking ovens
- ° Waste steam reclamation
- ° Air preheaters in steam boilers
- ° Air dryers
- ° Brick kilns (secondary recovery)
- ° Reverberatory furnaces (secondary recovery)
- ° Heating, ventilating and air conditioning systems

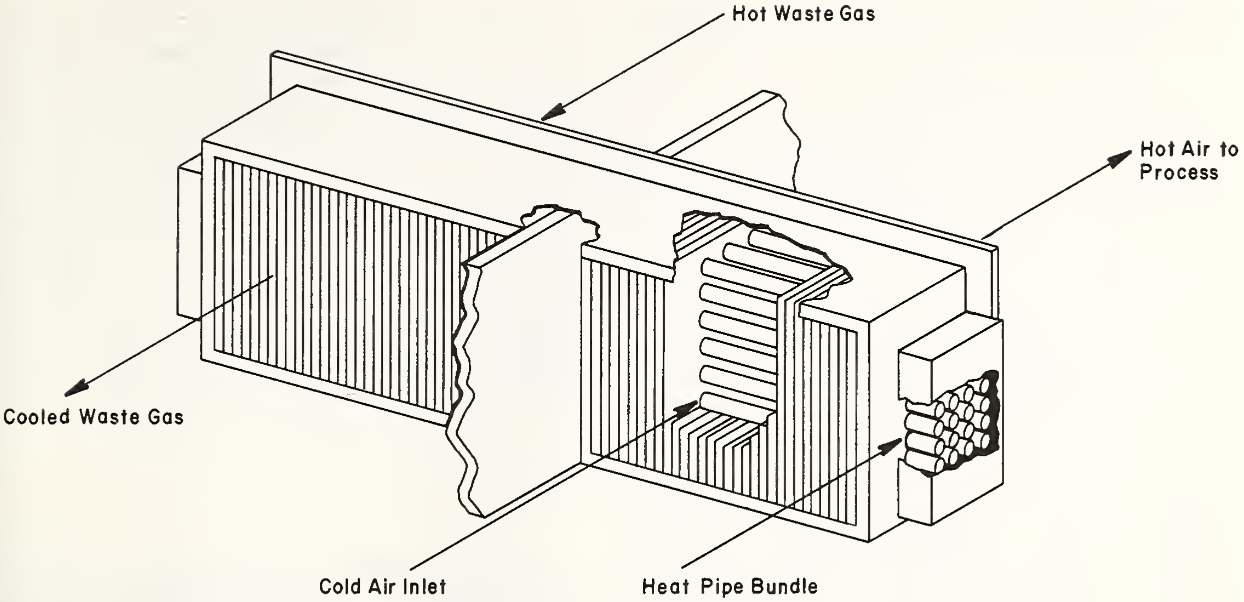


FIGURE 5.10 *Heat pipe bundle incorporated in gas to gas regenerator.*

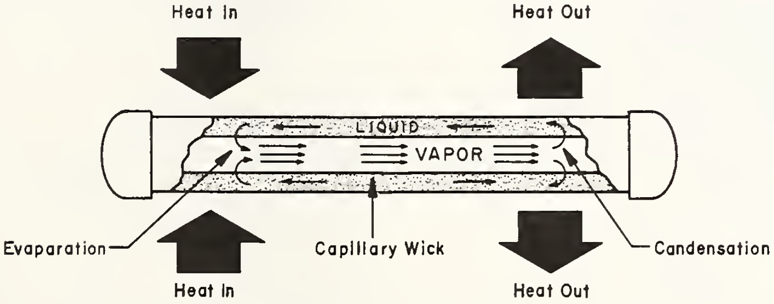


FIGURE 5.11 *Heat pipe schematic.*

Gas or Liquid to Liquid Regenerators

Finned-Tube Heat Exhangers

When waste heat in exhaust gases is recovered for heating liquids for purposes such as providing domestic hot water, heating the feedwater for steam boilers, or for hot water space heating, the finned-tube heat exchanger is generally used. Round tubes are connected together in bundles to contain the heated liquid and fins are welded or otherwise attached to the outside of the tubes to provide additional surface area for removing the waste heat in the gases. Figure 5.12 shows the usual arrangement for the finned-tube exchanger positioned in a duct and details of a typical finned-tube construction. This particular type of application is more commonly known as an economizer. The tubes are often connected all in series but can also be arranged in series-parallel bundles to control the liquid side pressure drop. The air side pressure drop is controlled by the spacing of the tubes and the number of rows of tubes within the duct. Finned-tube exchangers are available prepackaged in modular sizes or can be made up to custom specifications very rapidly from standard components. Temperature control of the heated liquid is usually provided by a bypass duct arrangement which varies the flow rate of hot gases over the heat exchanger. Materials for the tubes and the fins can be selected to withstand corrosive liquids and/or corrosive exhaust gases.

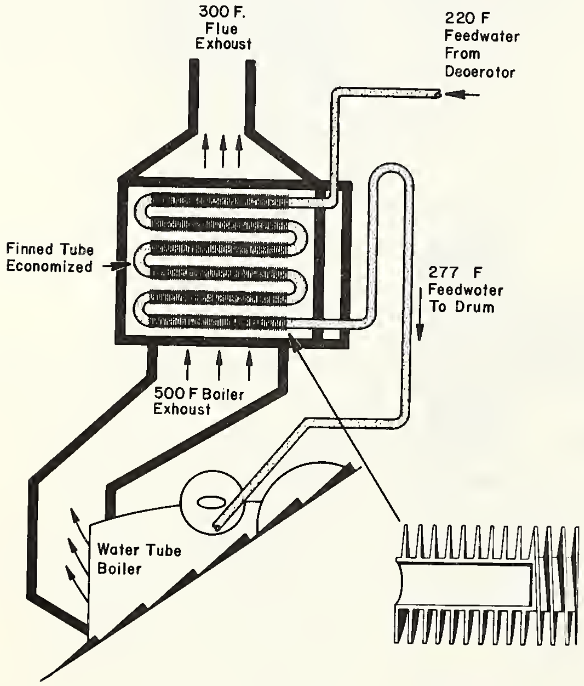


Figure 5.12 *Finned tube gas to liquid regenerator (economizer).*

Finned-tube heat exchangers are used to recover waste heat in the low to medium temperature range from exhaust gases for heating liquids. Typical applications are domestic hot water heating, heating boiler feedwater, hot water space heating, absorption-type refrigeration or air conditioning, and heating process liquids.

Shell and Tube Heat Exchanger

When the medium containing waste heat is a liquid or a vapor which heats another liquid, then the shell and tube heat exchanger must be used since both paths must be sealed to contain the pressures of their respective fluids. The shell contains the tube bundle, and usually internal baffles, to direct the fluid in the shell over the tubes in multiple passes. The shell is inherently weaker than the tubes so that the higher pressure fluid is circulated in the tubes while the lower pressure fluid flows through the shell. When a vapor contains the waste heat, it usually condenses, giving up its latent heat to the liquid being heated. In this application, the vapor is almost invariably contained within the shell. If the reverse is attempted, the condensation of vapors within small diameter parallel tubes causes flow instabilities. Tube and shell heat exchangers are available in a wide range of standard sizes with many combinations of materials for the tubes and shells.

Typical applications of shell and tube heat exchangers include heating liquids with the heat contained by condensates from refrigeration and air conditioning systems; condensate from process steam; coolants from furnace doors, grates, and pipe supports; coolants from engines, air compressors, bearings, and lubricants; and the condensates from distillation processes.

Waste Heat Boilers

Waste heat boilers are ordinarily water tube boilers in which the hot exhaust gases from gas turbines, incinerators, etc., pass over a number of parallel tubes containing water. The water is vaporized in the tubes and collected in a steam drum from which it is drawn off for use as heating or processing steam. Figure 5.13 indicates one arrangement that is used, where the exhaust gases pass over the water tubes twice before they are exhausted to the air. Because the exhaust gases are usually in the medium temperature range and in order to conserve space, a more compact boiler can be produced if the water tubes are finned in order to increase the effective heat transfer area on the gas side. The diagram shows a mud drum, a set of tubes over which the hot gases make a double pass, and a steam drum which collects the steam generated above the water surface. The pressure at which the steam is generated and the rate of steam production depend on the temperature of the hot gases entering the boiler, the flow rate of the hot gases, and the efficiency of the boiler. The pressure of a pure vapor in the presence of its liquid is a function of the temperature of the liquid from which it is evaporated. The abridged steam tables (table 7.2) tabulate this relationship between saturation pressure and temperature. Should the waste heat in the exhaust gases be insufficient for generating the required amount of process steam, it is sometimes possible to add auxiliary burners which burn fuel in the waste heat boiler or to add an afterburner to the exhaust gas duct just ahead of the boiler. Waste heat boilers are built in capacities from less than a thousand to almost a million ft^3/min of exhaust gas.

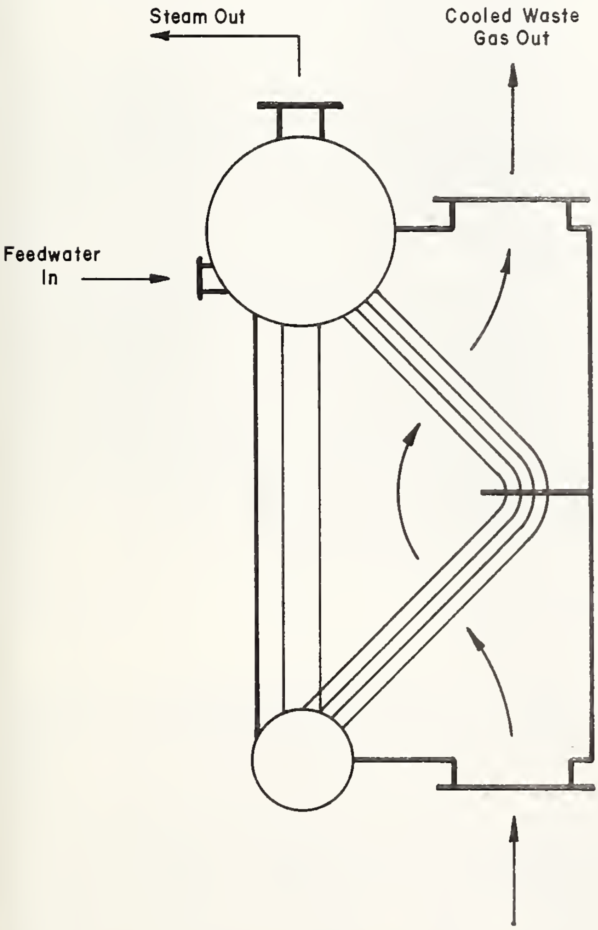


Figure 5.13 *Waste heat boiler for heat recovery from gas turbines or incinerators.*

Typical applications of waste heat boilers are to recover energy from the exhausts of gas turbines, reciprocating engines, incinerators, and furnaces.

Gas and Vapor Expanders

Industrial steam and gas turbines are in an advanced state of development and readily available on a commercial basis. Recently special gas turbine designs for low pressure waste gases have become available; for example, a turbine is available for operation from the top gases of a blast furnace. In this case, as much as 20MW of power could be generated, representing a recovery of 20 to 30 percent of the available energy of the furnace exhaust gas stream. Maximum top pressures are of the order of 40 lb/in^{2}g.

Perhaps of greater applicability than the last example are steam turbines used for producing mechanical work or for driving electrical generators. After removing the necessary energy for doing work, the steam turbine exhausts partially spent steam at a lower pressure than the inlet pressure. The energy in the turbine exhaust stream can then be used for process heat in the usual ways. Steam turbines are classified as back-pressure turbines, available with allowable exit pressure operation above 400 lb/in^{2}g, or condensing turbines which operate below atmospheric exit pressures. The steam used for driving the turbines can be generated in direct fired or waste heat boilers. A list of typical applications for gas and vapor expanders follows:

- Electrical power generation
- Compressor drives
- Pump drives
- Fan drives

Heat Pumps

In the commercial options previously discussed in this chapter, we find waste heat being transferred from a hot fluid to a fluid at a lower temperature. Heat must flow spontaneously "downhill"; that is, from a system at high temperature to one at a lower temperature. This can be expressed scientifically in a number of ways; all the variations of the statement of the second law of thermodynamics. Thermodynamics is reviewed in chapter 2. The practical impact of these statements is that energy as it is transformed again and again and transferred from system to system, becomes less and less available for use. Eventually that energy has such low intensity (resides in a medium at such low temperature) that it is no longer available at all to perform a useful function. It has been taken as a general rule of thumb in industrial operations that fluids with temperatures less than 250 F are of little value for waste heat extraction; flue gases should not be cooled below 250 F (or, better, 300 F to provide a safe margin), because of the risk of condensation of corrosive liquids. However, as fuel costs continue to rise, such waste heat can be used economically for space heating and other low temperature applications. It is possible to reverse the direction of spontaneous energy flow by the use of a thermodynamic system known as a heat pump. This device consists of two heat exchangers, a compressor and an expansion device. Figure 4.11.2 in Case Study 4.11, which discusses an application of the heat pump, shows the arrangement in schematic representation. A liquid or a mixture of liquid and vapor of a pure chemical species flows through the evaporator, where it absorbs heat at low temperature and in doing so is completely vaporized. The low temperature vapor is compressed by a compressor which requires external work. The work done on the vapor raises its pressure and temperature to a level where its energy becomes available for use. The vapor flows through a condenser where it gives up its energy as it condenses to a liquid. The liquid is then expanded through a device back to the evaporator where the cycle repeats. The heat pump was developed as a space heating system where low temperature energy from the ambient air, water, or earth is raised to heating system temperatures by doing compression work with an electric motor driven compressor. The performance of the heat pump is ordinarily described in terms of the coefficient of performance or COP, which is defined as:

$$\mathrm{COP} = \frac{\text{Heat transferred in condenser}}{\text{Compressor work}}$$

which in an ideal heat pump is found as:

$$\mathrm{COP} = \frac{T_H}{T_H - T_L}$$

where T_L is the temperature at which waste heat is extracted from the low temperature medium and T_H is the high temperature at which heat is given up by the pump as useful energy. The coefficient of performance expresses the economy of heat transfer.

In the past, the heat pump has not been applied generally to industrial applications. However, several manufacturers are now redeveloping their domestic heat pump systems as well as new equipment for industrial use. The best applications for the device in this new context are not yet clear, but it may well make possible the use of large quantities of low-grade waste heat with relatively small expenditures of work.

Summary

Table 5.1 presents the collation of a number of significant attributes of the most common types of industrial heat exchangers in matrix form. This matrix allows rapid comparisons to be made in selecting competing types of heat exchangers. The characteristics given in the table for each type of heat exchanger are: allowable temperature range, ability to transfer moisture, ability to withstand large temperature differentials, availability as packaged units, suitability for retrofitting, and compactness and the allowable combinations of heat transfer fluids.

TABLE 5.1

OPERATION AND APPLICATION CHARACTERISTICS OF INDUSTRIAL HEAT EXCHANGERS

SPECIFICATIONS FOR WASTE RECOVERY UNIT / COMMERCIAL HEAT TRANSFER EQUIPMENT	Low Temperature Sub-Zero - 250°F	Intermediate Temp. 250°F - 1200°F	High Temperature 1200°F - 2000°F	Recovers Moisture	Large Temperature Differentials Permitted	Packaged Units Available	Can Be Retrofit	No Cross-Contamination	Compact Size	Gas-to-Gas Heat Exchange	Gas-to-Liquid Heat Exchanger	Liquid-to-Liquid Heat Exchanger	Corrosive Gases Permitted with Special Construction
Radiation Recuperator			●		●	1	●	●		●			●
Convection Recuperator		●	●		●	●	●	●		●			●
Metallic Heat Wheel	●	●		2		●	●	3	●	●			●
Hygroscopic Heat Wheel	●			●		●	●	3	●	●			
Ceramic Heat Wheel		●	●		●	●	●		●	●			●
Passive Regenerator	●	●			●	●	●	●		●			●
Finned-Tube Heat Exchanger	●	●			●	●	●	●	●		●		4
Tube Shell-and-Tube Exchanger	●	●			●	●	●	●	●		●	●	
Waste Heat Boilers	●	●				●	●	●			●		4
Heat Pipes	●	●			5	●	●	●	●	●			●

1. Off-the-shelf items available in small capacities only.

2. Controversial subject. Some authorities claim moisture recovery. Do not advise depending on it.

3. With a purge section added, cross-contamination can be limited to less than 1% by mass.

4. Can be constructed of corrosion-resistant materials, but consider possible extensive damage to equipment caused by leaks or tube ruptures.

5. Allowable temperatures and temperature differential limited by the phase equilibrium properties of the internal fluid.

In regard to moisture recovery, it should be emphasized that many of the heat exchangers operating in the low temperature range may condense vapors from the cooled gas stream. Provisions must be made to remove those liquid condensates from the heat exchanger.

It is possible to list the manufacturers of industrial waste heat recovery equipment with their mail and cable addresses and telephone numbers, and then categorize their products according to type and function. However, the list would unlikely be complete and the information would not stay current. Rather than risk the almost impossible task of keeping up with changes in this presently volatile manufacturing field, it is more practical to compile a list of useful references from the technical periodical literature and from the product directories.

Technical Periodicals

Heating/Piping/Air Conditioning
Reinhold Publishing Corporation
10 S. LaSalle Street
Chicago, IL 60603

Industrial Gas Magazine *
209 Dunn Avenue
Stamford, CT 06905

Industrial Heating
National Industrial Publishing Company
Union Trust Building
Pittsburgh, PA 15219

Plant Engineering
Technical Publishing Company
1301 S. Grove Avenue
Barrington, IL 60010

Heating/Combustion Equipment News
Business Communications, Inc.
2800 Euclid Avenue
Cleveland, OH 44115

Power Magazine
McGraw-Hill Publications Company
New York, NY 10020

Manuals and Product Directories

ASHRAE Handbook and Product Directory—1973, Systems
Published every four years by:
American Society of Heating, Refrigerating, Ventilating & Air Conditioning Engineers, Inc.
345 East 47th Street
New York, NY 10017

ASHRAE Guide and Data Book—1975, Equipment
Published every four years by:
American Society of Heating, Refrigerating, Ventilating & Air Conditioning Engineers, Inc.
345 East 47th Street
New York, NY 10017

Thomas Register of American Manufacturers
Thomas Publishing Company
461 Eighth Avenue
New York,NY 10001

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*One issue published annually as a product directory.

Chapter 6

INSTRUMENTATION

Introduction

In order to make optimum use of waste heat, the engineer or manager needs to select techniques for measuring the essential parameters of the system with which he is dealing. This chapter contains reviews of available technology for such measurements, covering four vital areas: thermometry, flow and pressure measurement, infrared thermography, and flue gas analysis. This chapter is of necessity quite technical, and the reader who is interested in the generalities of waste heat management rather than in technical details may prefer to skip this chapter, with the exception of the section on infrared thermography (page 171), which is in the nature of a semi-technical review.

Some other topics of potential interest to the engineer interested in measurement technology for waste heat management are contained in a special report "Key Measurements in Power and Process" in Power Magazine, September 1975.

MEASUREMENT OF GAS FLOW RATES

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This section discusses some of the methods that may be used to measure gas flow rates in conduits. An accurate knowledge of gas flow rates is important since the amount of energy available for recovery from an exhaust stack depends not only on the temperature of the gas but also on the mass flow rate of the gas. Also, if a cooler gas flow is receiving energy from a warmer gas flow, its temperature rise will depend on the mass flow rates.

Generally, the methods described in this section, if carefully applied, can give results to about 5 percent accuracy using commercially available instruments. As will be made clear, all methods depend on accurate knowledge of gas temperatures; at higher temperatures errors in gas temperature measurement will be the greatest source of error, for not only does the temperature enter all measurements, but also the measurement of temperature in a flowing gas is subject to many errors.

Analysis of flow in closed circuits such as exhaust stacks, ventilation air ducts, inlet and exhaust ducts to furnaces, heat exchangers, etc., is based on the physical law of conservation of mass or the equation of "continuity." Consider flow in a conduit as shown in figure 6.1. If the flow is steady, and there are no leaks in the conduit, the mass flow rate at section 1 is equal to that at section 2, or

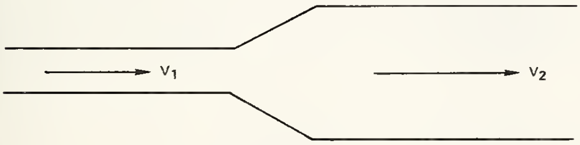

FIGURE 6.1 *Flow in a conduit.*

$$\dot{m} = \rho_1 A_1 V_1 = \rho_2 V_2 A_2 \tag{6.1}$$

where

ρ = Density (lbm/ft^3)
A = Cross-sectional area (ft^2)
V = Average velocity at section A (ft/s)

The volume flow rate (ft^3/s) is defined as

$$Q = AV \tag{6.2}$$

then

$$\dot{m} = \rho_1 Q_1 = \rho_2 Q_2 \tag{6.3}$$

the density at any section cannot be measured directly but is calculated using the perfect gas equation

$$\rho = \frac{P_a}{RT} \tag{6.4}$$

where

P_a = Absolute pressure (lbf/ft^2)
T = Absolute temperature (degrees Rankine = degrees Fahrenheit + 459.2)
R = Gas constant = $\bar{R}/M$
$\bar{R}$ = Universal gas constant = 1545 (ft lbf/lb · mol R)
M = Molecular weight of gas

Further, if the flow is incompressible, or $\rho_1 = \rho_2$, then $Q_1 = Q_2$, or

$$A_1 V_1 = A_2 V_2 \tag{6.5}$$

Often, to complete a flow analysis, as will be shown later, the pressure in the conduit must be known. At any section of a conduit one can write

$$P_T = P_S + P_V \tag{6.6}$$

where

P_T = Total or impact pressure (lbf/ft^2)
P_S = Static pressure (lbf/ft^2)
P_V = Velocity pressure (lbf/ft^2)

The static pressure is that which tends to expand the conduit and also moves the gas along in the conduit. The velocity pressure is the pressure arising from the kinetic energy of the moving gas.

The relation between P_V and V, the gas velocity, is

$$P_V = \frac{V^2}{2g_c} \tag{6.7}$$

$$g_c = 32.174 \ \frac{\text{ft}\cdot\text{lbm}}{\text{lbf}\cdot\text{s}^2}\text{, the gravitational constant}$$

Thus, if P_V is known (or measured) V can be determined and m calculated from eq (6.1).

Often velocity pressure is expressed in "inches of water" which is the height of water in a vertical tube that will exert a pressure on its closed bottom end equal to P_V. This height, often called "head," is obtained by first dividing eq (6.7) by ρ to get

$$h_g = \frac{P_V}{\rho} = \frac{V^2}{2g_c} \tag{6.8}$$

where h_g = head of gas flowing (feet), and then

$$h_g = \frac{\rho}{\rho_w} \tag{6.9}$$

where h_w is the head of water in inches, ρ_w is the density of water, and ρ is the density of the gas.

Although there are devices called quantity meters that measure gas flow rates directly, they are not suitable for the applications indicated above. Rather, devices called rate meters are used, which indirectly measure flow rates. Usually, velocity pressure or a pressure difference is measured, which then can be converted to velocity and used with eq (6.1) or eq (6.2). Simultaneously gas temperature must be measured so that ρ may be calculated from eq (6.4). Two indirect methods, the velocity traverse and the sharp-edge orifice, will be discussed in detail, and others will be mentioned.

The velocity in eq (6.1) is the average velocity across any section of a conduit. However, the velocity across any section is not uniform and so measurements are made at various points across the section to determine local velocities, and the local velocities are averaged to determine the average velocity. This measurement technique is called a velocity traverse. The local measurements are made with a pitot-static tube in combination with a manometer, as indicated in figure 6.2. The manometer is simply a device that contains an easily observable liquid in a glass tube. The total or impact pressure of the gas at the point of measurement works on one end of the liquid column in the glass tube, and the static pressure on the other end. The difference in liquid levels, according to eq (6.6), is an indication of the velocity pressure, i.e.,

$$P_V = P_T - P_S\,. \tag{6.10}$$

Commercially available manometers are usually calibrated (through eqs similar to (6.8) and 6.9)) to read directly in inches of water. Combining eqs (6.8) and (6.9) yields

$$v_i = 18.28 \ \frac{h_v}{\rho_i} \tag{6.11}$$

h_v = Velocity pressure directly from manometer (in water)

ρ_i = Actual local density of gas (lbm/ft^3)

v_i = Local gas velocity (ft/s).

Two important considerations must be made when taking velocity traverses in a conduit; location of the traverse relative to conduit fittings and the number and location of traverse points. Any conduit fitting such as a bend, transition section, damper, etc., disturbs the flow, making it difficult to obtain a good traverse. The more pronounced the disturbance, the greater the number of traverse points required. Reference 2 recommends locations of traverse sites relative to upstream and downstream disturbances, and the recommended minimum number of traverse points to be used.

For rectangular conduits the cross section is divided into as many equal rectangular areas (or squares for a square conduit) as traverse points. In any case the ratio of the width to the length of the subareas should be between one and two. The reading of velocity pressure is taken at the center of each subarea.

After the reading of velocity pressure is taken at each traverse point, the velocity at each point is calculated using eq (6.11). Density is calculated

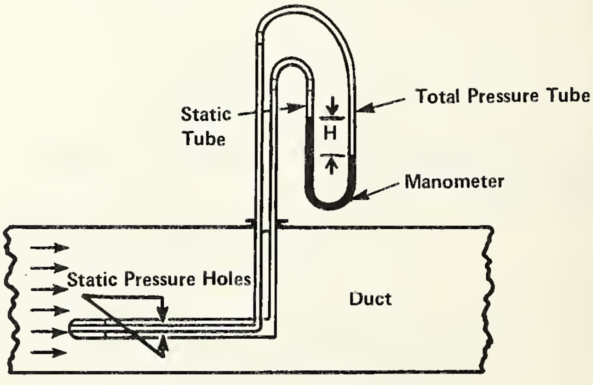

FIGURE 6.2 *Pitot-static tube in a duct.*

with eq (6.4). To find P_a in eq (6.4), P_T is found by removing the static pressure tube from the manometer and reading P_T alone. Then $P_S = (P_T - P_V)$. The atmospheric pressure obtained with a barometer located in the vicinity of the conduit is added to P_S to obtain P_a. In adding these two values, one must be sure that the units are the same.

If temperature varies across the cross section of the conduit, it must be measured at each traverse point, as it is required also for calculating P_i. If not, a one point reading close to the center of the conduit is sufficient. Suitable techniques for such temperature measurements are discussed in the following section.

When the velocity at each point in the traverse is properly determined, all velocities are added together and the sum divided by the number traverse points to obtain V, the average velocity. Then m is calculated using eq (6.1) with A the total area of the traversed section. If strong temperature variations occur across the section, ρ is the average value for the section.

In some cases very large conduits must be traversed, and pitot-static tubes of sufficient length to span the section will not be available. Possibly large circular conduits can be traversed on a diameter by going in from both sides. Also, a pitot-static tube mounted on a moving traverse mechanism spanning the section can be used. Large rectangular conduits may also be traversed using moving mechanism; if room is available around the rectangular conduit separate traverses can be made in quarter sections and the results combined.

When taking a reading at any traverse point, the pitot-static tube must be placed so that the tip points upstream and as parallel to the conduit centerline as possible. Errors are encountered when the tip is at an angle to the flow. One must be careful that the hose connections between the pitot-static tube and manometer are tight and leak proof. At no time should the hoses be "crimped," cutting off pressure to the manometer. The hoses should be inspected from time to time to make sure that they are free from excessive wear or small holes. Only manometer fluids recommended by the manometer manufacturer should be used, since manometers are usually calibrated for one particular fluid. Also, all instructions for use of the manometer as given by the manufacturer should be strictly followed.

Manometers are available in a variety of configurations and pressure ranges. The U-tube manometer has the largest range and will go to 10 or more inches of water. Inclined manometers may go to 10 in of water but are usually used in ranges up to 0.5, 1.0, and 2.0 in of water. Micromanometers or "Hooke gages" are used for small pressures or pressure differences and have a readability of thousandths of an inch of water. For a particular application, sufficient thought should be given to what range of pressures will be measured, and to the necessary readability, accuracy and precision. Before buying instruments, some of which may be very expensive, the advice of the instrument manufacturer should be sought.

The sharp edge orifice, sometimes classified as a differential type flow-meter, is an inexpensive device and easy to install and replace. It has been fairly well established that sharp edge orifices give accurate results when used in circular conduits. Application to noncircular conduits is not recommended. A typical installation consists of a thin plate with an orifice hole concentric with the conduit. The width of the cylindrical edge of the orifice hole should be between 0.01 and 0.02 times the diameter D, of the conduit. For thicker plates, the outlet edge should be beveled. The diameter ratio

$$\beta = \frac{d}{D} \tag{6.12}$$

should be between 0.25 and 0.70. A differential pressure gage, usually a manometer, measures the pressure difference across the orifice plate. As is shown below, this pressure difference is used to calculate the gas flow rate, m, in the conduit.

It can be shown that for the commonest sharp edge orifice, the so-called "1-d and 1/2-d tap" orifice, as discussed in reference (3)

$$m = 0.0997\ KYd^2\ \rho_1\ (h_1 - h_2) \tag{6.13}$$

where

$m =$ Mass flow rate of gas (lbm/s)
$d =$ Orifice diameter (in)
$h =$ Pressure (in of water)
$\rho_1 =$ Density of gas at point 1 (lbmft^3)

The constants K and Y are called the flow coefficient and the expansion factor respectively, and are also tabulated in reference (3).

Other differential type flow meters, considerably more elaborate in design than the sharp edge orifice, and thus relatively expensive, are the venturi meter (3), the Instrument Society of America (ISA) flow nozzle (3), and the American Society of Mechanical Engineers (ASME) long-radius flow nozzle (3). Pressure drops across the ISA nozzle and ASME nozzle are about the same as or slightly less than that across the sharp edge orifice, but considerably greater than that through the venturi meter.

For purposes of measuring stack gas volumetric flow rates, EPA recommends the type-S (stauscheibe or reverse type) pitot tube. Figure 6.3 shows the type-S pitot tube and its application is discussed in reference (2). Proper application requires a stack gas analysis to determine the molecular weight of the gas, as well as a pitot-static tube to calibrate the type-S pitot tube.

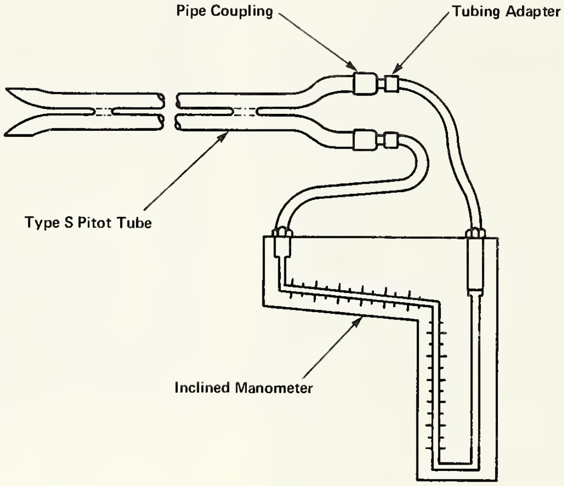

FIGURE 6.3 *Type S pitot tube manometer assembly.*

Page 161 Author credits should be omitted

THERMOMETRY

In this section, commercially available temperature measuring devices and suggested measurement techniques are presented. An accurate temperature determination of a gas stream in a conduit or exhaust stack is vital to making a meaningful waste heat survey or to monitor or control the performance of waste heat equipment.

The energy available for waste heat recovery is dependent on the mass flow rate, which was discussed in the preceding section, and on the temperature of the gas stream. The determination of the mass flow rate depends significantly on the gas density which in turn is a function of the gas temperature.

Temperature is also a meaningful parameter to monitor or control the operation of waste heat equipment. Temperatures may be monitored by observing a meter or scale reading or by recording a time history of the temperature. In some applications, measured temperatures can be used in conjunction with a feed-back system to automatically adjust the operating conditions.

The measurement of temperature of a gas flowing in a conduit or exhaust stack is complicated by the existence of a profile across the flow channel. In general, the gas temperature is not constant at a given cross section but has a profile which is similar in shape to the velocity profiles discussed in the preceding section. Disturbances which distort the velocity profile will also influence the temperature profile. Therefore, the same precautions that were outlined in the preceding section for measuring the flow velocity at a cross section with reference to disturbances in the flow should also be observed in measuring temperatures.

Several commercially available temperature instruments will be discussed in the remainder of this section. However, it is felt that the thermocouple represents the most suitable choice because of its versatility, ease of operation, accuracy, and availability. For this reason, a significant part of this section is devoted to this type of sensor.

The principle of thermoelectricity, discovered by Seebeck in 1821, is the basis for one of the most commonly used temperature sensing devices—the thermocouple. Seebeck discovered that if a circuit is formed consisting of two dissimilar metallic conductors A and B, and if one of the junctions of A and B is a temperature T_1 while the other junction is at a higher temperature T_2, a current will flow in the circuit. A current continues to flow as long as the two junctions are at differing temperatures. The emf producing this current is called the Seebeck thermal emf. Conductor A is said to be positive with respect to B if the current flows from A to B at the cooler ($T_1 < T_2$) of the two junctions.

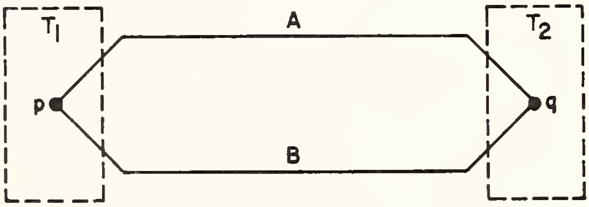

FIGURE 6.4 *Elementary thermocouple circuit.*

Later it was shown that the potential actually comes from two different sources: that resulting solely from contact of the two dissimilar metals and the junction temperature, and that due to temperature gradients along the conductors in the circuit. These two effects are named the Peltier and Thompson effects after their respective discoverers. In most cases with proper selection of materials the Thomson emf is quite small relative to the Peltier emf and may be disregarded.

In addition to the Seebeck effect, there are certain laws which thermoelectric circuits abide by:

Law of intermediate metals. Insertion of an intermediate metal into a thermocouple circuit will not affect the net emf, provided the two junctions introduced by the third metal are at identical temperatures. If the third metal C is introduced and if the new junction *r* and *s* are both held at

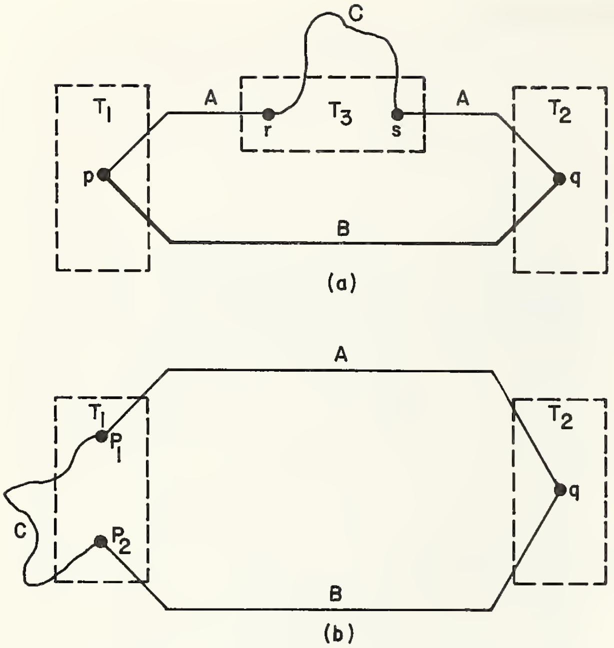

FIGURE 6.5 *Diagrams illustrating the law for intermediate metals.*

temperature T_3, the net potential for the circuit will remain unchanged. This permits insertion of a measuring device or circuit, without upsetting the temperature function and the thermocouple circuit; the third metal C may be introduced at either a measuring or reference junction, as long as couples P_1 and P_2 are maintained at the same temperature.

Law of intermediate temperatures. If a simple thermocouple circuit develops an emf, e_1, when its junctions are at temperatures T_1 and T_2, and an emf, e_2, when its junctions are at temperatures T_2 and T_3, it will develop an emf, $e_1 + e_2$, when its junctions are at temperatures T_1 and T_3. This makes possible direct correction for secondary junctions whose temperatures may be known but are not directly controllable.

Theoretically, any two unlike conducting materials could be used to form a thermocouple. Actually, of course, certain materials and combinations are better than others and some have practically become standard for given temperature ranges. Materials commonly used in thermocouples include copper, iron, platinum, rhodium, iridium, constantan (60% copper, 40% nickel), chromel (10% chromium, 90% nickel), and alumel (2% aluminum, 90% nickel, remainder silicon and manganese).

Combinations of the above materials and their characteristics are listed in table 6.3. Size of wire is of some importance. Usually the higher the temperature to be measured, the heavier should be the wire. As the size is increased, however, the time response of the couple to temperature change increases. Therefore, some compromise between response and life may be required.

Essentially, a thermoelectric thermometer or thermocouple consist of two wires, A and B joined together to form the "hot" or measuring junction, and a device for measuring emf connected to the free ends of A and B. Thermocouples may be prepared by twisting the two wires together and brazing or preferably, welding, shown in figure 6.6.

TABLE 6.1. *Characteristics of commercial thermocouples of various materials*

Type	Temperature limits, F	Comments
Copper-constantan	−300 to 650	Inexpensive; high output; copper oxidizes above 650° F.
Chromel-constantan	0 to 1000	Highest output of couples; good stability.
Iron-constantan	0 to 1500	Inexpensive; high output; iron oxidizes rapidly above 1500° F; temperature gradients along leads should be avoided to minimize Thomson emf's.
Chromel/alumel	600 to 2000 (continuous) Up to 2300 (intermittent)	Very resistant to oxidation within specified temperature ranges; emf tends to reduce with use.
Platinum-10% rhodium	1300 to 2700 (continuous) Up to 2850 (intermittent)	Expensive, with low output, hence should be used only for high-temperature applications; very stable and resistant to oxidation; specified for reproducing International Temperature Scale between 1166 and 1945° F.
Platinum-6% rhodium	Up to 3100	Not common.
Platinum-30% rhodium		
Iridium, iridium-rhodium	Up to 3500+	Not common.

The reference temperature T_r should be known. However, in certain commercial equipment, electrical or electromechanical compensation is sometimes built in. This compensation can also be accomplished manually if the T_r is known, as in figure 6.7, where the reference temperature is read by means of a thermometer and the necessary corrections made.

For a more accurately controlled temperature, however, it is common to have the reference junction at the ice point. This ice slurry is made using ice made from distilled water to eliminate contaminants that tend to shift the freezing point. Figure 6.8 shows such an arrangement.

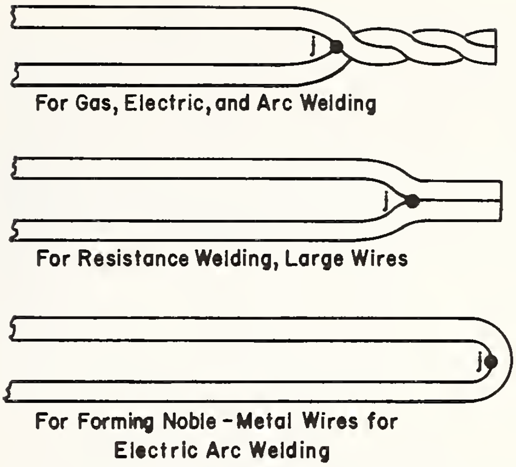

FIGURE 6.6 *Common forms of thermocouple construction.*

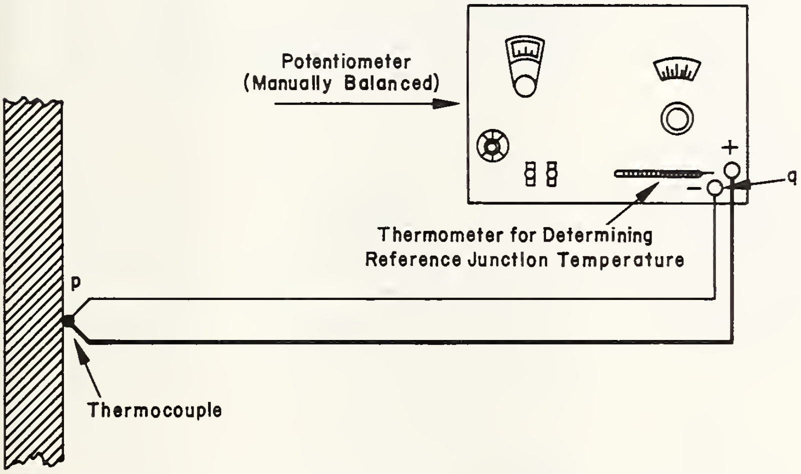

FIGURE 6.7 *Schematic diagram illustrating use of potentiometer terminals as a reference junction.*

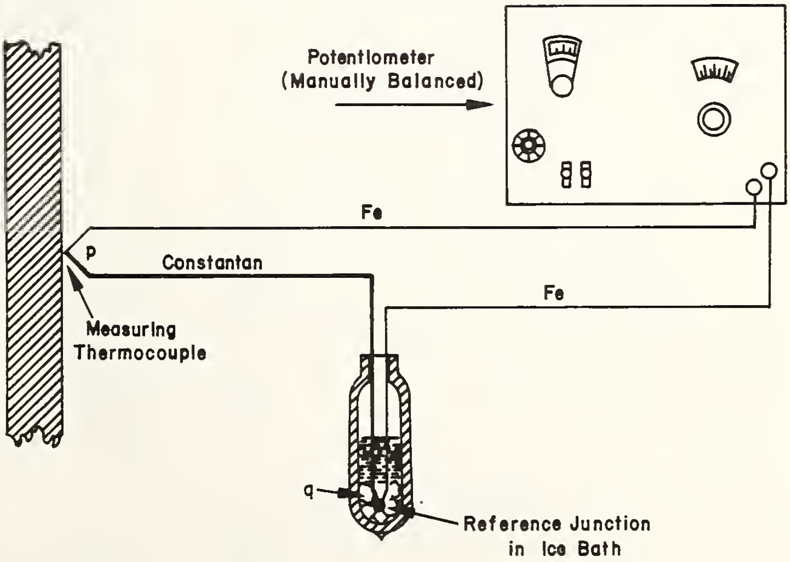

FIGURE 6.8 *System with fixed reference temperature (ice bath).*

Measurement of Thermocouple emf's

The basic fact of thermoelectric thermometry is that a thermocouple develops an emf which is a definite function of the difference in temperature of its hot and cold junctions. If the temperature of one junction is known, the temperature of the other junction can be determined by measuring the emf generated in the circuit. Measurement of temperature with the aid of a thermocouple, therefore, requires the use of an instrument capable of measuring emf. In practice, two methods of measurement of emf are in use, the millivoltmeter and the potentiometer.

The millivoltmeter consists of a galvanometer with a rigid pointer which moves over a scale graduated in millivolts or in degrees, F or C. The galvanometer indicates by its deflection the magnitude of the current passing through it, and if the circuit in which it is placed includes a thermocouple it measures the current, I, generated by the thermocouple in the circuit. If the circuit has a resistance, R, and emf is E, by Ohm's Law, $E = RI$, and if R is kept constant E is proportional to I and the scale can be calibrated in terms of millivolts rather than in milliamperes or microamperes. This calibration holds good only as long as R remains constant. Any change in R introduces an error in the indicated value of E.

Where precision is required in the measurement of thermal emfs, the potentiometer is invariably used. And because of its reliability and freedom from the uncertainties arising from changes in circuit resistance, meter calibration, etc., as well as its much greater openness of scale, it is used in many cases where the precision needed might not seem to justify the higher cost.

Thermocouples may also be connected in series, the combination is sometimes called a thermopile, shown in figure 6.9. The total output from thermocouples connected to form a thermopile will be equal to the sums of the individual emfs and if the thermocouples are identical, the total output will equal N times the output of a single couple. This arrangement is used for measuring average temperature and not for obtaining temperature distributions. Care must be used, for a short circuit may escape detection, giving an erroneous reading.

Figure 6.10 shows a parallel arrangement, for sensing average temperature levels. One observation gives, by calibration, the arithmetic mean of the temperatures sensed by N individual measuring junctions. This arrangement gives no indication of temperature distribution either.

When determining heat balances in industrial environments, for example the temperature of the gas or air streams flowing through flues and stacks, thermocouples play a very important role. We must, however, bear in mind that the measurement of the temperature of a gas is considerably different from such a measurement in liquids and solids. The principal problem is the highly practical one of causing the measuring junction to attain the same temperature as the medium in which it is immersed, or a temperature which bears a known relation to that of the medium.

In measuring the temperature of flue gases the thermocouple will be inclosed in a metal protecting tube which is screwed into an opening in the wall of the duct. The inner surface of the metal wall is exposed to temperatures of 500 to 1500 F while the outside temperature may be from 50 to 100 F.

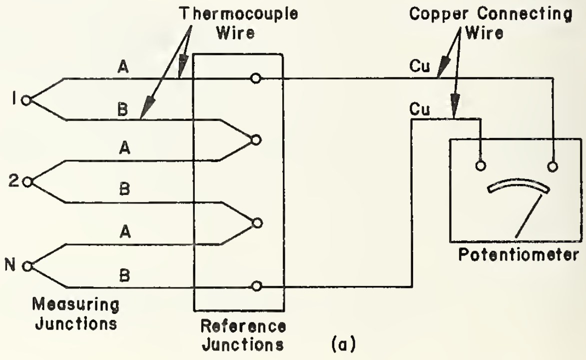

FIGURE 6.9 *Thermopile for sensing average temperature level.*

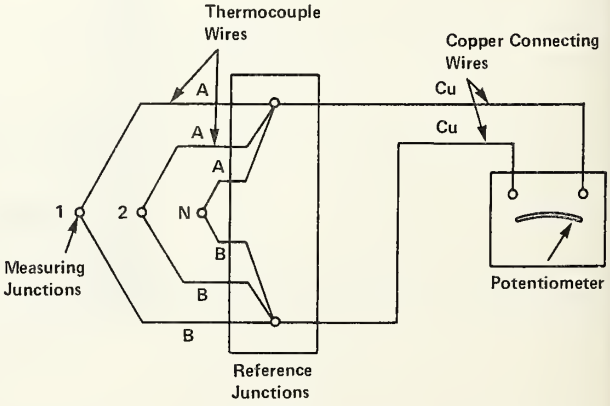

FIGURE 6.10 *Parallel arrangement for sensing average temperature levels.*

Therefore, there is a loss of heat through the walls, and the inner surface is at a temperature considerably lower than that of the flue gas. The protecting tube then is cooler where it enters the wall than at its tip where the thermocouple junction is placed so that a flow of heat takes place through the tube and the thermocouple wires from the tip of the tube towards the wall. This heat must be supplied from the gas stream, and to pick up the heat the tube must be cooler than the gas. The temperature of the junction must, therefore, be lower than that of the gas. The magnitude of the resulting error will depend upon the heat transfer coefficient, which involves the rate of flow of the gas, the nature of the portion of the surface of the protecting tube exposed to the gas, the length, diameter and materials of the tube and the thermocouple wires, and the temperature of wall to which the tube is attached. The errors are often significant.

A further source of error is that resulting from radiation. In the case under consideration, of a hot gas inside cooler walls, there will be a loss of heat from the tube or thermocouple junction by radiation to the walls, in addition to that by conduction through tube and wires. Experience has shown that a thermocouple located in the hot gases, where it can "see" the cooler surfaces of the boiler wall may indicate temperature 150 F too low when the actual gas temperature is 600 F. The radiation error decreases as the area of the radiating surface of the junction or tube is decreased. Consequently, this area should be made as small as is consistent with mechanical strength. The errors from conduction and radiation can be decreased by inducing a high velocity of flow past the sensitive element. This increases the convective transfer of heat to the element by providing a greater volume of hot gas, and also by stripping off the stagnant layer of adsorbed gas which clings to the thermocouple and forms a heat insulating covering. This is the principle of the suction or aspiration pyrometer. The radiation errors may be reduced by enclosing the sensitive element in a radiation shield, which may consist of two or more concentric cylinders of materials of low emissivity, which prevent the sensitive element from "seeing" any cooler surface, but permit a free circulation of gas. The radiation error may be eliminated if the shields are provided with independent sources of heat and each is provided with a thermocouple. By adjusting the auxiliary heating the thermocouples may all be made to read alike, and there can then be no loss of heat by radiation (or otherwise) between the shields or between shield and sensitive element.

If the purpose of the measurement of the temperature of the gas is to work out a "heat balance" in a test of the efficiency of a power plant it is not sufficient to measure the temperature at one point in the gas, since the flow is invariably more or less stratified. It is not sufficient to average the temperature simultaneously from a number of points since the velocity of flow and consequently the mass of gas represented by each measurement is variable from point to point. It is necessary, by careful testing, to evaluate the rate of flow at the various points and to weight the temperature readings according to the flow. And this distribution of gas velocities is subject to change with each change of boiler rating.

Traversing in equal increments across two diameters at right angles will give you the temperature profile of the stack, as shown in figure 6.11(a). A similar profile can be obtained for a rectangular stack as shown in figure 6.11(b). Remember accurate "point" measurements do not necessarily ensure accurate "average" values. For further mention of this method see the preceding discussion of pressure measurement.

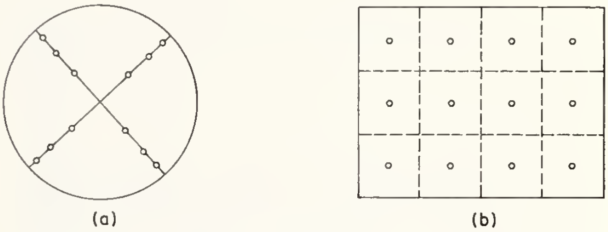

FIGURE 6.11 *Placement of thermocouple for measuring temperature profiles, in circular (a) and rectangular (b) stacks.*

We shall now consider the application of thermometry to waste heat recovery.

The heat transfer rate to a gas flow can be represented by

$$Q = \dot{m}\, c_p (T_2 - T_1) \qquad (6.14)$$

where

Q = Heat transfer rate (Btu/s)
$\dot{m}$ = Mass flow rate (lbm/s)
c_p = Specific heat of the gas at constant presure (Btu/lbm•F)

T_1 = Gas inlet temperature (F)
T_2 = Gas outlet temperature (F)

In the eq (6.14), the inlet and outlet temperatures are the average gas temperatures at the particular cross section. Since the temperature across a section is not uniform, measurements should be made at several points (similar to the velocity traverse method) across the section and these averaged to determine the average gas temperature.

A second consideration is the magnitude of the gas velocity. At high gas velocities, the stagnation temperature at the inlet and exit should be used in eq (6.15) to predict the heat transfer rates. The stagnation temperature is defined such that

$$T_0 = T + \frac{V^2}{2g_c J\,Cp} \qquad (6.15)$$

where

T_0 = Stagnation temperature (f)
T = Static temperature (F)
V = Gas velocity (ft/s)
g_c = Gravitational Constant (32,174 ft-lbm/lbf. s^2)
J = Joule's Constant (778 ft-lbf/Btu)
c_p = Specific heat of the gas at constant pressure (Btu/lbm-F)

The stagnation temperature is a measure of both the static thermal energy and the flowing kinetic energy. At low gas velocities, a static temperature probe can be used without significant error. At high gas velocities, a stagnation temperature probe should be used. A stagnation temperature probe brings the flow to rest with a minimum loss of energy, thus reclaiming the kinetic energy to the gas flow.

Other types of thermometry used in waste heat measurement include pressure bulb thermometry, pyrometry, and infrared thermography. Pressure bulb thermometry and pyrometry will be discussed briefly here, infrared thermography in more detail later in the chapter.

The pressure thermometer is generally a liquid—or vapor—filled well, A, connected by pressure tubing, B, to one of several pressure-activated devices, C, as shown in figure 6.12. The well may contain mercury, in which case the capillary tubing and pressure sensitive helix or spiral gage all will likely contain mercury. The expansion and contraction of the mercury in the bulb in response to temperature rise and fall is thus communicated as pressure change in the helix or spiral gage, which can then record or activate a mechanism as the application requires. Since the volume of mercury in the gage may be an appreciable fraction of that in the well (or "bulb"), it is clearly necessary for the indication/control gage to be maintained at fairly constant values of pressure

TABLE 6.2. *Constant pressure specific heats of gases at zero pressure*

Gas or vapor	Equation, $\bar{C}_{po}$ in Btu/lb mol R T in degrees Rankine	Range, R	Max. error, %
O_2	$\bar{C}_{po} = 11.515 - \frac{172}{\sqrt{T}} + \frac{1530}{T}$	540–5000	1.1
	$= 11.515 - \frac{172}{\sqrt{T}} + \frac{1530}{T}$		
	$+ \frac{0.05}{1000}(T - 4000)$	5000–9000	0.3
N_2	$\bar{C}_{po} = 9.47 - \frac{3.47 \times 10^3}{T}$		
	$+ \frac{1.16 \times 10^6}{T^2}$	540–9000	1.7
CO	$\bar{C}_{po} = 9.46 - \frac{3.29 \times 10^3}{T}$		
	$+ \frac{1.07 \times 10^6}{T^2}$	540–9000	1.1
H_2	$\bar{C}_{po} = 5.70 + \frac{0.578}{1000}T + \frac{20}{\sqrt{T}}$	540–4000	0.8
	$= 5.76 + \frac{0.578}{1000}T + \frac{20}{\sqrt{T}}$		
	$- \frac{0.33}{1000}(T - 4000)$	4000–9000	1.4
H_2O	$C_{po} = 19.86 - \frac{597}{\sqrt{T}} + \frac{7500}{T}$	540–5400	1.8
CO_2	$\bar{C}_{po} = 16.2 - \frac{6.53 \times 10^3}{T}$		
	$+ \frac{1.41 \times 10^6}{T^2}$	540–6300	0.8
CH_4	$\bar{C}_{po} = 4.52 + 0.00737T$	540–1500	1.2
C_2H_4	$\bar{C}_{po} = 4.23 + 0.01177T$	350–1100	1.5
C_2H_6	$\bar{C}_{po} = 4.01 + 0.01636T$	400–1100	1.5
C_6H_{16}	$\bar{C}_{po} = 7.92 + 0.0601T$	400–1100	est. 4
$C_{12}H_{26}$	$\bar{C}_{po} = 8.68 + 0.0889T$	400–1100	est. 4

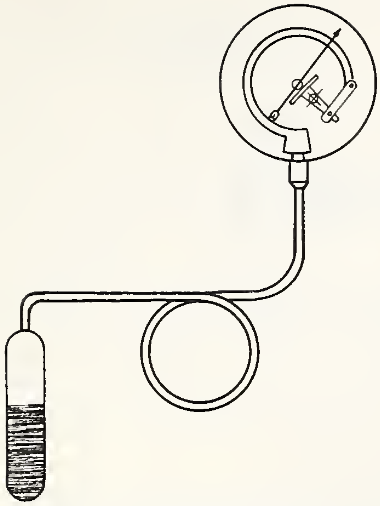

FIGURE 6.12 *Principle of operation of a pressure thermometer.*

TABLE 6.3. *Temperature limitation for pressure bulb*

Type of pressure	Approx. overall range of usefulness, F	Max approx. range for a given instrument, F
Liquid filled	−150 to 1000 (depending on liquid)	400 to 800 (depending on liquid)
Gas filled	−400 to 1000	800
Vapor filled	−300 to 700 (depending on vapor)	250

and temperature. In other forms of the pressure bulb, the thermometer bulb may be filled with an evaporable liquid, the principle of use is the same as for mercury. The liquid-vapor system has an advantage over the other systems in that the vapor pressure is set by the bulb temperature, and consequently changes in temperature within the capillary gage system are unimportant.

The term pyrometry is derived from the Greek words *pyros* meaning "fire" and *metron*, "to measure." Pyrometry normally refers to the measurement of temperatures in the range extending upward from about 1000 F.

There are two distinct instruments referred to as pyrometers. They are the total-radiation pyrometer and the optical pyrometer. The first, as the name implies, accepts a controlled sample of total radiation and, through determination of the heating effect of the sample, obtains a measure for temperature. As the name implies, the optical pyrometer employs optical means for estimating the change in average wave length and flux of visual radiation with temperature. Neither instrument, however, is dependent upon direct contact with the source and, within reason, neither is dependent on distance from the source. Neither is suited to measuring gas temperatures in industrial applications.

All bodies above absolute zero temperature radiate energy. Not only do they radiate or emit energy, but they also receive and absorb it from other sources. A piece of metal (steel say) when heated to about 1000 F begins to glow; as the temperature is raised the light becomes brighter and more intense. In addition, there is a change in color; it changes from a dull red, through orange to yellow, and finally approaches an almost white light at the melting temperature (2600 to 2800 F).

This energy which we can feel (even though we may not actually be in physical contact) as well as see is transmitted as electromagnetic waves traveling at the speed of light. All substances emit and absorb radiant energy at a rate depending on the absolute temperature and physical properties of the substance. Waves striking the surface of a substance are partially absorbed, partially reflected, and partially transmitted. These portions are measured in terms of absorptivity α, reflectivity ρ, and transmissivity τ, where

$$\alpha + \rho + \tau = 1$$

For an ideal reflector, a condition approached by a highly polished surface, $\rho \to 1$. Many gases represent substances of high transmissivity, for which $\tau \to 1$, and a black body approaches the ideal absorber, for which $\alpha \to 1$. A good absorber is also a good radiator, and an ideal radiator is one for which $\alpha = 1$. When referring to radiation as distinguished from absorption, the emissivity ϵ, is used rather than absorptivity α.

Figure 6.13 shows, in simplified form, the method of operation of the total-radiation pyrometer.

The essential parts of the device consist of some form of directing means, shown here as baffles but which is more often a lens, and an approximate black body receiver with means for sensing temperature. A balance is quickly established between the energy absorbed by the receiver and that dissipated by emission and by conduction through leads. The receiver equilibrium temperature then becomes the measure of source temperature, with the scale established by calibration.

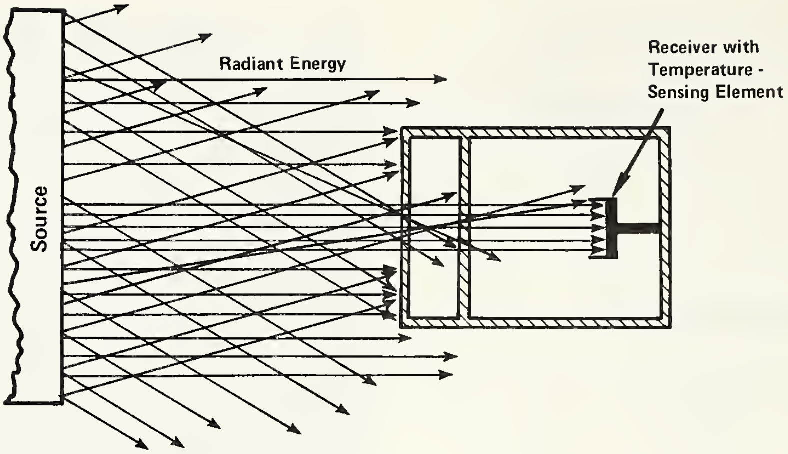

FIGURE 6.13 *A simplified form of total-radiation pyrometer.*

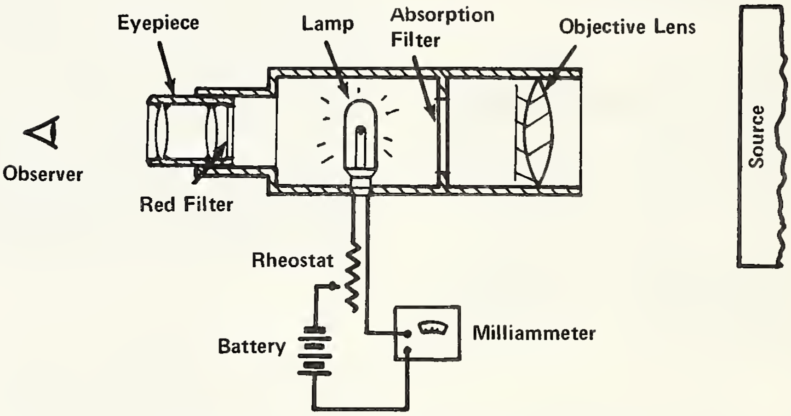

FIGURE 6.14 *Schematic diagram of an optical pyrometer.*

In a typical commercially available pyrometer sensitive to very low level radiation, a lens and mirror system is used to focus the radiant energy on a thermopile, whose output is measured by a voltage-balancing potentiometer. These instruments are used ideally in applications where the sources approach black body condition, that is, where the source has an emissivity ϵ approaching unity.

The static accuracy of the radiation unit calibrated under black body conditions is better than ±0.5 percent. Under operating conditions, the errors incurred by ambient-temperature fluctuation, the distance and focusing arrangement, the effect of absorbing media, and the uncertain total emissivities may increase the static error.

Optical pyrometers use a method of matching as the basis for their operation. In general, a reference temperature is provided in the form of an electrically heated lamp filament, and a measure of temperature is obtained by optically comparing the visual radiation from the filament with that from the unknown source. In principle, the radiation from one of the sources, as viewed by the observer, is adjusted to match that from the other source.

Figure 6.14 illustrates schematically an arrangement of a variable intensity pyrometer. In use, the pyrometer is sighted at the unknown temperature source at a distance such that the objective lens focuses the image of the source in the plane of the lamp filament. The eye piece is then adjusted so that the filament and the source appear superimposed and in focus to the observer. In general, the filament will appear either hotter or colder than the unknown source, as shown in figure 6.15(a) or (b).

By adjusting the battery current, the filament may be made to disappear as indicated in figure 6.15(c). The current indicated by the milliameter to obtain this condition may then be used as the temperature readout.

When accurately calibrated, the pyrometer will give excellent results above 1500 F, provided its use is restricted to the application for which it is designed, measurement of the temperature of the interior and a uniformly heated enclosure. When used to measure the temperature of a hot object in the open, the optical pyrometer will always read low.

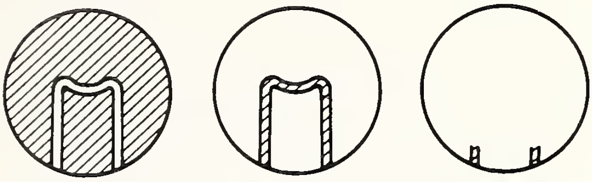
(a) Filament too Hot (b) Filament too Cold (c) Filament and Source at Same Temperature

FIGURE 6.15 *Appearance of filament under test conditions.*

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Page 169 At end of section on "Thermometry" add "The figures and part of the text for this section were adapted from T. G. Beckwith and N. L. Buck, "Mechanical Measurements" (2nd edition, Reading, Massachusetts 1969), are copyright by Addison-Wesley Publishing Company, Inc., and are used by their permission."

INFRARED THERMOGRAPHY

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National Bureau of Standards

Infrared thermography is an important tool in waste heat measurement. This is a measurement technique dependent upon an infrared radiation detection system which measures heat flux directly. Often significant heat losses occur due to incomplete confinement of heat in reactors, furnaces, or heat exchangers. These heat losses occur by conduction, radiation, and convective gas flow heat transfer. In the temperature range of 200-1000 F these losses are often large and have a large infrared radiation component.

The development of IR thermographic equipment, which uses a passive detection device to measure these heat losses, has enabled a powerful tool to be focused on this type of industrial heat loss. With this equipment one can find heat losses which result from:

1) Deterioration or damage to equipment;
2) Maintenance problems;
3) Nonoptimal design practices;
4) Operational procedures which waste heat.

In addition to the monitoring functions described above the IR thermographic equipment provides graphic, accurate information for calculations which can be used to predict energy savings. These savings can result from the modification of the heat equipment and operations and before and after IR images yield proof of the improvement in quantitative terms.

Although other means of temperature measurement or heat flow measurement such as thermocouples techniques can be used, these methods require interpolation, extrapolation, and averaging in order to relate the sample measurement to the total heat flux. The IR scanning image, however, can yield a picture of the total heat flux of a hot surface, thus avoiding many of the problems and expenses of previous techniques.

A further important advantage of this technique is the "living color" documentation of the evidence. Thus color (or black and white) thermograms are presented in a language which simplifies the communication between the engineers performing the survey and the management which ultimately must make the decisions on investments in energy conservation improvements.

An infrared (IR) thermographic camera is a device which converts invisible IR radiation into electrical signals, which are then amplified and transmitted by cable to one or more monitors, where they are converted to visible light and displayed on a screen.

A typical infrared thermographic camera operates by using rotating prisms to scan an image generated by a camera lens. The infrared radiation from the prism-scanning device falls on a small photovoltaic cell (usually indium antimonide), which is maintained at very low temperature with liquid nitrogen. Low temperature is essential to the photovoltaic cell, to retain the efficiency and stability of the cell. The photovoltaic cell converts the IR radiation to electrical signals, which are then amplified and converted to visible light by using them to modulate an electron beam which scans a screen like that in a television set, regenerating in visible light the infrared image which entered the camera.

The image generated by this technique is black and white, with darker shades representing lower temperatures. If quantitative data are required, isothermal contours can be generated; the detector can be set so that areas having a certain temperature range appear supersaturated white; by applying a series of such adjustments, the equipment can generate a series of isothermal contours outlining the temperature variation on the object being surveyed. However, when a thermographic image is scanned in this way, the nonlinearity of the instrument's response to temperature is reflected in a nonlinearity of the contour ranges, and calibration curves must be referred to.

Color thermograms are produced by using a three-gun camera such as is used in color television sets. Each color range is indicated by a particular combination of colors. Color thermograms are especially useful for displaying the results of a thermographic survey in slides or illustrations because of their striking appearance.

There are basically two approaches to the use of thermograms in waste heat management. In the qualitative approach, one uses the thermograms to detect individual "hot areas" indicating equipment in need of repair or replacement. In the quantitative approach, one uses the technique to "map" the temperature of the surface of equipment and then use this information to estimate heat losses by radiation. The estimated heat losses are then used in heat balances or energy audits. Before arranging for an IR survey team to visit a plant, plant management should decide what information they require and whether they can use a qualitative survey or need the more expensive quantitative survey.

In a thermographic survey of an industrial site, there are many important factors that should be considered. The relative importance of some of the more detailed factors may vary from plant to plant, however, the steps in performing a survey will follow a general pattern if meaningful information is to be obtained. An outline of a general procedure that has been found to be helpful in covering the more important steps of a thermographic survey is given below:

I. Preliminary Planning

The IR survey team should be informed about the principal energy using installations in the plant and what types of general energy intensive industrial equipment are used. Planning of the general strategy of the survey is also important so that the team is prepared for emphasis on flow detection, quantitative heat analysis, maintenance monitoring, or measuring effects of operational procedure. To make optimum use of the IR survey team's time, the plant staff should brief them carefully, well in advance of the scheduled visit.

II. On Site Analysis

The first meeting of plant management, engineering and operational personnel with the IR survey team is a time to agree upon the objectives and method of the survey. If the survey team is given an opportunity to discuss previous surveys and display thermograms, plant management may get valuable input for getting the most for their money. The plant operating personnel, understanding the capabilities and limitations of the equipment and its use, will be well equipped to help the team and to make suggestions on critical areas for careful inspection. Plant operations and equipment should be discussed and explained. Descriptions of the type and size of the equipment, the fuel used, maintenance schedules, and areas in the plant of special interest are important subjects to discuss. It is also helpful if plant personnel mention any special problems in production that relate to heat or energy, access points for inspection and control of furnaces, and the plant energy instrumentation and control system. These discussions can be followed by a plant "walk-through" for the IR survey team. This step will always be helpful in selecting scanning targets for the IR camera. These targets will include such items as the surfaces of furnaces, kilns and ovens; boilers, steam systems; electric power systems, prime movers; grinders; hot conveyors; flues; bearings; heat exchangers; products being processed at an elevated temperature; processing machinery operating an elevated temperature; and plant walls. Sections III-VI below are an outline of the procedure which the team should follow, and are given to give the reader an idea of what goes into an IR survey.

III. Placement of Thermographic Equipment

A. The thermographic equipment should be located to produce an IR image of each scanned subject which yields a clear, easily identified image. Thermograms of the greatest value to plant management will be those displaying scanned subjects which are not cluttered with spurious details or lack identifying silhouettes.

B. Select distance from subject to maximize the viewing area of the monitor. When larger areas are planned for the initial scan, always consider the possibility of scanning smaller areas which may indicate thermographic patterns of interest.

C. In each viewing area, a reference temperature source should exist. Usually, such a reference source can be a surface within the area being scanned where the temperature is known or can be measured by simple means such as a thermocouple, thermistor or surface pyrometer.

IV. Estimation of a Temperature Range

A. Estimate the peak temperature of area being scanned from known temperatures of the process such as the product, flue gases, or other adjacent surfaces.

B. Estimate the temperature range to include the peak temperature, lower temperatures of interest, and the temperature of the surface to be used for calibration.

C. Select an aperture and calibration curves to optimize temperature definition in the image.

V. Operation of the Instrument During the Survey

A. Refer to the manufacturer's manual for detailed instructions in connecting equipment components and in ascertaining critical operational limits.

B. Focus the thermographic scanner on the subject and investigate the entire target.

1. Investigate all irregularities (hot and cold) and determine causes—consult with the operator and plant engineers.
2. Estimate temperatures selected in scanned areas and rationalize these values with available information from plant personnel.
3. Select specific target subjects of interest to the survey.
4. Adjust the sensitivity and black level control for optimum display on monitor screen. The surfaces of interest in each display should fall within the gray area of the black and white monitor or within the color scale of the color monitor. Avoid settings which cause areas of interest to be represented as black or as saturated white.
5. Select a reference calibration technique and perform calibration procedures for areas where quantitative data are desired.

C. Photograph monitor displays of areas of significant interest. Record the instrument location, sensitivity settings, aperture, target description (including surface material and condition), and calibration procedures for each thermogram. It is important to maintain an adequate notebook with full details of each measurement.

D. Scan other targets in the plant for possible informative thermograms. Always scan the entire surface of subjects of interest.

E. Photograph the areas that were recorded from the monitor. An attempt to include in the photograph the identical areas recorded from the monitor is always helpful in defining the hot and cool areas on the thermogram. These photographs should be identified and correlated with entries in the notebook.

VI. Analysis of the Results of the Survey

A. Determine the location and significance of faults and apparent weaknesses in the design of equipment and processes.

B. Estimate heat losses from equipment operation and processes via thermal radiation. In cases where major heat losses are measured, quantitative estimates can be made.

C. Analyze possible solutions to improving the energy efficiency of the equipment and processes.

D. Determine possible areas in the system where heat recovery equipment may be of significant value.

E. Recommend a course of action to the plant management.

Rather than developing an in-house team to perform the infrared heat survey with their own equipment, many firms may find it preferable to obtain these services on outside contracts. Fortunately, numerous organizations have been established which specialize in performing thermographic surveys for industrial plants. These consulting teams use procedures similar to those outlined in this section but it may be useful to compare their services with the procedure described above. It is very important to agree upon the goals of the survey with the consulting team so that the appropriate information is obtained. Since the capabilities of these groups vary somewhat and one may have difficulty finding the best team it may be most useful to contact equipment manufacturers to obtain references to one or more of these service organizations having expertise in the general type of plant to be surveyed.

In buying thermographic services, some of the factors that should be considered are listed below:

1. Obtain services from an organization that has

personnel experienced in performing thermographic surveys in plants with equipment of the same general nature as that to be covered in the plant being considered. For example, some service organizations specialize in performing surveys on electrical power lines looking for defects in transformers or insulation; others specialize in surveying single and multifamily dwellings to analyze the quality and efficiency of the building insulation.

2. Obtain services from organizations with thermographic equipment well suited to the survey to be performed. Each manufacturer of thermographic equipment features characteristics which may or may not be of importance in the survey to be performed.

Examples of such characteristics include:

a) high resolution of display
b) high scanning rates
c) portability of equipment
d) various fields of view of the camera
e) color monitor
f) high sensitivity at the lower temperatures
g) simplicity in obtaining quantitative data.

There is one final point to be emphasized. To obtain the maximum benefit from a thermographic survey, one must involve both operating and maintenance personnel. Persons from both groups should accompany the survey team, not only to explain any problems they may be experiencing which could indicate a need for especially careful examination of certain areas, but also to discuss at first hand with the survey team (who may not be entirely familiar with the types of equipment used) the implications of the measurements. The same plant personnel should be involved with managemement in the review of the results of the thermographic survey.

FLUE GAS ANALYSIS

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The principal reason for measuring flue gases is to assess the efficiency of the combustion process involved. Therefore, the number of chemical species which are necessary to define the efficiency can be limited to a very few. These are oxygen, carbon dioxide, carbon monoxide, hydrocarbons, water, and in some cases the oxides of nitrogen and sulfur. Methods for measuring these materials with adequate accuracy have been available for a number of years. In fact, the early development of gas analytical methods in the latter part of the last century was largely a result of the need for accurate measurement of flue gases. The accuracy of such measurements has not improved significantly during the last century but the ease of determination and the rapidity of measurements has increased markedly due primarily to the development of instrumental methods of analysis. The early measurements were made using manual volumetric techniques which in general depended for success on the skill and dedication of the operator. These methods are still in use but modification to the operations have increased the reliability and decreased the level of technical competence necessary to assure good results. In addition to the improved manual methods there are available a number of instruments which are capable of continuous or intermittent measurement of one or more species and which are suitable for measurement in the rather harsh environment of a flue or stack.

There are many instrumental techniques by which the composition of a flue gas could be determined. Some of these are reasonably accurate and simple. Many are complex in operation and delicate in nature requiring great care on the part of the operator both in manipulation of the sample and in interpretation of results. Therefore, it might be a good idea to define the problem of flue gas analysis more precisely with a view toward limiting the discussion to those instruments and methods that can most conveniently be utilized for the specific application. Thus, we will not discuss general purpose gas analysis instruments which are capable of performing the analysis but which for various reasons are seldom employed for flue gas analysis. Nor will we discuss experimental methods which have been described in the literature but which do not yet have a history of application on which to base some measure of confidence.

The factors which are of importance in flue gas analysis and the manner in which they modify the analytical approach are described below.

The composition of a flue gas can be predicted with a high degree of certainty because both the nature of the fuel and the nature of the combustion process are known with some certainty. Thus a method of analysis can be applied which is not rigorously specific to all compounds but which merely needs to be specific for the particular compound in the presence of a limited number of other compounds. Carbon dioxide, for instance, may be determined by a nonspecific method which measures the acid gas fraction of the flue gas. In most cases, carbon dioxide is the only acid gas of consequence, the other acid gases being present at a level below the level of precision of the method. Alternatively, the other acid gases may be measured roughly and a small resulting correction may be applied to the carbon dioxide value.

Accuracy in gas analysis is achieved in either of two ways. First, the measurement may be made by a technique for which the accuracy can be defined by the uncertainties involved in the measurement. Second, precise but not accurate methods of analysis can be calibrated with standards of known composition or by comparison to a method of known accuracy. Classical gas analytical methods in which a weight or volume change is the basis of measurement can be defined in terms of accuracy because the mass or volume measuring devices can be calibrated by reference to fundamental units of length or mass. Instrumental methods on the other hand usually produce a signal dependent on a property of the gas being measured which cannot be related to any fundamental unit. The principal exceptions are those electrochemical instruments which measure a total current. Other types of instruments are not usually intended to be absolute measuring devices and are always calibrated by reference to a standard of some sort. The standard in the case of a flue gas analyzer may be a gas sample of known composition. The composition of the standard must be known with accuracy if measurements made by the instrument are to be related to the true concentration of the species being measured. Alternatively, a "calibration curve" for an instrument can be prepared as needed by comparing the instrument signal to the concentration of sulfur dioxide or the oxides of nitrogen, a standard for the analysis can be prepared using weighed quantities of an intermediate product in the chemical reactions by which a material is determined. A nitrate salt, for instance, is the basis for standardization of several wet chemical methods intended for the measurement of nitrogen dioxide. The nitrogen dioxide is collected in solution as the nitrate ion.

The accuracy which results from the use of any of these techniques varies. Gas mixtures of known stability can be certified with a total uncertainty of about ±1 percent. Several such standards are now available as National Bureau of Standards Standard Reference Materials. These include standards for measurement of carbon dioxide, carbon monoxide, nitric oxide, oxygen and hydrocarbons. The accuracy of these standards can usually be transferred to the measurement by an instrument, providing that the precision with which the measurement can be made does not introduce substantial errors.

Some caution must be exercised in the use of supposedly accurate methods. It is no great feat to calibrate a gas burette with a total uncertainty of ±1 percent and to measure subsequent gas volumes with the same uncertainty. Much work has been reported on the errors in analyses based on this type of apparatus and significant systematic errors have often been found. These arise from problems associated with incomplete reactions, solubility, adsorption and from competing reactions in which a second gaseous species not present in the original sample is generated in the absorption process. Obviously, no method should be applied to flue gas analysis unless the absence or magnitude of systematic errors under the condition of the analysis are known.

The rapidity with which sequential analyses can be performed may be a factor in the choice of a method and may necessitate compromises in other criteria concerning the analysis. Rapid adjustment of combustion equipment obviously requires short term information concerning the composition of the combustion products. An instrument capable of real-time measurement of a substance in the flue gas is the first choice but few such instruments are available. Further, most instruments must of necessity be located at some distance from the sampling point and may in addition require sample treatment to remove gross quantities of water and interferring substances prior to analysis. Some instruments do not respond instantly to changes in concentration and often the equilibration time is substantial.

Other analyses, those involving control or measurement of pollutants for example, may not require real time measurements but only the accumulation of long term data.

Chemical methods, both those involving measurement of changes in gas phase volumes and those in which the species of interest is adsorbed in a reagent, require rather lengthy periods in which to complete an analysis. Instrumental methods require comparatively shorter analytical times.

There is no simple "black box" type of analyzer that can perform analyses of flue gases. Some approach this concept but even these require some degree of competence on the part of the operator if meaningful results are to be obtained. A high degree of skill is seldom required in the use of most instrumental methods but this is not true of the chemical techniques. Accurate results with the chemical methods usually require careful attention to analytical details and some knowledge of the general field of gas analysis.

The necessity for periodic calibration of instruments can be a complicating factor in some analyses. Some instruments such as flame ionization hydrocarbon analyzers maintain adequate calibration for long periods of time and a single daily calibration is all that is necessary. Instruments which are especially sensitive to changes in ambient temperature or pressure may require calibration at much shorter intervals if ambient conditions are changing. The periodicity of the calibration can be modified considerably by the requirements for accuracy. An instrument which is intended to produce results with the highest accuracy will probably have to be calibrated more often than an instrument intended to monitor gross changes in concentration of a gas.

The degree of complexity of calibration is affected by the standardization. There exist sophisticated gas analysis systems in which the calibration is performed automatically at preset intervals. Most systems, however, require manual injection of the standards.

It is not necessary to standardize volumetric or gravimetric apparatus after the initial calibration of the components. It is advisable to periodically analyze reference standards with the apparatus to be certain that systematic errors unknown to the analyst have not arisen.

Reliability is a "catch-all" when defining the properties of an instrument or a chemical method. Reliability can include such factors as instrument stability but it also refers to the dependability of the instrument or method in terms of the expected period of trouble-free and accurate operation. Most chemical methods are as reliable as the analyst and except for occasional breakage, apparatus may give dependable service for many years requiring only routine maintenance and periodic confirmation of the calibration.

Instrumental methods of analysis on the other hand generally do not operate for very long periods of time without high-quality maintenance. The trend in recent years to solid state electronics has eliminated many maintenance problems associated with vacuum tube circuitry. However, many sensing devices used in gas analysis instruments do not have infinite lifetimes and inattention to the gradual changes in performance which occur can lead to inaccurate results. In addition, the nature of flue gases is such that much care is necessary to protect the instruments from the rather large quantities of water, corrosive gases and particulate matter that may be present.

The frequent recalibration of instruments such as nondispersive infrared devices will aid in determining changes in instrument characteristics due to changes in the infrared source and to contamination of the optics. Gas chromatographic columns may change retention characteristics due to bleeding of the liquid phase or due to contamination by normal constituents of the sample.

This section deals with analysis of single components by both manual or wet chemical method and by instrumental methods. The section is obviously not all inclusive. Nor are the methods described in great detail. The intent is to describe a few representative methods in order to give some concept of the degree of difficulty of the analysis and of the type of measurements which are necessary. It should be kept in mind that there are many factors involved other than the final chemical or instrumental measurement. Many of the methods, for instance, require an accurate measurement of the volume, temperature or pressure of the sample. A description of these measurements would by itself constitute a major undertaking. The method by which the sample is obtained from the combustion system may have very significant effects on the values obtained by the analyst. The placement of sampling probes in the gas stream, for instance, must be done with a knowledge of the characteristics of the stack or flue or the chemical results may be of little value. All of the factors must be understood or controlled in the analysis but ultimately the method by which the concentration of the substance is measured is the key factor.

a) Carbon Dioxide

The carbon dioxide concentration normally encountered in analysis of flue gases may lie between nine and fourteen volume percent. Obviously, the concentration of carbon dioxide cannot exceed 21 percent if the only process involved is combustion. Measurement of carbon dioxide in this concentration region can be performed with an inaccuracy of a few percent by the manual methods and to about the same limits with a properly calibrated instrument.

Manual Methods

Carbon dioxide may be analyzed quite accurately by a gravimetric technique in which the carbon dioxide from a measured volume of air or other gas is absorbed by a solid alkaline reagent. However, the method is not suitable for measurement of carbon dioxide in flue gases because of the care necessary to insure selectivity and because of the necessity for accurate measurement of the volume of a stream of gas and the necessity for measuring the gain in weight of the absorbent with accuracy. It is more convenient to measure directly the volume of carbon dioxide in the sample because the apparatus involved is more rugged and easier to maintain in calibration. The apparatus consists essentially of a sample pipette in which the sample is collected, a gas burette in which the volume of the sample is measured before and after removal of the carbon dioxide and an absorption pipette in which the carbon dioxide is removed. The sample is obtained by flushing the gas through the sample pipette until a representative sample is obtained or by drawing the sample into a previously evacuated pipette. The sample pipette is then attached to the gas burette and a portion of the sample is transferred to the burette. The sample in the burette is equilibrized at a measured temperature and pressure and the volume is noted. The sample is then transferred intact to the absorption pipette which contains a concentrated solution of an alkali such as potassium or sodium hydroxide. After absorption is complete, the sample is transferred back into the gas burette and the volume is measured under the same conditions of temperature and pressure at which the original sample volume was measured. The difference between the two volumes represents the volume concentration of carbon dioxide present in the original sample.

Many different designs for this purpose have appeared over the years ranging from simple pipettes and burettes connected with rubber tubing to rather elaborate all-glass systems in which the gas burette is thermostatted and pressure compensating devices are provided for eliminating errors in adjusting the sample pressure before and after absorption.

There is also available a small device by which measurements of high concentration of carbon dioxide may be made by observing the pressure change occurring when carbon dioxide is removed from a sample. The sample is drawn into a small jar equipped with a Bourdon tube vacuum gage and is sealed off. The jar contains a layer of concentrated potassium hydroxide isolated from the sample by a layer of oil floating on top. After the sample is sealed the jar is shaken to bring the reagent into contact with the sample. The reduction in pressure after the reaction is complete is proportional to the amount of carbon dioxide originally present in the sample.

Instrumental Methods

Nondispersive infrared analyzers are a convenient means for the continuous measuremement of carbon dioxide in flue gases. In principle these instruments are quite simple. The energy from an infrared source is passed through a cell containing the sample and is directed into a detector cell containing the species to be analyzed. If the species to be analyzed is not present in the sample cell then the specific wavelengths absorbed by the species will pass unaffected to the detector cell. The detector cell, containing the species to be detected, will absorb the specific wavelength and a signal can be generated. If, however, the sample cell contains the species to be detected then less light will reach the detector and the signal will be reduced correspondingly. There are many combinations of optical filters, reference and sample cells and detectors which allow a fair degree of specificity and a relatively high degree of sensitivity over a wide range of concentration and these instruments can be engineered to give accurate and rapid results for a particular application. Inaccuracies of ±1 percent or less are commonly reported but because the instruments are not "absolute," attaining this degree of accuracy requires standards of equal or better reliability.

Some sample treatment is necessary especially where large amounts of water vapor are present but such sample treatment systems need not be elaborate or expensive. Interferences do not seriously affect the use of nondispersive infrared analyzer for analysis of flue gases because both the nature and concentration of potential interferents can usually be predicted with some degree of certainty.

b) Oxygen

Maximum efficiency in a combustion process requires careful control of the air supply. Enough air must be supplied for complete combustion but excess air results in wasted energy. The oxygen content in the flue gas for maximum efficiency will vary depending on the type of fuel used but generally will lie below 10 percent.

Manual Methods

Oxygen may be continuously measured by instrumental means or individual samples may be measured chemically with the same volumetric apparatus described for carbon dioxide. The only difference is that the reagent used in the absorption pipette is an alkaline solution of pyrogallol prepared by dissolving pyrogallic acid in strong alkali. The reagent absorbs carbon dioxide also and it is necessary to remove the carbon dioxide from the sample prior to adsorption.

This can be done easily by first measuring the carbon dioxide by absorption in potassium hydroxide and then replacing the pipette containing the carbon reagent with a pipette containing the oxygen reagent and measuring the oxygen remaining in the sample. The manipulations are not difficult and a short period of practice is sufficient to allow an operator to measure both oxygen and carbon dioxide in a sample with reasonable accuracy.

Instrumental

There are two properties of oxygen which can be utilized in continuous measuring instruments. These are the paramagnetism of oxygen and the other is the electrical potential developed by oxygen in oxidation-reduction processes. Methods depending on both properties have been widely used and the selection of a method will depend largely on the nature of the flue gas to be analyzed. In particular, the paramagnetic instruments cannot be used on gases containing dust of strongly paramagnetic or ferromagnetic substances.

i) Paramagnetic Instruments

There are two different types of instruments in which the paramagnetism of oxygen is utilized as a measure of its concentration. The first, the "Pauling Meter," measures the degree of displacement of a small body from an inhomogeneous magnetic field as a function of the concentration of oxygen.

In the second type, the so-called "magnetic wind instrument," the change in paramagnetism with temperature is utilized to measure oxygen. The paramagnetism of oxygen decreases with increasing temperature. Oxygen is attracted into a magnetic field. Cooler oxygen outside the heated area will displace the warmer oxygen. This effect can rather simply be utilized to cause a continuous movement of oxygen, the rate of movement of which will be proportional to the concentration of oxygen. The "wind" effect can be sensed with a thermal conductivity cell. This latter type of instrument is extremely rugged and can have a long useful lifetime. If the gas is cleaned of particulate matter and condensable materials the lifetime can be many years. Thermal conductivity devices of the kind employed here operate at relatively low currents and last for long periods of time. The magnetic field is supplied by a large permanent magnet.

The accuracy of such instruments, both the Pauling and magnetic wind types must be established by the use of standards. Standards of oxygen are readily available from ambient air and even if the ambient air is only relatively pure it will still serve as a satisfactory standard. Accuracy of ±1 to 2 percent are not difficult to attain.

The Pauling meter is somewhat fragile and must be protected from mechanical shock and from pneumatic shock such as might occur in a sudden change in sampling rate. The magnetic wind instruments are extremely rugged.

ii) Electrochemical Instruments

There are many oxidation reduction reactions which can be used to measure oxygen. There is one method, however, which is ideally suited for measurement of oxygen in flue gases. The measuring element of the system is a zirconium oxide barrier separating the sample from a reference gas. The zirconium oxide in the form of a wafer is plated on both sides with porous platinum to serve as electrodes. The cell is operated at a thermostated temperature of about 1500 F. Oxygen molecules on the reference side will normally be a higher partial pressure than on the sample side if flue gases are being analyzed and if normal air is used as a reference. Because of the concentration differential oxygen molecules from the reference side will pass into the zirconium oxide electrolyte. The sensing cell can be placed directly into the gas stream to be measured and no sample conditioning is required.

The cell can be calibrated quite easily outside of the sample stream and will operate for periods of more than a year with only occasional checks of the attendant equipment. The cell can be used for most types of flue gas except those containing high concentration of hydrocarbons. These will combust on the surface of the cell resulting in a depletion of oxygen and a consequent erroneous reading.

c) Carbon Monoxide

Carbon monoxide is sometimes measured in flue gases as an index of the completeness of combustion. All combustion processes will produce some carbon monoxide but the amount should be minimized.

Manual Methods

When carbon monoxide is passed over heated iodine pentoxide the carbon monoxide is oxidized to carbon dioxide while the iodine pentoxide is reduced to iodine which is volatilized. The iodine absorbed is determined by subsequent titration with standard sodium thiosulfate solution.

This method is quite accurate and is almost as simple as most other routine wet chemical gas analysis methods. Accuracy depends on the quality of the reagents and the care taken in performing the analysis. Uncertainties no greater than a few percent can be achieved routinely. It is necessary to remove interfering substances prior to exposure of the sample to the iodine pentoxide. The number of interfering substances is large and a rather robust cleaning procedure must be employed. In spite of this, the method is perhaps the best wet chemical technique for measurement of carbon monoxide.

Instrumental Methods

The most widely used instrument for the continuous measurement of carbon monoxide is the nondispersive infrared analyzer. Higher concentrations of carbon dioxide and water vapor are usually found in conjunction with carbon monoxide and both interfere to some extent especially if their concentrations are varying. It is necessary, therefore, to remove or compensate for the presence of these compounds. Both may be scrubbed from the sample stream with suitable reagents but if the analyzer is used to monitor carbon monoxide continuously the maintenance of the scrubber becomes a problem.

Alternatively, a cell containing carbon dioxide can be interposed between the sample cell and the detector. Thus, all radiant energy at wavelengths absorbed by carbon dioxide is removed before reaching the detector cell and variations of carbon dioxide in the sample cell will not affect the response of the detector. A similar scheme may be used with water vapor but it is simpler in this case to saturate the sample stream ahead of the sample cell thus producing a constant concentration of water at all times.

The accuracy of this type of instrument is claimed to be ±1 percent but periodic standardization with gases of known concentrations is necessary. If a wide range of concentration is to be measured it will be necessary to use standards covering the range because the response is not linear with concentration except over a short range. These instruments are reliable in operation and only require occasional cleaning and replacement of the infrared source.

d) Hydrocarbons

Unburned hydrocarbons in the flue gases of an oil fired burner are wasted energy. They arise not only from incomplete combustion but also from improper adjustment of burner equipment.

Instrumental Methods

The analysis of hydrocarbons in a flue gas will in most cases be concerned with low concentrations. There are no satisfactory manual methods applicable to this type of analysis but there are several instrumental methods. Three that are relatively simple in operation, accurate and dependable are nondispersive infrared analyzers, flame ionization detectors and hot wire detectors. The principle of the nondispersive infrared analyzer has been discussed and for hydrocarbons it is only necessary to adjust the gas in the detector cell for the specific hydrocarbons or class of hydrocarbons to be detected. As with all nondispersive infrared instruments considerable sample conditioning may be necessary prior to analysis.

Flame ionization detectors utilize the current flow through a small hydrogen flame as an indication of hydrocarbon content. A small burner is supplied with hydrogen as fuel and sample air as a source of oxygen for combustion. A potential is established between the burner and a small electrode mounted slightly above the flame. If only inorganic gases are present then little current is observed but when an easily ionized molecule is present in the flame a current will flow. Any molecule containing carbon and hydrogen as constituent atoms is easily ionized and will give a strong signal relative to inorganic molecules.

A problem with flame ionization detectors may arise if the oxygen concentration varies during the course of analysis or between calibrations. Sample conditioning is required to remove particulates and large amounts of water vapor. Standardization with gas mixtures of appropriate hydrocarbons is necessary. However, the response of the instrument is nearly proportional to the carbon atom concentration and a calibration with one hydrocarbon can be used to determine others. Uncertainties of analyses of less than one percent can be achieved in routine operation.

The "hot wire" hydrocarbon detector measures hydrocarbons by combustion on the surface of a wire and measures the change in temperature of the hot wire. The wire must be hot enough to effect combustion and must consist of a material of high electrical resistance and chemical stability. The hot wire is included as one arm of a Wheatstone bridge circuit and the change in temperature is indicated as a change in resistance of the hot wire.

Any combustible gas will react on the hot wire, including carbon monoxide and hydrogen. If a determination of hydrocarbons as a separate component is necessary then the other combustibles must be determined independently. The accuracy of these devices is not very great especially at low concentration and their calibration is affected by deterioration of the wire. They are best used for intermittent measurements but can be used continuously. Low concentration of oxygen in the sample can affect the rate of combustion and consequently an error can be introduced. However, it is a simple matter to add a fixed proportion of clean air to the sample to insure complete combustion.

e) Oxides of Nitrogen and Sulfur Dioxide

These substances are only indirectly related to the efficient utilization of fuel but their measurement and control effects the manner in which the combustion equipment must be operated. In general, the concentrations to be expected fall in the range below one percent.

Manual Methods

Many chemical methods are available for the measurement of nitric oxides and sulfur dioxide in flue gases. The methods for the oxides of nitrogen are all colorimetric and are based on reaction of nitrogen dioxide with organic materials to form highly colored dyes. The nitrogen dioxide is collected in a reagent and nitric oxide must first be oxidized to nitrogen dioxide. The phenoldisulfonic acid method has been widely used and consists of collection in an oxidizing medium followed by reaction with phenoldisulfonic acid. The oxidant in the collecting medium is hydrogen peroxide and the product is essentially nitric acid. The acid nitrates the phenoldisulfonic acid forming a yellow compound which is measured with a spectrophotometer. Calibration of the spectrophotometer is accomplished with standards of potassium nitrate reacted with phenoldisulfonic acid.

The accuracy of the method is about ±4 percent and the precision is only ±5 percent. However, this is one of the best chemical methods available for measurement of nitrogen oxides at the concentrations encountered in flue gases.

The history of the analysis of sulfur dioxide is long. Several methods are applicable to flue gas analysis. The peroxide method involves oxidation of sulfur dioxide to sulfate ion with a dilute solution of hydrogen peroxide and titration of the resulting sulfuric acid with a standardized base. The method is fairly rapid and accuracy of ±1 percent is not difficult to attain. However, the method is not specific and any acid gas soluble in the absorbing solution will be measured as sulfur dioxide.

A more specific but somewhat more tedious technique is the barium sulfate method. Sulfur dioxide is oxidized to sulfate during collection in hydrogen peroxide solution. The sulfate is subsequently precipitated as barium sulfate by the addition of barium chloride. The precipitate is filtered from the solution, dried and weighed. The accuracy of the method is similar to the peroxide method but it has the distinct advantage of being specific.

Instrumental Methods

The relevant chemical methods for sulfur dioxide and the oxides of nitrogen may be used in a family of instruments which are essentially automated manual methods. Absorbents and reagents are metered at accurate flow rates through scrubbers and reaction vessels after which some property of the solution is measured either continuously or periodically. The instruments have simple photometers for colorimetric reaction and conductivity cells or coulometers when appropriate.

These instruments while highly sophisticated mechanically require considerable maintenance because of the sometimes complicated liquid flow systems.

In addition to these, there are instruments for both sulfur dioxide and nitrogen oxides which are capable of direct continuous measurement in the gas phase. A molecule of nitric oxide reacts with a molecule of ozone to form a molecule of nitrogen dioxide. The nitrogen dioxide thus formed momentarily contains excess energy which is subsequently emitted as light. A photomultiplier can be used to measure the quantity of light, which when all other conditions are fixed, will be proportional to the concentration of nitric oxide.

Nitrogen dioxide in a sample containing both oxides can be measured if it first reduced to nitric oxide with a suitable catalyst and measured along with the nitric oxide. A subsequent measurement of the sample without reduction of the nitrogen dioxide will allow a calculation of the nitrogen dioxide by difference. These instruments are very sensitive and stable but do require calibration with standards of nitric oxide or nitrogen dioxide. The accuracy is about ±1 percent. The maintenance requirements are moderate requiring occasional replacement of the catalyst for conversion of nitrogen dioxide to nitric oxide. The instruments contain an ozone generator for generation of the necessary ozone. The ozone may be generated from ambient air or from a supply of oxygen.

There are currently no widely used analyzers for direct measurement in the vapor phase of high concentration of sulfur dioxide. An instrument is available which measures low concentration by measuring the spectral emission of sulfur in a flame. The instrument may be used at high concentrations by dilution of the sample with clean air. The accuracy of a properly calibrated instrument for measuring high concentrations is only between 2 and 5 percent. The instrument requires a continuous supply of hydrogen and occasional cleaning.

All of these instruments require sample conditioning prior to analysis. The direct reading instruments for sulfur oxides and nitric oxide are specific and require removal of only those materials which would contaminate the system. Many of the automated chemical systems, however, do not employ chemical reactions which are truly specific and it may be necessary to scrub interfering substances from the sample stream.

f) Water Vapor

The measurement of water vapor is of importance in several areas. Corrosion control especially in gases containing high concentration of sulfur compounds does not directly benefit energy utilization but certainly represents a potentially profitable operation. Careful control of the moisture content of high sulfur flue gases can significantly reduce corrosion. Measurement of water vapor in control of condensate recovery represents a direct means of energy savings.

Manual Methods

Perhaps the simplest method for measuring water vapor in a flue gas is with a wet-dry bulb thermometer. If the mercury bulb of a thermometer is covered with a continuous film of water and a stream of air is passed over the bulb the water will evaporate at a rate determined by the water content of the passing air. The bulb will be cooled by evaporation and the indicated temperature will be somewhat lower than the true air temperature. If a second thermometer is placed in the air stream but with no water on the bulb then the difference in reading between the two thermometers will be a function of the moisture content of the air. These devices are true indicators of relative humidity because they measure moisture content relative to saturation at the air temperature. Conversion of wet-dry bulb readings are simplified by the existence of numerous "hygrometric charts" which allow direct conversion of the two temperatures to relative humidity. The true concentration of water vapor can be calculated from measured relative humidity, dry bulb temperature and data concerning the saturation pressure of water.

Many different configurations of these "hygrometers" have appeared over the years. Most are designed for measurement in the ambient atmosphere where the air movement over the bulbs is accomplished by moving the thermometers. The air, of course, can be mechanically moved over the bulbs and this is the usual method of sampling flue gases.

Some care is needed in the design of hygrometers of this type particularly in the manner in which water is supplied to the bulb. Normally a light fabric wick fed by a small reservoir of water is used. The measurement of rapidly changing relative humidities is difficult to perform with accuracy because the temperature of the wet bulb reflects to some extent the temperature of the water in the reservoir.

No calibration of this type of hygrometer is needed providing that the thermometers are calibrated. The accuracy, however, is not too great. At 100 F, for example, a difference of about 6 percent relative humidity can be distinguished when the air is near saturation. Thus, a slight error in the wet bulb temperature either due to operator error or to faulty design can be a significant source of error.

If the moisture content of the flue gases is higher, above a dew point of 260 F, then it is necessary to use a different method. A simple approach is to draw at a measured flow rate continuously through a condenser held at a considerably lower temperature than the flue gas. The volume of condensate accumulating with time can be used to calculate the concentration of water vapor in the stack.

Instrumental Methods

The history of the development of instruments for the measurement of water vapor is long and frustrating. Many instruments are available which are capable of detecting and measuring water vapor at high concentration. However, application to flue gas analysis has not been too successful. If a hot flue gas is supersaturated with water vapor with respect to ambient conditions then it is necessary to make the measurement in the stack. This may be complicated by a number of factors including temperature, rate of flow and particularly by the presence of oxides of sulfur.

If a satisfactory measurement is to be made it will probably have to be based on a measurement in the gas phase by a method specific for water. Infrared measurement of the water vapor utilizing very strong absorption bonds in the spectrum of water is probably the best choice.

There are several gas analysis instruments which are capable of quantitative analysis of multicomponent gas mixtures and which could be adapted to the continuous analysis of flue gas. However, the cost, complexity and degree of training of the operator would be prohibitive and it is not likely that such systems would be used in any but the most unusual situations. It is possible to use a single instrument to analyze for two and perhaps three components.

There are only two basic types of instruments involved in gas analysis which are capable of generating a series of signals each of which can be related to a single component in a mixture. These are mass spectrometers and gas chromatographs. Mass spectrometers are generally expensive and delicate and require experienced operators and constant maintenance. It is possible to design an analytical system based on a mass spectrometer which will rapidly and automatically analyze samples of flue gas but the cost of such a system would be enormous. The specificity for the components of flue gas, except for water and nitrogen dioxide, would be excellent and an accuracy of 1 or 2 percent could be obtained.

Gas chromatographs have been applied to the total analysis of flue gases with some success. These systems are rather complicated requiring essentially several chromatographs operating in tandem. The time required for analysis is rather long and a chromatograph cannot be considered a continuous analyzer. However, chromatographic determination of several components of a mixture, such as oxygen, carbon dioxide and carbon monoxide could be performed in a relatively short time with a fair degree of accuracy and excellent specificity.

The present state of multicomponent analyzers is really not such that they are well suited for flue gas analysis. A system composed of individual instruments specific for each component making continuous measurement would seem preferable.

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Chapter 7

ENGINEERING DATA FOR WASTE HEAT RECOVERY

In this chapter are compiled data and graphs which are used in the various case studies and which are valuable in analyzing large classes of waste heat problems. The first part of this chapter consists of these graphs:

Figure 7.1 Natural gas combustion chart.
Figure 7.2 Heat content of the major components of flue gases.
Figure 7.3 Heat capacity of the major components of flue gases.
Figure 7.4 Theoretical dew points of flue gases from industrial fuels.
Figure 7.5 Fuel savings available using combustion air preheating.
Figure 7.6 Heat recovered by recuperation as a function of flow rate and air preheat temperature.
Figure 7.7 Fuel saving nomographs for air preheaters.

The second part of the chapter consists of tables of data:

Table 7.1 Heat transfer coefficients for steel pipes and plates.
Table 7.2 Steam table extracts.
Table 7.3 Apparent specific heats of steels.
Table 7.4 Physical properties of fuel oil at 60°.
Table 7.5 Typical properties of petroleum fuels.
Table 7.6 Properties of gases.
Table 7.7 Conversion table to SI units.

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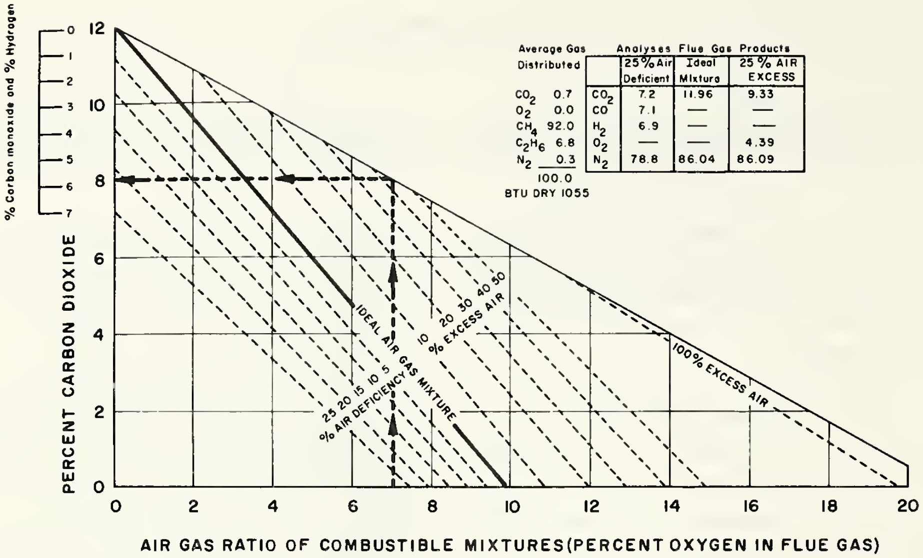

Average Gas Distributed		Analyses Flue Gas Products	25% Air Deficient	Ideal Mixture	25% AIR EXCESS
CO_2	0.7	CO_2	7.2	11.96	9.33
O_2	0.0	CO	7.1	—	—
CH_4	92.0	H_2	6.9	—	—
C_2H_6	6.8	O_2	—	—	4.39
N_2	0.3	N_2	78.8	86.04	86.09
	100.0				

BTU DRY 1055

Figure 7.1 *Natural gas combustion chart.*

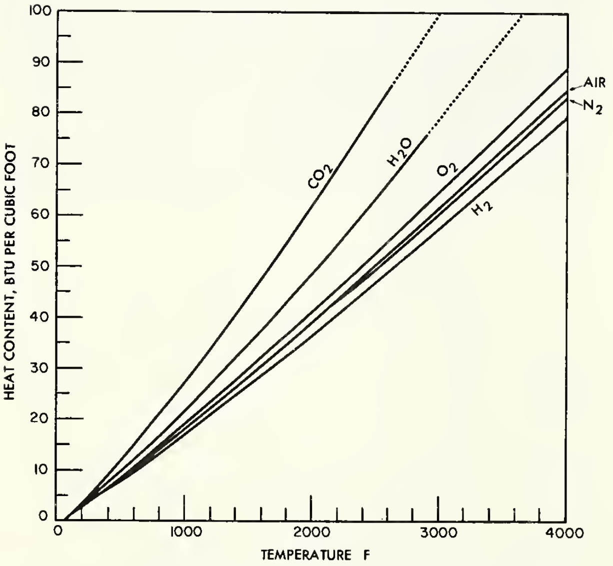

Figure 7.2 *Heat content of gases found in flue products, based on gas volumes at 60 °F (dashed line indicated dissociation).*

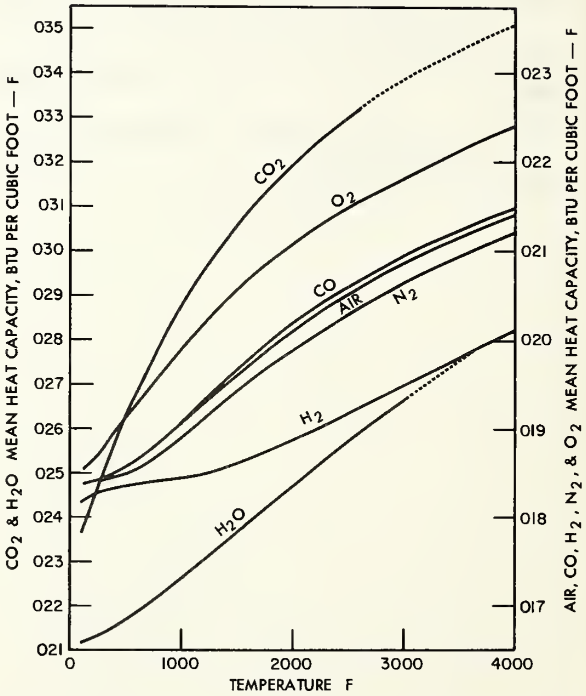

Figure 7.3 *Mean heat capacity of gases found in flue products from 60 to T F (dashed line indicated dissociation).*

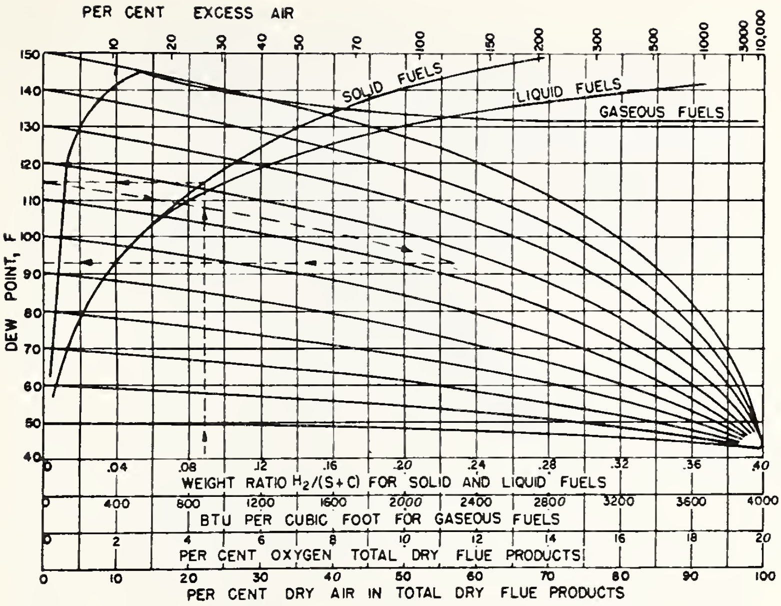

FIGURE 7.4 *Theoretical dew points of the combustion products of industrial fuel.*

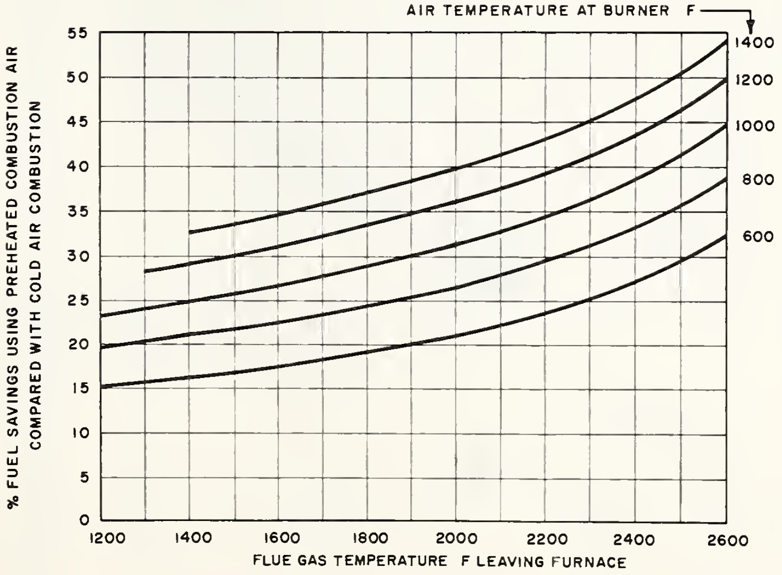

FIGURE 7.5 *Fuel savings as a function of flue gas temperature F leaving furnace.*

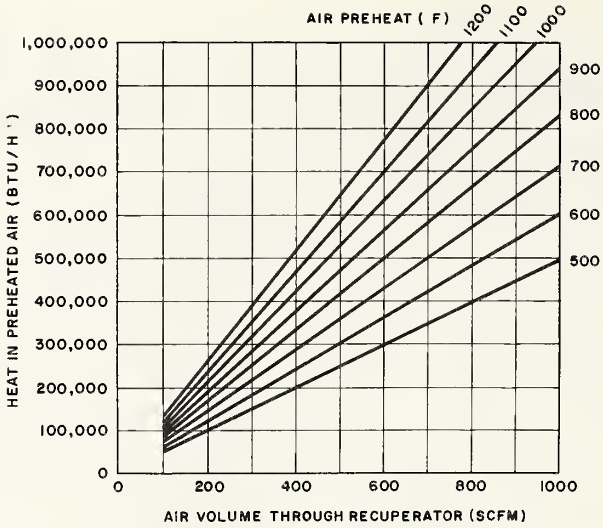

FIGURE 7.6 *Heat recovered as a function of air volume through recuperator (SCFM).*

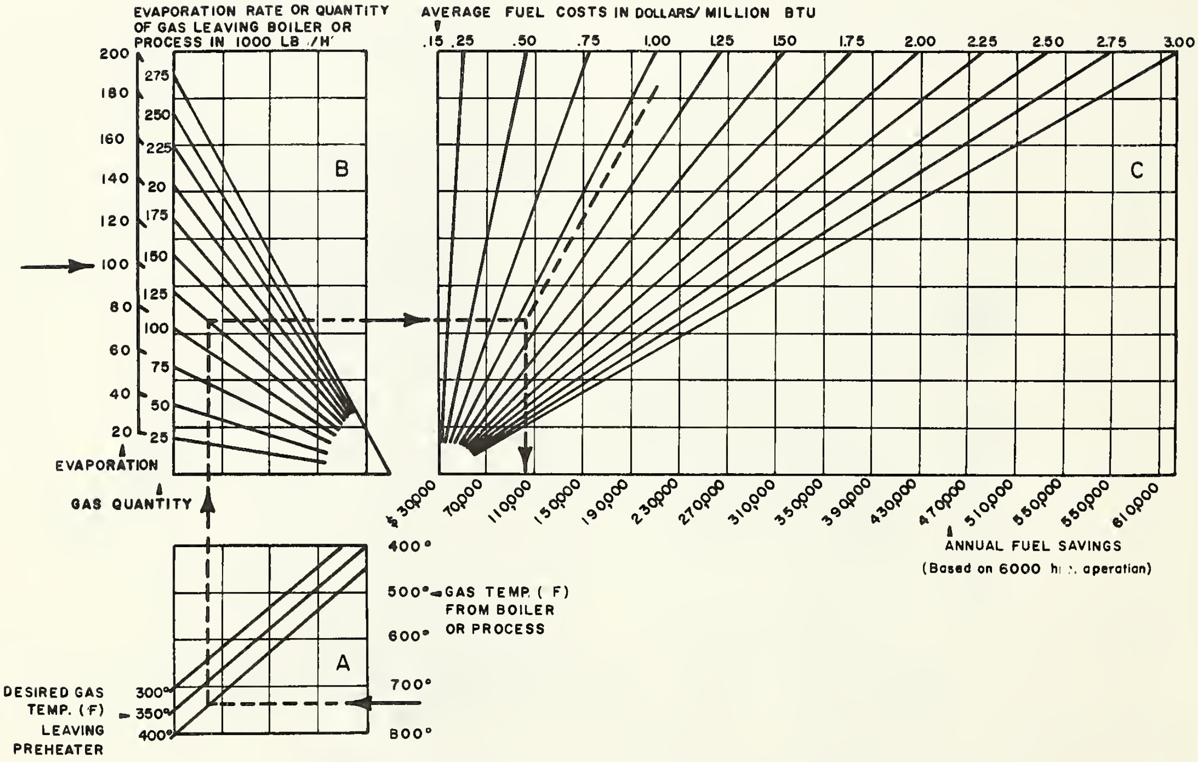

FIGURE 7.7 *Nomograph for estimation of annual fuel savings using a regenerative air preheater.*

Table 7.1 Values of $(h_c + h_r)$

(For horizontal bare or insulated standard steel pipe of various sizes and for flat plates in a room at 80 F)

Nominal pipe diam, in.	$(\Delta t)_s$, temperature difference, deg F, from surface to room														
	50	100	150	200	250	300	400	500	600	700	800	900	1000	1100	1200
½	2.12	2.48	2.76	3.10	3.41	3.75	4.47	5.30	6.21	7.25	8.40	9.73	11.20	12.81	14.65
1	2.03	2.38	2.65	2.98	3.29	3.62	4.33	5.16	6.07	7.11	8.25	9.57	11.04	12.65	14.48
2	1.93	2.27	2.52	2.85	3.14	3.47	4.18	4.99	5.89	6.92	8.07	9.38	10.85	12.46	14.28
4	1.84	2.16	2.41	2.72	3.01	3.33	4.02	4.83	5.72	6.75	7.89	9.21	10.66	12.27	14.09
8	1.76	2.06	2.29	2.60	2.89	3.20	3.88	4.68	5.57	6.60	7.73	9.05	10.50	12.10	13.93
12	1.71	2.01	2.24	2.54	2.82	3.13	3.83	4.61	5.50	6.52	7.65	8.96	10.42	12.03	13.84
24	1.64	1.93	2.15	2.45	2.72	3.03	3.70	4.48	5.37	6.39	7.52	8.83	10.28	11.90	13.70
FLAT PLATES															
Vertical	1.82	2.13	2.40	2.70	2.99	3.30	4.00	4.79	5.70	6.72	7.86	9.18	10.64	12.25	14.06
HFU	2.00	2.35	2.65	2.97	3.26	3.59	4.31	5.12	6.04	7.07	8.21	9.54	11.01	12.63	14.45
HFD	1.58	1.85	2.09	2.36	2.63	2.93	3.61	4.38	5.27	6.27	7.40	8.71	10.16	11.76	13.57

HFU, horizontal, facing upward; HFD, horizontal, facing downward.

TABLE 7.2. *Properties of steam and water*

Temp F	Press. psia	Volume, ft³/lb			Enthalpy, Btu/lb			Entropy, Btu/lb x F			Temp F
		Water v_f	Evap v_{fg}	Steam v_g	Water h_f	Evap h_{fg}	Steam h_g	Water s_f	Evap s_{fg}	Steam s_g	
32	0.08859	0.01602	3305	3305	-0.02	1075.5	1075.5	0.0000	2.1873	2.1873	32
35	0.09991	0.01602	2948	2948	3.00	1073.8	1076.8	0.0061	2.1706	2.1767	35
40	0.12163	0.01602	2446	2446	8.03	1071.0	1079.0	0.0162	2.1432	2.1594	40
45	0.14744	0.01602	2037.7	2037.8	13.04	1068.1	1081.2	0.0262	2.1164	2.1426	45
50	0.17796	0.01602	1704.8	1704.8	18.05	1065.3	1083.4	0.0361	2.0901	2.1262	50
60	0.2561	0.01603	1207.6	1207.6	28.06	1059.7	1087.7	0.0555	2.0391	2.0946	60
70	0.3629	0.01605	868.3	868.4	38.05	1054.0	1092.1	0.0745	1.9900	2.0645	70
80	0.5068	0.01607	633.3	633.3	48.04	1048.4	1096.4	0.0932	1.9426	2.0359	80
90	0.6981	0.01610	468.1	468.1	58.02	1042.7	1100.8	0.1115	1.8970	2.0086	90
100	0.9492	0.01613	350.4	350.4	68.00	1037.1	1105.1	0.1295	1.8530	1.9825	100
110	1.2750	0.01617	265.4	265.4	77.98	1031.4	1109.3	0.1472	1.8105	1.9577	110
120	1.6927	0.01620	203.25	203.26	87.97	1025.6	1113.6	0.1646	1.7693	1.9339	120
130	2.2230	0.01625	157.32	157.33	97.96	1019.8	1117.8	0.1817	1.7295	1.9112	130
140	2.8892	0.01629	122.98	123.00	107.95	1014.0	1122.0	0.1985	1.6910	1.8895	140
150	3.718	0.01634	97.05	97.07	117.95	1008.2	1126.1	0.2150	1.6536	1.8686	150
160	4.741	0.01640	77.27	77.29	127.96	1002.2	1130.2	0.2313	1.6174	1.8487	160
170	5.993	0.01645	62.04	62.06	137.97	996.2	1134.2	0.2473	1.5822	1.8295	170
180	7.511	0.01651	50.21	50.22	148.00	990.2	1138.2	0.2631	1.5480	1.8111	180
190	9.340	0.01657	40.94	40.96	158.04	984.1	1142.1	0.2787	1.5148	1.7934	190
200	11.526	0.01664	33.62	33.64	168.09	977.9	1146.0	0.2940	1.4824	1.7764	200
210	14.123	0.01671	27.80	27.82	178.15	971.6	1149.7	0.3091	1.4509	1.7600	210
212	14.696	0.01672	26.78	26.80	180.17	970.3	1150.5	0.3121	1.4447	1.7568	212
220	17.186	0.01678	23.13	23.15	188.23	965.2	1153.4	0.3241	1.4201	1.7442	220
230	20.779	0.01685	19.364	19.381	198.33	958.7	1157.1	0.3388	1.3902	1.7290	230
240	24.968	0.01693	16.304	16.321	208.45	952.1	1160.6	0.3533	1.3609	1.7142	240
250	29.825	0.01701	13.802	13.819	218.59	945.4	1164.0	0.3677	1.3323	1.7000	250
260	35.427	0.01709	11.745	11.762	228.76	938.6	1167.4	0.3819	1.3043	1.6862	260
270	41.856	0.01718	10.042	10.060	238.95	931.7	1170.6	0.3960	1.2769	1.6729	270
280	49.200	0.01726	8.627	8.644	249.17	924.6	1173.8	0.4098	1.2501	1.6599	280
290	57.550	0.01736	7.443	7.460	259.4	917.4	1176.8	0.4236	1.2238	1.6473	290
300	67.005	0.01745	6.448	6.466	269.7	910.0	1179.7	0.4372	1.1979	1.6351	300
310	77.67	0.01755	5.609	5.626	280.0	902.5	1182.5	0.4506	1.1726	1.6232	310
320	89.64	0.01766	4.896	4.914	290.4	894.8	1185.2	0.4640	1.1477	1.6116	320
340	117.99	0.01787	3.770	3.788	311.3	878.8	1190.1	0.4902	1.0990	1.5892	340
360	153.01	0.01811	2.939	2.957	332.3	862.1	1194.4	0.5161	1.0517	1.5678	360
380	195.73	0.01836	2.317	2.335	353.6	844.5	1198.0	0.5416	1.0057	1.5473	380
400	247.26	0.01864	1.8444	1.8630	375.1	825.9	1201.0	0.5667	0.9607	1.5274	400
420	308.78	0.01894	1.4808	1.4997	396.9	806.2	1203.1	0.5915	0.9165	1.5080	420
440	381.54	0.01926	1.1976	1.2169	419.0	785.4	1204.4	0.6161	0.8729	1.4890	440
460	466.9	0.0196	0.9746	0.9942	441.5	763.2	1204.8	0.6405	0.8299	1.4704	460
480	566.2	0.0200	0.7972	0.8172	464.5	739.6	1204.1	0.6648	0.7871	1.4518	480
500	680.9	0.0204	0.6545	0.6749	487.9	714.3	1202.2	0.6890	0.7443	1.4333	500
520	812.5	0.0209	0.5386	0.5596	512.0	687.0	1199.0	0.7133	0.7013	1.4146	520
540	962.8	0.0215	0.4437	0.4651	536.8	657.5	1194.3	0.7378	0.6577	1.3954	540
560	1133.4	0.0221	0.3651	0.3871	562.4	625.3	1187.7	0.7625	0.6132	1.3757	560
580	1326.2	0.0228	0.2994	0.3222	589.1	589.9	1179.0	0.7876	0.5673	1.3550	580
600	1543.2	0.0236	0.2438	0.2675	617.1	550.6	1167.7	0.8134	0.5196	1.3330	600
620	1786.9	0.0247	0.1962	0.2208	646.9	506.3	1153.2	0.8403	0.4689	1.3092	620
640	2059.9	0.0260	0.1543	0.1802	679.1	454.6	1133.7	0.8686	0.4134	1.2821	640
660	2365.7	0.0277	0.1166	0.1443	714.9	392.1	1107.0	0.8995	0.3502	1.2498	660
680	2708.6	0.0304	0.0808	0.1112	758.5	310.1	1068.5	0.9365	0.2720	1.2086	680
700	3094.3	0.0366	0.0386	0.0752	822.4	172.7	995.2	0.9901	0.1490	1.1390	700
705.5	3208.2	0.0508	0	0.0508	906.0	0	906.0	1.0612	0	1.0612	705.5

Table 7.2. *Properties of steam and water*—Continued

Press. psia	Temp F	Volume, ft^3/lb			Enthalpy, Btu/lb			Entropy, Btu/lb x F			Energy, Btu/lb		Press. psia
		Water v_f	Evap v_{fg}	Steam v_g	Water h_f	Evap h_{fg}	Steam h_g	Water s_f	Evap s_{fg}	Steam s_g	Water u_f	Steam u_g	
0.0886	32.018	0.01602	3302.4	3302.4	0.00	1075.5	1075.5	0	2.1872	2.1872	0	1021.3	0.0886
0.10	35.023	0.01602	2945.5	2945.5	3.03	1073.8	1076.8	0.0061	2.1705	2.1766	3.03	1022.3	0.10
0.15	45.453	0.01602	2004.7	2004.7	13.50	1067.9	1081.4	0.0271	2.1140	2.1411	13.50	1025.7	0.15
0.20	53.160	0.01603	1526.3	1526.3	21.22	1063.5	1084.7	0.0422	2.0738	2.1160	21.22	1028.3	0.20
0.30	64.484	0.01604	1039.7	1039.7	32.54	1057.1	1089.7	0.0641	2.0168	2.0809	32.54	1032.0	0.30
0.40	72.869	0.01606	792.0	792.1	40.92	1052.4	1093.3	0.0799	1.9762	2.0562	40.92	1034.7	0.40
0.5	79.586	0.01607	641.5	641.5	47.62	1048.6	1096.3	0.0925	1.9446	2.0370	47.62	1036.9	0.5
0.6	85.218	0.01609	540.0	540.1	53.25	1045.5	1098.7	0.1028	1.9186	2.0215	53.24	1038.7	0.6
0.7	90.09	0.01610	466.93	466.94	58.10	1042.7	1100.8	0.3	1.8966	2.0083	58.10	1040.3	0.7
0.8	94.38	0.01611	411.67	411.69	62.39	1040.3	1102.6	0.1117	1.8775	1.9970	62.39	1041.7	0.8
0.9	98.24	0.01612	368.41	368.43	66.24	1038.1	1104.3	0.1264	1.8606	1.9870	66.24	1042.9	0.9
1.0	101.74	0.01614	333.59	333.60	69.73	1036.1	1105.8	0.1326	1.8455	1.9781	69.73	1044.1	1.0
2.0	126.07	0.01623	173.74	173.76	94.03	1022.1	1116.2	0.1750	1.7450	1.9200	94.03	1051.8	2.0
3.0	141.47	0.01630	118.71	118.73	109.42	1013.2	1122.6	0.2009	1.6854	1.8864	109.41	1056.7	3.0
4.0	152.96	0.01636	90.63	90.64	120.92	1006.4	1127.3	0.2199	1.6428	1.8626	120.90	1060.2	4.0
5.0	162.24	0.01641	73.515	73.53	130.20	1000.9	1131.1	0.2349	1.6094	1.8443	130.18	1063.1	5.0
6.0	170.05	0.01645	61.967	61.98	138.03	996.2	1134.2	0.2474	1.5820	1.8294	138.01	1065.4	6.0
7.0	176.84	0.01649	53.634	53.65	144.83	992.1	1136.9	0.2581	1.5587	1.8168	144.81	1067.4	7.0
8.0	182.86	0.01653	47.328	47.35	150.87	988.5	1139.3	0.2676	1.5384	1.8060	150.84	1069.2	8.0
9.0	188.27	0.01656	42.385	42.40	156.30	985.1	1141.4	0.2760	1.5204	1.7964	156.28	1070.8	9.0
10	193.21	0.01659	38.404	38.42	161.26	982.1	1143.3	0.2836	1.5043	1.7879	161.23	1072.3	10
14.696	212.00	0.01672	26.782	26.80	180.17	970.3	1150.5	0.3121	1.4447	1.7568	180.12	1077.6	14.696
15	213.03	0.01673	26.274	26.29	181.21	969.7	1150.9	0.3137	1.4415	1.7552	181.16	1077.9	15
20	227.96	0.01683	20.070	20.087	196.27	960.1	1156.3	0.3358	1.3962	1.7320	196.21	1082.0	20
30	250.34	0.01701	13.7266	13.744	218.9	945.2	1164.1	0.3682	1.3313	1.6995	218.8	1087.9	30
40	267.25	0.01715	10.4794	10.497	236.1	933.6	1169.8	0.3921	1.2844	1.6765	236.0	1092.1	40
50	281.02	0.01727	8.4967	8.514	250.2	923.9	1174.1	0.4112	1.2474	1.6586	250.1	1095.3	50
60	292.71	0.01738	7.1562	7.174	262.2	915.4	1177.6	0.4273	1.2167	1.6440	262.0	1098.0	60
70	302.93	0.01748	6.1875	6.205	272.7	907.8	1180.6	0.4411	1.1905	1.6316	272.5	1100.2	70
80	312.04	0.01757	5.4536	5.471	282.1	900.9	1183.1	0.4534	1.1675	1.6208	281.9	1102.1	80
90	320.28	0.01766	4.8777	4.895	290.7	894.6	1185.3	0.4643	1.1470	1.6113	290.4	1103.7	90
100	327.82	0.01774	4.4133	4.431	298.5	888.6	1187.2	0.4743	1.1284	1.6027	298.2	1105.2	100
120	341.27	0.01789	3.7097	3.728	312.6	877.8	1190.4	0.4919	1.0960	1.5879	312.2	1107.6	120
140	353.04	0.01803	3.2010	3.219	325.0	868.0	1193.0	0.5071	1.0681	1.5752	324.5	1109.6	140
160	363.55	0.01815	2.8155	2.834	336.1	859.0	1195.1	0.5206	1.0435	1.5641	335.5	1111.2	160
180	373.08	0.01827	2.5129	2.531	346.2	850.7	1196.9	0.5328	1.0215	1.5543	345.6	1112.5	180
200	381.80	0.01839	2.2689	2.287	355.5	842.8	1198.3	0.5438	1.0016	1.5454	354.8	1113.7	200
250	400.97	0.01865	1.8245	1.8432	376.1	825.0	1201.1	0.5679	0.9585	1.5264	375.3	1115.8	250
300	417.35	0.01889	1.5238	1.5427	394.0	808.9	1202.9	0.5882	0.9223	1.5105	392.9	1117.2	300
350	431.73	0.01913	1.3064	1.3255	409.8	794.2	1204.0	0.6059	0.8909	1.4968	408.6	1118.1	350
400	444.60	0.0193	1.14162	1.1610	424.2	780.4	1204.6	0.6217	0.8630	1.4847	422.7	1118.7	400
450	456.28	0.0195	1.01224	1.0318	437.3	767.5	1204.8	0.6360	0.8378	1.4738	435.7	1118.9	450
500	467.01	0.0198	0.90787	0.9276	449.5	755.1	1204.7	0.6490	0.8148	1.4639	447.7	1118.8	500
550	476.94	0.0199	0.82183	0.8418	460.9	743.3	1204.3	0.6611	0.7936	1.4547	458.9	1118.6	550
600	486.20	0.0201	0.74962	0.7698	471.7	732.0	1203.7	0.6723	0.7738	1.4461	469.5	1118.2	600
700	503.08	0.0205	0.63505	0.6556	491.6	710.2	1201.8	0.6928	0.7377	1.4304	488.9	1116.9	700
800	518.21	0.0209	0.54809	0.5690	509.8	689.6	1199.4	0.7111	0.7051	1.4163	506.7	1115.2	800
900	531.95	0.0212	0.47968	0.5009	526.7	669.7	1196.4	0.7279	0.6753	1.4032	523.2	1113.0	900
1000	544.58	0.0216	0.42436	0.4460	542.6	650.4	1192.9	0.7434	0.6476	1.3910	538.6	1110.4	1000
1100	556.28	0.0220	0.37863	0.4006	557.5	631.5	1189.1	0.7578	0.6216	1.3794	553.1	1107.5	1100
1200	567.19	0.0223	0.34013	0.3625	571.9	613.0	1184.8	0.7714	0.5969	1.3683	566.9	1104.3	1200
1300	577.42	0.0227	0.30722	0.3299	585.6	594.6	1180.2	0.7843	0.5733	1.3577	580.1	1100.9	1300
1400	587.07	0.0231	0.27871	0.3018	598.8	576.5	1175.3	0.7966	0.5507	1.3474	592.9	1097.1	1400
1500	596.20	0.0235	0.25372	0.2772	611.7	558.4	1170.1	0.8085	0.5288	1.3373	605.2	1093.1	1500
2000	635.80	0.0257	0.16266	0.1883	672.1	466.2	1138.3	0.8625	0.4256	1.2881	662.6	1068.6	2000
2500	668.11	0.0286	0.10209	0.1307	731.7	361.6	1093.3	0.9139	0.3206	1.2345	718.5	1032.9	2500
3000	695.33	0.0343	0.05073	0.0850	801.8	218.4	1020.3	0.9728	0.1891	1.1619	782.8	973.1	3000
3208.2	705.47	0.0508	0	0.0508	906.0	0	906.0	1.0612	0	1.0612	875.9	875.9	3208.2

Source: *Thermodynamic and Transport Properties of Steam,* 1967; reprinted with permission of the American Society of Mechanical Engineers, New York, N.Y.

TABLE 7.2. *Properties of steam and water*—Continued

Abs press. lb/sq in. (sat. temp)		Temperature, F														
		100	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500
	v	0.0161	392.5	452.3	511.9	571.5	631.1	690.7								
1	*h*	68.00	1150.2	1195.7	1241.8	1288.6	1336.1	1384.5								
(101.74)	*s*	0.1295	2.0509	2.1152	2.1722	2.2237	2.2708	2.3144								
	v	0.0161	78.14	90.24	102.24	114.21	126.15	138.08	150.01	161.94	173.86	185.78	197.70	209.62	221.53	233.45
5	*h*	68.01	1148.6	1194.8	1241.3	1288.2	1335.9	1384.3	1433.6	1483.7	1534.7	1586.7	1639.6	1693.3	1748.0	1803.5
(162.24)	*s*	0.1295	1.8716	1.9369	1.9943	2.0460	2.0932	2.1369	2.1776	2.2159	2.2521	2.2866	2.3194	2.3509	2.3811	2.4101
	v	0.0161	38.84	44.98	51.03	57.04	63.03	69.00	74.98	80.94	86.91	92.87	98.84	104.80	110.76	116.72
10	*h*	68.02	1146.6	1193.7	1240.6	1287.8	1335.5	1384.0	1433.4	1483.5	1534.6	1586.6	1639.5	1693.3	1747.9	1803.4
(193.21)	*s*	0.1295	1.7928	1.8593	1.9173	1.9692	2.0166	2.0603	2.1011	2.1394	2.1757	2.2101	2.2430	2.2744	2.3046	2.3337
	v	0.0161	0.0166	29.899	33.963	37.985	41.986	45.978	49.964	53.946	57.926	61.905	65.882	69.858	73.833	77.807
15	*h*	68.04	168.09	1192.5	1239.9	1287.3	1335.2	1383.8	1433.2	1483.4	1534.5	1586.5	1639.4	1693.2	1747.8	1803.4
(213.03)	*s*	0.1295	0.2940	1.8134	1.8720	1.9242	1.9717	2.0155	2.0563	2.0946	2.1309	2.1653	2.1982	2.2297	2.2599	2.2890
	v	0.0161	0.0166	22.356	25.428	28.457	31.466	34.465	37.458	40.447	43.435	46.420	49.405	52.388	55.370	58.352
20	*h*	68.05	168.11	1191.4	1239.2	1286.9	1334.9	1383.5	1432.9	1483.2	1534.3	1586.3	1639.3	1693.1	1747.8	1803.3
(227.96)	*s*	0.1295	0.2940	1.7805	1.8397	1.8921	1.9397	1.9836	2.0244	2.0628	2.0991	2.1336	2.1665	2.1979	2.2282	2.2572
	v	0.0161	0.0166	11.036	12.624	14.165	15.685	17.195	18.699	20.199	21.697	23.194	24.689	26.183	27.676	29.168
40	*h*	68.10	168.15	1186.6	1236.4	1285.0	1333.6	1382.5	1432.1	1482.5	1533.7	1585.8	1638.8	1992.7	1747.5	1803.0
(267.25)	*s*	0.1295	0.2940	1.6992	1.7608	1.8143	1.8624	1.9065	1.9476	1.9860	2.0224	2.0569	2.0899	2.1224	2.1516	2.1807
	v	0.0161	0.0166	7.257	8.354	9.400	10.425	11.438	12.446	13.450	14.452	15.452	16.450	17.448	18.445	19.441
60	*h*	68.15	168.20	1181.6	1233.5	1283.2	1332.3	1381.5	1431.3	1481.8	1533.2	1585.3	1638.4	1692.4	1747.1	1802.8
(292.71)	*s*	0.1295	0.2939	1.6492	1.7134	1.7681	1.8168	1.8612	1.9024	1.9410	1.9774	2.0120	2.0450	2.0765	2.1068	2.1359
	v	0.0161	0.0166	0.0175	6.218	7.018	7.794	8.560	9.319	10.075	10.829	11.581	12.331	13.081	13.829	14.577
80	*h*	68.21	168.24	269.74	1230.5	1281.3	1330.9	1380.5	1430.5	1481.1	1532.6	1584.9	1638.0	1692.0	1746.8	1802.5
(312.04)	*s*	0.1295	0.2939	0.4371	1.6790	1.7349	1.7842	1.8289	1.8702	1.9089	1.9454	1.9800	2.0131	2.0446	2.0750	2.1041
	v	0.0161	0.0166	0.0175	4.935	5.588	6.216	6.833	7.443	8.050	8.655	9.258	9.860	10.460	11.060	11.659
100	*h*	68.26	168.29	269.77	1227.4	1279.3	1329.6	1379.5	1429.7	1480.4	1532.0	1584.4	1637.6	1691.6	1746.5	1802.2
(327.82)	*s*	0.1295	0.2939	0.4371	1.6516	1.7088	1.7586	1.8036	1.8451	1.8839	1.9205	1.9552	1.9883	2.0199	2.0502	2.0794
	v	0.0161	0.0166	0.0175	4.0786	4.6341	5.1637	5.6831	6.1928	6.7006	7.2060	7.7096	8.2119	8.7130	9.2134	9.7130
120	*h*	68.31	168.33	269.81	1224.1	1277.4	1328.1	1378.4	1428.8	1479.8	1531.4	1583.9	1637.1	1691.3	1746.2	1802.0
(341.27)	*s*	0.1295	0.2939	0.4371	1.6286	1.6872	1.7376	1.7829	1.8246	1.8635	1.9001	1.9349	1.9680	1.9996	2.0300	2.0592
	v	0.0161	0.0166	0.0175	3.4661	3.9526	4.4119	4.8585	5.2995	5.7364	6.1709	6.6036	7.0349	7.4652	7.8946	8.3233
140	*h*	68.37	168.38	269.85	1220.8	1275.3	1326.8	1377.4	1428.0	1479.1	1530.8	1583.4	1636.7	1690.9	1745.9	1801.7
(353.04)	*s*	0.1295	0.2939	0.4370	1.6085	1.6686	1.7196	1.7652	1.8071	1.8461	1.8828	1.9176	1.9508	1.9825	2.0129	2.0421
	v	0.0161	0.0166	0.0175	3.0060	3.4413	3.8480	4.2420	4.6295	5.0132	5.3945	5.7741	6.1522	6.5293	6.9055	7.2811
160	*h*	68.42	168.42	269.89	1217.4	1273.3	1325.4	1376.4	1427.2	1478.4	1530.3	1582.9	1636.3	1690.5	1745.6	1801.4
(363.55)	*s*	0.1294	0.2938	0.4370	1.5906	1.6522	1.7039	1.7499	1.7919	1.8310	1.8678	1.9027	1.9359	1.9676	1.9980	2.0273
	v	0.0161	0.0166	0.0174	2.6474	3.0433	3.4093	3.7621	4.1084	4.4505	4.7907	5.1289	5.4657	5.8014	6.1363	6.4704
180	*h*	68.47	168.47	269.92	1213.8	1271.2	1324.0	1375.3	1426.3	1477.7	1529.7	1582.4	1635.9	1690.2	1745.3	1801.2
(373.08)	*s*	0.1294	0.2938	0.4370	1.5743	1.6376	1.6900	1.7362	1.7784	1.8176	1.8545	1.8894	1.9227	1.9545	1.9849	2.0142
	v	0.0161	0.0166	0.0174	2.3598	2.7247	3.0583	3.3783	3.6915	4.0008	4.3077	4.6128	4.9165	5.2191	5.5209	5.8219
200	*h*	68.52	168.51	269.96	1210.1	1269.0	1322.6	1374.3	1425.5	1477.0	1529.1	1581.9	1635.4	1689.8	1745.0	1800.9
(381.80)	*s*	0.1294	0.2938	0.4369	1.5593	1.6242	1.6776	1.7239	1.7663	1.8057	1.8426	1.8776	1.9109	1.9427	1.9732	2.0025
	v	0.0161	0.0166	0.0174	0.0186	2.1504	2.4662	2.6872	2.9410	3.1909	3.4382	3.6837	3.9278	4.1709	4.4131	4.6546
250	*h*	68.66	168.63	270.05	375.10	1263.5	1319.0	1371.6	1423.4	1475.3	1527.6	1580.6	1634.4	1688.9	1744.2	1800.2
(400.97)	*s*	0.1294	0.2937	0.4368	0.5667	1.5951	1.6502	1.6976	1.7405	1.7801	1.8173	1.8524	1.8858	1.9177	1.9482	1.9776
	v	0.0161	0.0166	0.0174	0.0186	1.7665	2.0044	2.2263	2.4407	2.6509	2.8585	3.0643	3.2688	3.4721	3.6746	3.8764
300	*h*	68.79	168.74	270.14	375.15	1257.7	1315.2	1368.9	1421.3	1473.6	1526.2	1579.4	1633.3	1688.0	1743.4	1799.6
(417.35)	*s*	0.1294	0.2937	0.4307	0.5665	1.5703	1.6274	1.6758	1.7192	1.7591	1.7964	1.8317	1.8652	1.8972	1.9278	1.9572
	v	0.0161	0.0166	0.0174	0.0186	1.4913	1.7028	1.8970	2.0832	2.2652	2.4445	2.6219	2.7980	2.9730	3.1471	3.3205
350	*h*	68.92	168.85	270.24	375.21	1251.5	1311.4	1366.2	1419.2	1471.8	1524.7	1578.2	1632.3	1687.1	1742.6	1798.9
(431.73)	*s*	0.1293	0.2936	0.4367	0.5664	1.5483	1.6077	1.6571	1.7009	1.7411	1.7787	1.8141	1.8477	1.8798	1.9105	1.9400
	v	0.0161	0.0166	0.0174	0.0162	1.2841	1.4763	1.6499	1.8151	1.9759	2.1339	2.2901	2.4450	2.5987	2.7515	2.9037
400	*h*	69.05	168.97	270.33	375.27	1245.1	1307.4	1363.4	1417.0	1470.1	1523.3	1576.9	1631.2	1686.2	1741.9	1798.2
(444.60)	*s*	0.1293	0.2935	0.4366	0.5663	1.5282	1.5901	1.6406	1.6850	1.7255	1.7632	1.7988	1.8325	1.8647	1.8955	1.9250
	v	0.0161	0.0166	0.0174	0.0186	0.9919	1.1584	1.3037	1.4397	1.5708	1.6992	1.8256	1.9507	2.0746	2.1977	2.3200
500	*h*	69.32	169.19	270.51	375.38	1231.2	1299.1	1357.7	1412.7	1466.6	1520.3	1574.4	1629.1	1684.4	1740.3	1796.9
(467.01)	*s*	0.1292	0.2934	0.4364	0.5660	1.4921	1.5595	1.6123	1.6578	1.6990	1.7371	1.7730	1.8069	1.8393	1.8702	1.8998

TABLE 7.2. *Properties of steam and water*—Continued

Abs press. lb/sq in. (sat. temp)		Temperature, F														
		100	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500
600 (486.20)	v	0.0161	0.0166	0.0174	0.0186	0.7944	0.9456	1.0726	1.1892	1.3008	1.4093	1.5160	1.6211	1.7252	1.8284	1.9309
	h	69.58	169.42	270.70	375.49	1215.9	1290.3	1351.8	1408.3	1463.0	1517.4	1571.9	1627.0	1682.6	1738.8	1795.6
	s	0.1292	0.2933	0.4362	0.5657	1.4590	1.5329	1.5844	1.6351	1.6769	1.7155	1.7517	1.7859	1.8184	1.8494	1.8792
700 (503.08)	v	0.0161	0.0166	0.0174	0.0186	0.0204	0.7928	0.9072	1.0102	1.1078	1.2023	1.2948	1.3858	1.4757	1.5647	1.6530
	h	69.84	169.65	270.89	375.61	487.93	1281.0	1345.6	1403.7	1459.4	1514.4	1569.4	1624.8	1680.7	1737.2	1794.3
	s	0.1291	0.2932	0.4360	0.5655	0.6889	1.5090	1.5673	1.6154	1.6580	1.6970	1.7335	1.7679	1.8006	1.8318	1.8617
800 (518.21)	v	0.0161	0.0166	0.0174	0.0186	0.0204	0.6774	0.7828	0.8759	0.9631	1.0470	1.1289	1.2093	1.2885	1.3669	1.4446
	h	70.11	169.88	271.07	375.73	487.88	1271.1	1339.2	1399.1	1455.8	1511.4	1566.9	1622.7	1678.9	1735.0	1792.9
	s	0.1290	0.2930	0.4358	0.5652	0.6885	1.4869	1.5484	1.5980	1.6413	1.6807	1.7175	1.7522	1.7851	1.8164	1.8464
900 (531.95)	v	0.0161	0.0166	0.0174	0.0186	0.0204	0.5869	0.6858	0.7713	0.8504	0.9262	0.9998	1.0720	1.1430	1.2131	1.2825
	h	70.37	170.10	271.26	375.84	487.83	1260.6	1332.7	1394.4	1452.2	1508.5	1564.4	1620.6	1677.1	1734.1	1791.6
	s	0.1290	0.2929	0.4357	0.5649	0.6881	1.4659	1.5311	1.5822	1.6263	1.6662	1.7033	1.7382	1.7713	1.8028	1.8329
1000 (544.58)	v	0.0161	0.0166	0.0174	0.0186	0.0204	0.5137	0.6080	0.6875	0.7603	0.8295	0.8966	0.9622	1.0266	1.0901	1.1529
	h	70.63	170.33	271.44	375.96	487.79	1249.3	1325.9	1389.6	1448.5	1504.4	1561.9	1618.4	1675.3	1732.5	1790.3
	s	0.1289	0.2928	0.4355	0.5647	0.6876	1.4457	1.5149	1.5677	1.6126	1.6530	1.6905	1.7256	1.7589	1.7905	1.8207
1100 (556.28)	v	0.0161	0.0166	0.0174	0.0185	0.0203	0.4531	0.5440	0.6188	0.6865	0.7505	0.8121	0.8723	0.9313	0.9894	1.0468
	h	70.90	170.56	271.63	376.08	487.75	1237.3	1318.8	1384.7	1444.7	1502.4	1559.4	1616.3	1673.5	1731.0	1789.0
	s	0.1289	0.2927	0.4353	0.5644	0.6872	1.4259	1.4996	1.5542	1.6000	1.6410	1.6787	1.7141	1.7475	1.7793	1.8097
1200 (567.19)	v	0.0161	0.0166	0.0174	0.0185	0.0203	0.4016	0.4905	0.5615	0.6250	0.6845	0.7418	0.7974	0.8519	0.9055	0.9584
	h	71.16	170.78	271.82	376.20	487.72	1224.2	1311.5	1379.7	1440.9	1449.4	1556.9	1614.2	1671.6	1729.4	1787.6
	s	0.1288	0.2926	0.4351	0.5642	0.6868	1.4061	1.4851	1.5415	1.5883	1.6298	1.6679	1.7035	1.7371	1.7691	1.7996
1400 (587.07)	v	0.0161	0.0166	0.0174	0.0185	0.0203	0.3176	0.4059	0.4712	0.5282	0.5809	0.6311	0.6798	0.7272	0.7737	0.8195
	h	71.68	171.24	272.19	376.44	487.65	1194.1	1296.1	1369.3	1433.2	1493.2	1551.8	1609.9	1668.0	1726.3	1785.0
	s	0.1287	0.2923	0.4348	0.5636	0.6859	1.3652	1.4575	1.5182	1.5670	1.6096	1.6484	1.6845	1.7185	1.7508	1.7815
1600 (604.87)	v	0.0161	0.0166	0.0173	0.0185	0.0202	0.0236	0.3415	0.4032	0.4555	0.5031	0.5482	0.5915	0.6336	0.6748	0.7153
	h	72.21	171.69	272.57	376.69	487.60	616.77	1279.4	1358.5	1425.2	1486.9	1546.6	1605.6	1664.3	1723.2	1782.3
	s	0.1286	0.2921	0.4344	0.5631	0.6851	0.8129	1.4312	1.4968	1.5478	1.5916	1.6312	1.6678	1.7022	1.7344	1.7657
1800 (621.02)	v	0.0160	0.0165	0.0173	0.0185	0.0202	0.0235	0.2906	0.3500	0.3988	0.4426	0.4836	0.5229	0.5609	0.5980	0.6343
	h	72.73	172.15	272.95	376.93	487.56	615.58	1261.1	1347.2	1417.1	1480.6	1541.1	1601.2	1660.7	1720.1	1779.7
	s	0.1284	0.2918	0.4341	0.5626	0.6843	0.8109	1.4054	1.4768	1.5302	1.5753	1.6156	1.6528	1.6876	1.7204	1.7516
2000 (635.80)	v	0.0160	0.0165	0.0173	0.0184	0.0201	0.0233	0.2488	0.3072	0.3534	0.3942	0.4320	0.4680	0.5027	0.5365	0.5695
	h	73.26	172.60	273.32	377.19	487.53	614.48	1240.9	1353.4	1408.7	1447.1	1536.2	1596.9	1657.0	1717.0	1777.1
	s	0.1283	0.2916	0.4337	0.5621	0.6834	0.8091	1.3794	1.4578	1.5138	1.5603	1.6014	1.6391	1.6743	1.7075	1.7389
2500 (668.11)	v	0.0160	0.0165	0.0173	0.0184	0.0200	0.0230	0.1681	0.2293	0.2712	0.3068	0.3390	0.3692	0.3980	0.4259	0.4529
	h	74.57	173.74	274.27	377.82	487.50	612.08	1176.7	1303.4	1386.7	1457.5	1522.9	1585.9	1647.8	1709.2	1770.4
	s	0.1280	0.2910	0.4329	0.5609	0.6815	0.8048	1.3076	1.4129	1.4766	1.5269	1.5703	1.6094	1.6456	1.6796	1.7116
3000 (695.33)	v	0.0160	0.0165	0.0172	0.0183	0.0200	0.0228	0.0982	0.1759	0.2161	0.2484	0.2770	0.3033	0.3282	0.3522	0.3753
	h	75.88	174.88	275.22	378.47	487.52	610.08	1060.5	1267.0	1363.2	1440.2	1509.4	1574.8	1638.5	1701.4	1761.8
	s	0.1277	0.2904	0.4320	0.5597	0.6796	0.8009	1.1966	1.3692	1.4429	1.4976	1.5434	1.5841	1.6214	1.6561	1.6888
3200 (705.08)	v	0.0160	0.0165	0.0172	0.0183	0.0199	0.0227	0.0335	0.1588	0.1987	0.2301	0.2576	0.2827	0.3065	0.3291	0.3510
	h	76.4	175.3	275.6	378.7	487.5	609.4	800.8	1250.9	1353.4	1433.1	1503.8	1570.3	1634.8	1698.3	1761.2
	s	0.1276	0.2902	0.4317	0.5592	0.6788	0.7994	0.9708	1.3515	1.4300	1.4866	1.5335	1.5749	1.6126	1.6477	1.6806
3500	v	0.0160	0.0164	0.0172	0.0183	0.0199	0.0225	0.0307	0.1364	0.1764	0.2066	0.2326	0.2563	0.2784	0.2995	0.3198
	h	77.2	176.0	276.2	379.1	487.6	608.4	779.4	1224.6	1338.2	1422.2	1495.5	1563.3	1629.2	1693.6	1757.2
	s	0.1274	0.2899	0.4312	0.5585	0.6777	0.7973	0.9508	1.3242	1.4112	1.4709	1.5194	1.5618	1.6002	1.6358	1.6691
4000	v	0.0159	0.0164	0.0172	0.0182	0.0198	0.0223	0.0287	0.1052	0.1463	0.1752	0.1994	0.2210	0.2411	0.2601	0.2783
	h	78.5	177.2	277.1	379.8	487.7	606.9	763.0	1174.3	1311.6	1403.6	1481.3	1552.2	1619.8	1685.7	1750.6
	s	0.1271	0.2893	0.4304	0.5573	0.6760	0.7940	0.9343	1.2754	1.3807	1.4461	1.4976	1.5417	1.5812	1.6177	1.6516
5000	v	0.0159	0.0164	0.0171	0.0181	0.0196	0.0219	0.0268	0.0591	0.1038	0.1312	0.1529	0.1718	0.1890	0.2050	0.2203
	h	81.1	179.5	279.1	381.2	488.1	604.6	746.0	1042.9	1252.9	1364.6	1452.1	1529.1	1600.9	1670.0	1737.4
	s	0.1265	0.2881	0.4287	0.5550	0.6726	0.7880	0.9153	1.1593	1.3207	1.4001	1.4582	1.5061	1.5481	1.5863	1.6216
6000	v	0.0159	0.0163	0.0170	0.0180	0.0195	0.0216	0.0256	0.0397	0.0757	0.1020	0.1221	0.1391	0.1544	0.1684	0.1817
	h	83.7	181.7	281.0	382.7	488.6	602.9	736.1	945.1	1188.8	1323.6	1422.3	1505.9	1582.0	1654.2	1724.2
	s	0.1258	0.2870	0.4271	0.5528	0.6693	0.7826	0.9026	1.0176	1.2615	1.3574	1.4229	1.4748	1.5194	1.5593	1.5962
7000	v	0.0158	0.0163	0.0170	0.0180	0.0193	0.0213	0.0248	0.0334	0.0573	0.0816	0.1004	0.1160	0.1298	0.1424	0.1542
	h	86.2	184.4	283.0	384.2	489.3	601.7	729.3	901.8	1124.9	1281.7	1392.2	1482.6	1563.1	1638.6	1711.1
	s	0.1252	0.2859	0.4256	0.5507	0.6663	0.7777	0.8926	1.0350	1.2055	1.3171	1.3904	1.4466	1.4938	1.5355	1.5735

Table 7.3 Mean Apparent Specific Heats of Steels

(Btu/lb F)

Composition	Temperature ranges, F 122 to 212	302 to 392	392 to 482	482 to 572	572 to 662	662 to 752	842 to 932	1022 to 1112	1202 to 1292	1292 to 1382	1382 to 1472	1562 to 1652
Pure iron	0.112	0.122	0.126	0.130	0.135	0.140	0.155	0.175	0.198	0.232	0.218	0.1
Carbon steels												
0.60 C, 0.38 Mn	.115	.124	.128	.132	.137	.142	.158	.180	.207	.264	.209	.232
0.08 C, 0.31 Mn	.115	.125	.130	.133	.136	.142	.158	.177	.205	.272	.229	.195
0.23 C, 0.635 Mn	.116	.124	.127	.133	.137	.143	.158	.179	.202	.342	.227	...
0.415 C, 0.643 Mn	.116	.123	.126	.131	.136	.140	.155	.169	.184	.378	.149	.131
0.80 C, 0.32 Mn	.117	.127	.131	.135	.140	.145	.160	.170	.184	.497	.147	...
1.22 C, 0.35 Mn	.116	.129	.130	.133	.138	.143	.152	.167	.195	.499	.155	...
Alloy steels												
0.23 C, 1.51 Mn, 0.105 Cu	.114	.122	.126	.130	.135	.141	.155	.177	.200	.346	.196	.128
0.325 C, 0.55 Mn, 0.17 Cr, 3.47 Ni*	.115	.125	.128	.131	.136	.141	.158	.179	.391	.228	.144	.153
0.33 C, 0.53 Mn, 0.80 Cr, 3.38 Ni*	.118	.125	.129	.134	.139	.143	.161	.185	.312	.281	.133	.139
0.325 C, 0.55 Mn, 0.71 Cr, 3.41 Ni*	.117	.125	.128	.131	.136	.142	.159	.182	.274	.330	.136	.137
0.34 C, 0.55 Mn, 0.78 Cr, 3.53 Ni, 0.39 Mo*	.116	.125	.129	.133	.139	.145	.160	.184	.251	.397	.152	.152
0.315 C, 0.69 Mn, 1.09 Cr, 0.073 Ni	.118	.125	.128	.132	.137	.142	.157	.177	.200	.358	.223	.137
0.35 C, 0.59 Mn, 0.88 Cr, 0.26 Ni, 0.20 Mo	.114	.123	.126	.130	.136	.142	.157	.176	.197	.386	.211	...
0.485 C, 0.90 Mn, 1.98 Si, 0.637 Cu	.119	.125	.129	.133	.138	.144	.159	.179	.198	.216	.326	...
High-alloy steels												
1.22 C, 13.00 Mn, 0.22 Si*	.124	.133	.136	.139	.141	.143	.138	...	...	...	...	...
0.28 C, 0.89 Mn, 28.37 Ni*	.119	.120	.119	.119	.118	.117	.124	.130	.123	.124	.123	.126
0.08 C, 0.37 Mn, 19.11 Cr, 8.14 Ni*	.122	.127	.129	.131	.134	.136	.142	.155	.149	.150	.153	.153
0.13 C, 0.25 Mn, 12.95 Cr*	.113	.123	.127	.132	.138	.145	.163	.186	.209	.216	.165	.160
0.27 C, 0.28 Mn, 13.69 Cr*	.113	.122	.127	.131	.136	.143	.162	.186	.221	.237	.187	.157
0.715 C, 0.25 Mn, 4.26 Cr, 18.45 W, 1.075 V*	0.098	0.104	0.108	0.111	0.116	0.120	0.132	0.143	0.152	0.171	0.171	0.176

* Steels heated discontinuously; all others heated at steady rate of about 5.4 F per min.

Table 7.4 Physical Properties of Fuel Oil at 60 F

Fuel oil (CS-12-48) Grade No.	Gravity, API	Sp gr	Lb per gal	Btu per lb	Net Btu per gal
6	3	1.0520	8.76	18,190	152,100
6	4	1.0443	8.69	18,240	151,300
6	5	1.0366	8.63	18,290	149,400
6	6	1.0291	8.57	18,340	148,800
6	7	1.0217	8.50	18,390	148,100
6	8	1.0143	8.44	18,440	147,500
6	9	1.0071	8.39	18,490	146,900
6	10	1.0000	8.33	18,540	146,200
6	11	0.9930	8.27	18,590	145,600
6	12	.9861	8.22	18,640	144,900
6, 5	14	.9725	8.10	18,740	143,600
6, 5	16	.9593	7.99	18,840	142,300
5	18	.9465	7.89	18,930	140,900
4, 5	20	.9340	7.78	19,020	139,600
4, 5	22	.9218	7.68	19,110	138,300
4, 5	24	.9100	7.58	19,190	137,100
4, 2	26	.8984	7.49	19,270	135,800
4, 2	28	.8871	7.39	19,350	134,600
2	30	.8762	7.30	19,420	133,300
2	32	.8654	7.21	19,490	132,100
2	34	.8550	7.12	19,560	130,900
1, 2	36	.8448	7.04	19,620	129,700
1, 2	38	.8348	6.96	19,680	128,500
1	40	.8251	6.87	19,750	127,300
1	42	0.8156	6.79	19,810	126,200

The relation between specific gravity and degrees API is expressed by the formula:

$$\frac{141.5}{131.5 + \text{API}} = \text{sp gr at 60 F.}$$

For each 10 F above 60 F add 0.7 API.
For each 10 F below 60 F subtract 9.7 AP

Table 7.5 Typical Properties of Commercial Petroleum Products Sold in the East and Midwest

	Gasolines		Premium Fuels			Fuel Oils				
	12 psia Reid vapor pressure natural gasoline	Straight run gasoline	No. 1 fuel, light diesel, or stove oil	Diesel or "gas house" gas oil	Reduced crude, heavy gas oil or premium residuum*	No. 2	No. 3	Cold No. 5	No. 5	Bunker C or No. 6
Gravity, °API	79	63	38–42	35–38	19–22	32–35	28–32	20–25	16–24	6–14
Viscosity	...	...	34–36†	34–36†	15–50‡	34–36†	35–45†	< 20‡	20–40‡	50–300‡
Conradson carbon, wt %	None	None	Trace	Trace	3–7	Trace	<0.15	1–2	2–4	6–15
Pour point, °F	<0	<0	<0	<10	...	<10	<20	<15	15–60	>50
Sulfur, wt %	<0.1	<0.2	<0.1	0.1–0.6	<2.0	0.1–0.6	0.2–1.0	0.5–2.0	0.5–2.0	1–4
Water and sed., wt %	None	None	Trace	Trace	<1.0	<0.05	<0.10	0.1–0.5	0.5–1.0	0.5–2.0
Distillation, °F										
10%	110	150	390–410	400–440	550–650	420–440	450–500	<600	<600	600–700
50%	150	230	440–460	490–510	800–900	490–530	540–560	...	...	850–950
90%	235	335	490–520	580–600	...	580–620	600–670	<700	>700	...
End point	315	370	510–560	630–660	...	630–660	650–700	...	...	...
Flash point, °F	<100	<100	100–140	110–170	>150	130–170	>130	>130	>130	>150
Aniline point, °F	145	125	140–160	150–170	...	120–140	120–140	...	...	...
C to H ratio	5.2	5.6	6.1–6.4	6.2–6.5	6.9–7.3	6.6–7.1	7.0–7.3	7.3–7.7	7.3–7.7	7.7–9.0
Average MBtu recovered/gal of oil:										
Carbureted water gas	101	101	102	101	93	91	86	82	83	74
High-Btu oil gas	90	89	90	90	82	80	76	72	73	64
Tar + carbon, wt % of oil										
Carbureted water gas	...	...	20	22	35	31	36	42	42	52
High-Btu oil gas	...	...	32	34	45	42	46	52	52	60

(Data on gasolines and data below C/H ratio not given in reports.)

* The 6 wt % Conradson carbon oil used in the Hall High Btu Oil Gas Tests (A.G.A. Gas Production Research Committee. Hall High Btu Oil Gas Process, New York, 1949.) and "New England Gas Enriching Oil" are typical examples of this group.

† Saybolt Universal seconds at 100 F.

‡ Saybolt Furol seconds at 122 F.

Remarks:

No. 1 Fuel Oil—A distillate oil intended for vaporizing pot-type burners. A volatile fuel.
No. 2 Fuel Oil—For general purpose domestic heating; for use in burners not requiring No. 1 oil. Moderately volatile.
No. 3 Fuel Oil—Formerly a distillate oil for use in burners requiring low viscosity oil. Now incorporated as a part of No. 2.
No. 4 Fuel Oil—For burner installations not equipped with preheating facilities.
No. 5 Fuel Oil—A residual type oil. Requires preheating to 170–220 F.
No. 6 Fuel Oil—Preheating to 220–260 F suggested. A high viscosity oil.

Table 7.6 Gas Constants and Volume of the Pound-Mol for Certain Gases

	$R' = R/M$, specific gas constant, ft per F	M, mol wt, lb per lb-mol	R, universal gas constant, ft-lb per (lb-mol)(F)	Mv, cu ft per lb-mol *
Hydrogen	767.04	2.016	1546	378.9
Oxygen	48.24	32.000	1544	378.2
Nitrogen	55.13	28.016	1545	378.3
Nitrogen, "atmospheric"†	54.85	28.161	1545	378.6
Air	53.33	28.966	1545	378.5
Water vapor	85.72	18.016	1544	378.6
Carbon dioxide	34.87	44.010	1535	376.2
Carbon monoxide	55.14	28.010	1544	378.3
Hydrogen sulfide	44.79	34.076	1526	374.1
Sulfur dioxide	23.56	64.060	1509	369.6
Ammonia	89.42	17.032	1523	373.5
Methane	96.18	16.042	1543	378.2
Ethane	50.82	30.068	1528	374.5
Propane	34.13	44.094	1505	368.7
n-Butane	25.57	58.120	1486	364.3
iso-Butane	25.79	58.120	1499	367.4
Ethylene	54.70	28.052	1534	376.2
Propylene	36.01	42.078	1515	379.1

* At 60 F, 30 in. Hg, dry.

† Includes other inert gases in trace amounts.

Table 7.7 *Conversion table to SI units*

The policy of the National Bureau of Standards is to encourage and lead in national use of the metric system, formally called the International System of Units (SI). This publication uses customary English units, however, for the convenience of engineers and others who use them habitually. The reader interested in conversion to SI units is referred to:

(1) NBS SP 330, 1974 Edition, "The International System of Units".

(2) American National Standard for Metric Practice, Z210.1–1976 (ASTM E380–76 and IEEE Std 268–1976).

The following table shows conversion factors for the units used in this handbook.

Quantity	To convert from	To	Multiply by
Length	inch	meter (m)	2.540×10^{-2}
	foot	m	3.048×10^{-1}
	mile	m	1.609×10^{3}
Area	sq in	m^3	6.452×10^{-4}
	sq ft	m^2	9.290×10^{-2}
Volume	cu in	m^3	1.639×10^{-5}
	cu ft	m^3	2.832×10^{-2}
	gallon	m^3	3.785×10^{-3}
Temperature	°F	°C	$t_C = (t_F - 32)/1.8$
Temp. difference	Δt_F	ΔT_K	1/1.8
Mass	pound	kg	4.536×10^{-1}
	ounce	kg	2.835×10^{-2}
Pressure	psi	Pa	6.895×10^{3}
	in of water	Pa	2.488×10^{2}
	in of Hg	Pa	3.386×10^{3}
	mm Hg	Pa	1.333×10^{2}
Energy	Btu	J	1.055×10^{3}
	MBtu	J	1.055×10^{9}
	kWh	J	3.600×10^{6}
	ft lb	J	1.356×10^{0}
Power	Btu/h	W	2.931×10^{-1}
	hp	W	7.460×10^{2}
Flow	gpm	m^3/s	6.309×10^{-5}
	cfm	m^3/s	4.719×10^{-4}
Density	lb/cu ft	kg/m^3	1.602×10^{1}
	lb/gal	kg/m^3	1.198×10^{2}
Heat capacity	Btu/lb·°F	J/(kg·K)	4.187×10^{3}
	Btu/cu ft·°F	$J/(m^3 \cdot K)$	6.707×10^{4}
Conductivity	Btu·in/h·sq ft·°F	W/(m·K)	1.442×10^{-1}
Heat of combustion	Btu/gal	J/m^3	2.787×10^{5}
	Btu/lb	J/kg	2.326×10^{3}
	Btu/cu ft	J/m^3	3.726×10^{4}
Barrel (Petroleum)	42 gal	m^3	1.590×10^{-1}

Chapter 8

SOURCES OF ASSISTANCE FOR DESIGNING AND INSTALLING WASTE HEAT SYSTEMS

Page 197 The sixth entry in the list should read:

American Gas Association
1515 Wilson Blvd.
Arlington, VA 22209

Just as there are many applications for waste heat recovery systems and many types of systems, there are many sources of assistance. One of the best sources in the initial stages for determining the possibilities and feasibility of a project are consulting energy-use engineers whom you may retain to assist you in making a survey of your plant. If you are not aware of specific consulting engineers in your area, the energy suppliers may be able to suggest two or three from which to choose. In some locations there are professional societies that have local energy conservation or efficient energy utilization committees that may have a list of consultants from which to select. A list of some organizations and their national office addresses are listed below:

American Society of Heating Refrigerating and Air Conditioning Engineers, Inc.
United Engineering Center
345 East 47th Street
New York, NY 10017

American Society for Engineering Education
One Dupont Circle
Washington, DC 20036

Institute of Electrical and Electronics Engineers, Inc.
United Engineering Center
345 East 47th Street
New York, NY 10017

American Consulting Engineers Council
1555 15th Street, N.W.
Washington, DC 20005

National Society of Professional Engineers
2029 K Street, N.W.
Washington, DC 20006

American Gas Association
1515 Wilson Boulevard
Arlington, VA 22209

Edison Electric Institute
90 Park Avenue
New York, NY 10016

An energy consultant should be able to help you in identifying the best cost/effectiveness ideas and major possibilities for energy reduction. Where desired he may carry the responsibility for you in designing and supervising the changes required to implement waste heat utilization.

Another source of assistance is the equipment manufacturers and their representatives. Generally, a wealth of information regarding application details, approximate costs, installation procedures and other equipment changes is available. Quite often they will have recent case histories similar to the one applicable to your plant.

The responsibility for a project that involves two or more suppliers of equipment for an existing process is sometimes difficult to assign without a consulting engineer. On some projects such as the installation of a large recuperator on an existing furnace the major equipment supplier may take responsibility for the other changes.

APPENDIX

Index of Notation

a Cross sectional area (ft^2)

A End-of-period payment

A Surface area (ft^2)

A_w Surface area calculated using

$D = \frac{D_1 + D_2}{2}$ (ft^2)

B Availability

c Specific heat (Btu/lb•F)

c_p Specific heat at constant pressure (Btu/lb•F) or (Btu/ft^3•F)

c_{pm} Average specific heat for a multicomponent gas (Btu/lb•F) or (Btu/ft^3•F)

D Diameter (ft)

D_e Equivalent diameter, $4d/p$ (ft)

e Escalation rate of price

e Absolute roughness (ft)

E Expected cost

F Fahrenheit temperature (F)

F Future sum of money

F_a Radiation configuration factor

F_e Radiation surface factor

f Friction factor

g Acceleration of gravity (ft/s^2)

g_c Gravitational constant ($lb \cdot ft/lb_f \cdot s^{-}$)

Gr Grashof number

h Enthalpy per unit mass

h_c Convection coefficient (Btu/h•ft^2•F)

h_{cr} Combined convection, radiation coefficient (Btu/h•ft^2•F)

h_r Radiation coefficient (Btu/h•ft^2•F)

H Enthalpy

H_f Pressure drop (in of H_2O)

H_{sd} Stack draft (in of H_2O)

i Discount rate

J Flow rate

K_L Flow loss coefficient

k Thermal conductivity (Btu•ft/h•ft^2•F)

lb/in^2 Pounds per square inch

lb/in^2a Pounds per square inch (absolute)

lb/in^2g Pounds per square inch (gage)

L Length (ft)

L Radiant mean beam length (ft)

M Mass (lb)

M Molecular weight

m Mass flow rate

NTU_o Modified overall heat transfer unit

N Number of interest periods

NPV Net present value

P Present sum of money

p Absolute pressure (lb_f/ft^2)

PP Payback period (yr)

q Heat transfer rate (Btu/h)

Q Heat

R Absolute Fahrenheit temperature (R)

Re Reynolds number

ROI Return on Investment

s Entropy per unit mass

S Entropy

T Temperature

T_{abs} Absolute temperature

T_b Black body temperature

T_m Mean temperature

u Internal energy per unit mass

U Internal energy

U Overall heat transfer coefficient

v Specific volume (ft^3/lb)

V Average velocity (ft/s)

V Volume

W Work

x Coordinate direction (ft)

X Percent by volume of gas component

z Vertical coordinate or distance in vertical direction (ft)

Greek Symbols

α Absorptivity

β Volumetric coefficient of expansion, $1/R$

Δ Difference between two values

ϵ Heat transfer effectiveness

μ Efficiency

ρ Density (lb/ft^3)

σ Stefan-Boltzmann constant, 0.1714 ($Btu/h{\cdot}ft^2{\cdot}R^4$)

Σ Summation over all components

Subscripts

a Ambient conditions

c Cold fluid

f Pertaining to liquids

g Pertaining to a gas or a gas component

h Hot fluid

m Mean value

o Reference state

s Surface

s Shell side

t Tube side

w Wall

w Property pertaining to water vapor

∞ Bulk temperature

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