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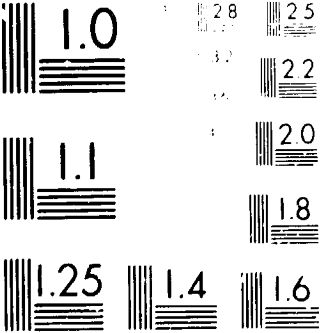
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→ Marie Dietrich

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WASTE HEAT

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976

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W A S T E   H E A T

/Basic principles for heat recovery and utilization applicable in bricks, refractories, structural and fine ceramics industries/

by: M.Němeček

Waste Heat

1. Instroduction

Silicate industries, like the others ones, depend on abundance of thermal energy contained first of all in fosile fuels. The low price of manufactured silicate products depends on low price fuel used in silicate technologies. Thus consumers must respect increasing energy and fuel prices. It leads to conclusion, that industrial conservation and recovering of energy is extremely important.

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 lesson 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 further.

One of the benefits of the energy conservation activity 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, the necessary 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 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 lesson 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. Therefore, the message of this lesson is saving energy. 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.

The problem addressed in this lesson 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.

Anybody 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 price 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 lesson is to help the engineer of manager learn about available options in waste heat recovery, to decide whether he can lower costs or increase profits by installing waste heat equipment, and to help him choose the best available equipment for this purpose.

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.

2. Waste Heat

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.

The essential quality of heat addressed here is not amount, but value. This distinction becomes apparent when one compares the value of recovering similar amounts of heat at 100°C or 550°C, 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 550°C is quite useful; the same heat in a dirty flow at 100°C might not be worth bothering with, and certainly would be much harder to use.

Sources of waste heat energy can be divided according to temperature into three temperature ranges. The high temperature range refers to temperatures above 650°C. The medium temperature range is between 230°C and 650°C and the low temperature range is below 230°C.

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.

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 1650°C, while measured flame temperatures in practical combustors are just under 1650°C. 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.

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

Glass melting furnace	1000 - 1600°C
Cement kiln /Dry process/	630 - 730°C
Open hearth furnace	650 - 750°C

The next list 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 100 to 200 kPa. Steam can be generated at almost any desired pressure and steam turbines used when economical/.

Steam boiler exhausts	250 - 500°C
Gas turbine exhausts	370 - 540°C
Heat treating furnaces	420 - 650°C
Drying and baking ovens	250 - 600°C
Annealing furnace cooling systems	430 - 650°C

Next table 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.

Process steam condensate	55 - 90°C
Cooling water from:	
Furnace doors	30 - 55°C
Bearings	30 - 90°C

Welding machines	30 - 90°C
Injection molding machines	30 - 90°C
Annealing furnaces	65 - 230°C
Forming dies	25 - 90°C
Air compressors	25 - 50°C
Pumps	25 - 90°C
Internal combustion engines	65 - 120°C
Air conditioning and refrigeration condensers	30 - 45°C
Drying, baking and curing ovens	195 - 230°C
Hot processed solids	195 - 230°C

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 30°C to heat the domestic water supply. Since the hot water must be heated to about 70°C, obviously the air conditioner waste heat is not hot enough. However, since the cold water enters the domestic water system at about 10°C, energy interchange can take place raising the water to something less than 30°C. 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.

3. Uses of 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.

This equipment that is used to recover waste heat can range from something as simple as a pipe, a furnace channel or duct to something as complex as a waste heat boiler.

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

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.

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.

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.

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.

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.

Waste Heat Management

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.

Analysis of the Plant
/Aspects A 1/

Analysis of Heat
Sources in the Plant
/Aspects A 2/

Programme of Waste
Heat Recovery
/Aspects A 3/

In the Plant
Inside

In the Plant
Outside

Project of Waste Heat
Recovery Equipment
/Aspects A 4/

Realisation of the
Waste
Heat Recovery
Equipment

Liste of Aspects

A 1:

- energetic efficiency
- commercial prospects of production
- season/uninterrupted service
- commercial effect of heat recovery
- other significant effects /social, human/ecological, etc./

A 2:

- industry branch
- structure of technology
- climate
- use of the recovered waste heat
- degree of automation by the installed equipment
- skill of the working staff
- other factors: technical, commercial, economic, human/ecological
- social/political importance

A 3:

- varieties of waste heat recovery devices
- possibilities of energy conservation by improvements in technology and/or equipments
- expenses for engineering studies
- limited expenses for investment of heat recovery equipment
- possibility to apply waste heat utilization projects to other similary plants

A 4:

- skill of own man-power
- competence of domestic research centres to solve heat recovery tasks
- competence of domestic producers to manufacture and to deliver heat recovery equipments
- investment sources
- commercial advantage of purchase heat recovery equipment

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 scheme pp.7,8 is listed a survey path containing the important information needed to obtain a over-all look at the problem of heat recovery. Then the flow chart for the process and its heat balances must be studied to determine where opportunities for waste heat recovery exist. Also the results of engineering and economic studies for each process unit have to be 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.

4. Theoretical Requirements of Waste Heat Management

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 150°C cannot be used directly to heat steam to 200°C. 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 /heat balance/ and energy quality and its availability to be transferred /heat transfer/. The fifth factor, the economic waste heat use is treated in an other lesson and the first one, the use for the waste heat, in chapter 6.

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. The measurement 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 voltage of the electrical energy entering /unless kilowatt-hour meters are used/. All the testing routines 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.

$$H_{los} = h_{cr}A(T_s - T_o$$

where

H_{los} - rate of heat loss in energy units W
h_{cr} - heat transfer coefficient in W/m^2K
A - area of surface losing heat in m^2

T_s - surface temperature K
T_o - ambient temperature K

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, such as infrared thermography, thermovision sets and other modern methods, 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.

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:

air input rate
exhaust gas flow rate
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.

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 90°C 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.

Thus the heat balance of a surveyed plant may be expressed by equation, valid within a time unit as follows:

$$H_{in} + W_{in} - H_{los} - H_{out} - W_{out} = B \qquad /1/$$

The equation is based upon the law of conservation of energy in forms of enthalpy H and exerted technical work W. Here we have neglected several energy terms such as kinetic energy, magnetic or electromagnetic energy, which need not be considered for simple applications in energy balances of furnaces, driers and other devices used in silicate industry. Symbols represent following terms of that simple heat balance:

H_{in} kW the sum of input enthalpies, as for the enthalpy of pushing in material, and other media, chemical energy of fossil fuels, energy of electric heat, enthalpy of combusting air and so on,

W_{in} kW the sum of technical work inserted by mechanical part of devices or, rarely, exerted by thermodynamic cycles upon gases , vapors or liquids in the surveyed plant

H_{los} kW heat losses as enthalpy lost to the ambient environment by heat transfer on surfaces.

H_{out} kW the sum of output enthalpies flowing out to the ambient environment within pushing out material, and technologic aids, flue gases, evaporated vapors or steam and other media.

W_{out} kW   the sum of technical work performed as energy output into ambient environment

B kW   the sum of energy spent for material transformation and the enthalpy

$$H_a = \frac{1}{t} \int_{o}^{t} \frac{dH_a}{dt}\, dt$$

accumulated within the time unit in materials, walls and other mass belonging to the surveyed plant. When the plant operates in steady state, the term $\frac{dH_a}{dt}$ of heat accumulation in B is zero.

If write equation /1/ over in the form when on its left hand side is $H_{in} + W_{in}$ only, divide it by $H_{in} + W_{in}$ and multiply it by 100, we get

$$100 = {}_pH_{los} + {}_pH_{out} + {}_pW_{out} + {}_pB \qquad /2/$$

where index p denctes the percent parts of the heat balance. Then, for example, the sum ${}_pH_{out} + {}_sW_{out}$ means one kind of energetic efficiency /in percent/ of the plant.

There may be a practical representations of energy balance, as shown in figures 1 and 2

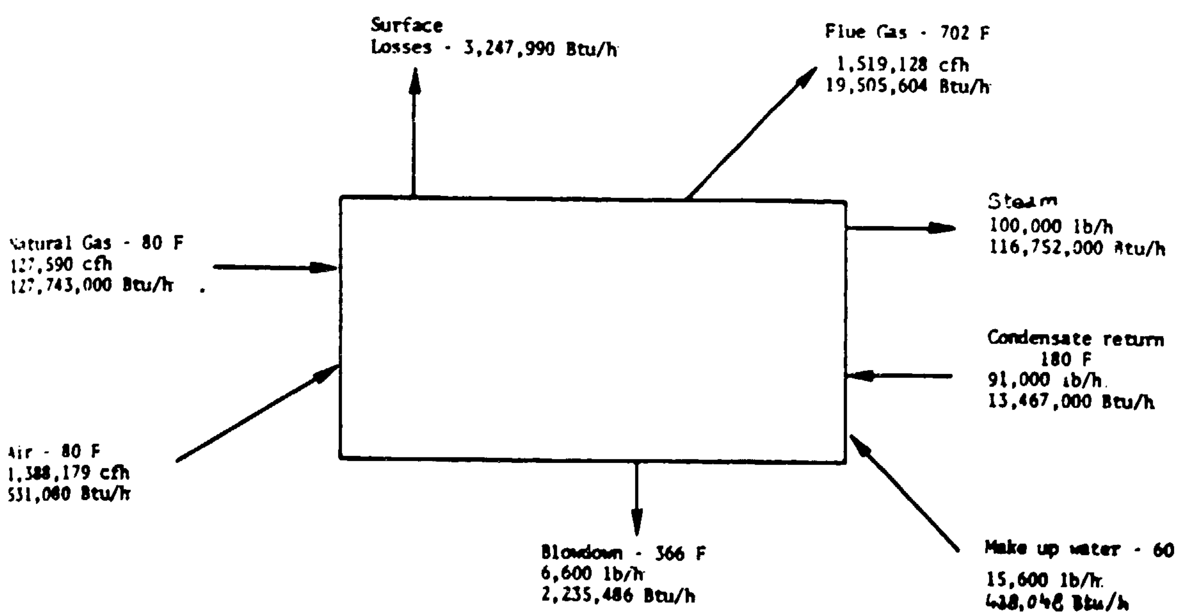

Figure 1. Simplest diagramme of heat balance of a steam generator burning natural gas with 10 percent excess air

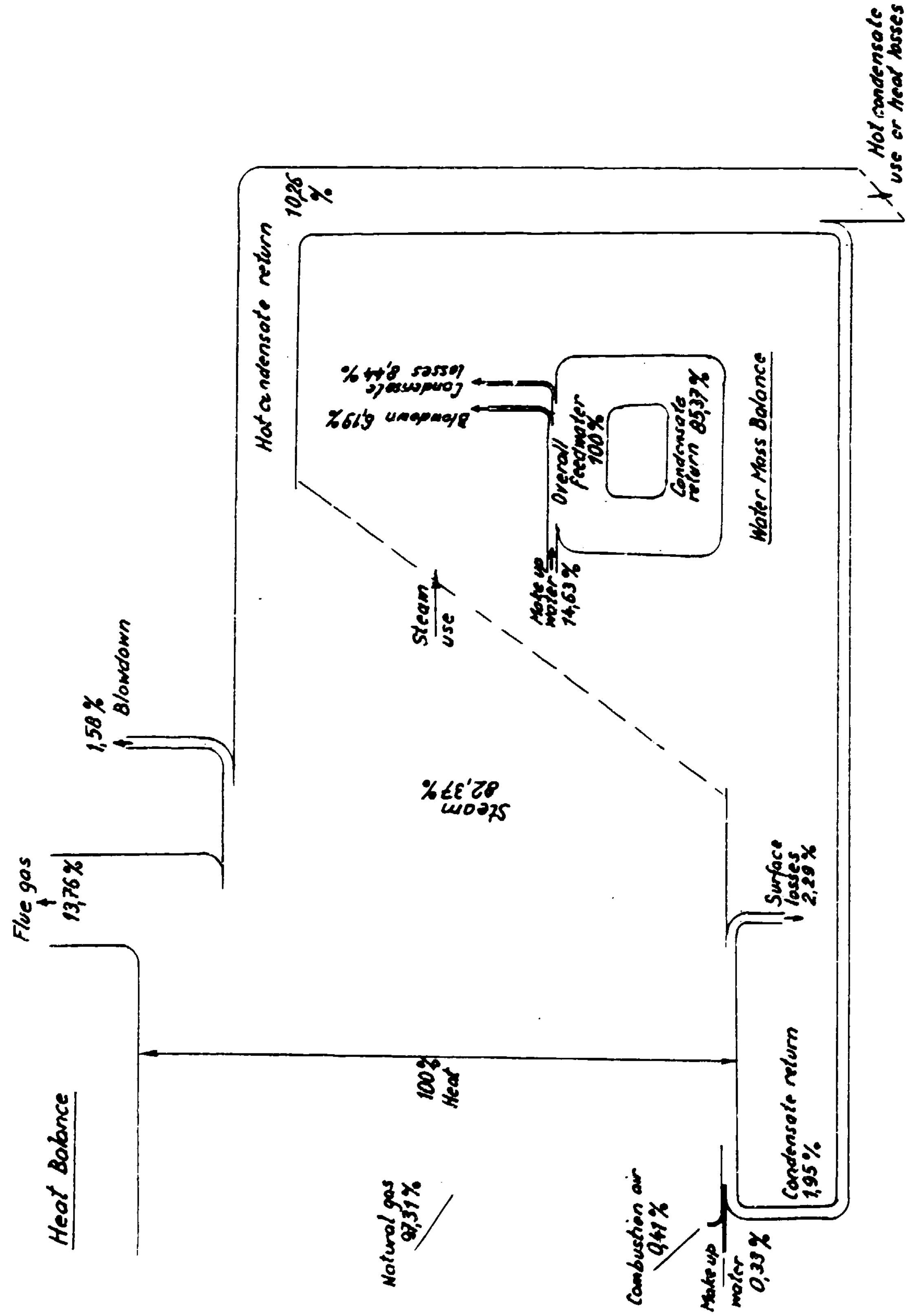

Figure 2. Other type of diagramme of heat balance of the same steam generator as in fig.1, expressing the back flows

Note that the diagrammes may include both the quantity and quality of balanced heat.

Heat Transfer

Up to this point, we have treated the estimation of the quantity of waste heat /heat balance/ and quality of waste heat; now we shall consider briefly heat transfer, which is the necessary step in transferring the waste heat from the flow in which it is not 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 this chapter, and heat transfer is treated in references. For the reader's convenience a few points are mentioned here. Heat exchangers are divided into three broad group. 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. 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 /see also the other lesson of this meeting/. 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 other lesson.

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:

$$q = -kA\, dT/dx \qquad /3/$$

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:

$$q = h_c A (T_s - T_f) \qquad /4/$$

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:

$$q = \sigma F_e F_s A (T^4_{abs\ 1} - T^4_{abs\ 2}) \qquad /5/$$

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 $5{,}6688.10^{-8}$ W/m^2K^4. Although radiation is usually associated with solid surface, 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

$$q = h_r A \left(T_1 - T_2 \right) \qquad /6/$$

where by comparison to eq./3/ the coefficient of radiation, h_r, is defined as

$$h_r = \frac{\sigma F_e F_s A \left(T^4_{abs\ 1} - T^4_{abs\ 2} \right)}{T_1 - T_2} \qquad /7/$$

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 equation /7/ the rate of heat flow by radiation is a function of the temperature difference and can be aditively 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 lesson. For more detailed information on this subject the reader is referred to references in appendix. For a brief information some formulas for approximate estimation of coefficients of convective heat transfer can be given /see next paragraph/.

Estimation of convective heat transfer coefficients

To anticipate characteristic properties of projected heat exchangers, the estimation of heat transfer coefficients has to be made. The convective heat transfer plays a substantial role in most types of heat exchangers. Let us show some examples of formulas /obtained most frequently by experimental way/ advisable for calculating heat transfer coefficients h_c. In all the formulas this coefficient h_c is expressed in the dimensionless form of Nusselt number

$$Nu = \frac{h_c \cdot l}{k} \qquad /8/$$

where l is characteristic geometric dimension and
k is the thermal conductivity coefficient of fluid at the temperature in prescribed point of fluid flow.

As characteristic dimension l for forced or free convection it has to be taken: for tube the inner /outer/ diameter d for flow through /outside/ of tube /cylinder/ respectively; this same is valid for tube bundles; for noncircular cylindrical body it is the transverse dimension in the cross-section of the body perpendicular to the flow. For flow over a plane surface which is heated or cooled the characteristic dimension has to be taken as the distance from the leading edge of the plane. For flow perpendicular to plane of a heated horizontal plate /free convection/ has to be taken the mean or less dimension of the plate. The identical choose of characteristic dimension is valid for Reynolds, Peclet and Grashof numbers. For flow over various types of finned tubes the characteristic dimension is the minimum hydraulic radius, defined by $l = \frac{4f}{\sigma}$ has to be taken. Here f is the minimal free area of the cross-section between two neighbouring finned tubes, σ is the wetted perimeter of this area.

Forced convection in tubes

$$Nu = 0{,}023\ Re^{0,8} Pr^{n} \qquad /9/$$

n = 0,4 for heating, n = 0,3 for cooling. The equation is valid for variety of usual gases and liquids. The physical properties are taken at the bulk temperature except the viscosity in the Reynolds group which is evaluated at the mean film temperature $\frac{1}{2}(T_{surface} + T_{bulk\ fluid})$. With very viscous liquids there will be a marked difference at any cross-section between the viscosity μ_s of the fluid adjacent to the surface and the value μ at the axis or at the bulk temperature of the fluid. It was presented a modified formula including a viscosity correction term

$$Nu = 0{,}027\ Re^{0{,}8} Pr^{0{,}33} \left(\frac{\mu}{\mu_s}\right)^{0{,}14} \qquad /10/$$

For gases with Prandtl group about 0,74 after substituting this value into eq./9/ we get

$$Nu = 0{,}02\ Re^{0{,}8} \qquad /11/$$

For water which is very frequently used as the cooling liquid the equation /9/ becomes

$$h = 1063\ (1+0{,}00293\ T)\ u^{0{,}8} d^{-0{,}2} \qquad Wm^{-2}K^{-1} \qquad /12/$$

if $T\ (K)$, $d\ (m)$, $u\ (ms^{-1})$.

At low Reynolds number $/Re < 10^4/$ the value Nu depends on the length l of the entrance region of the tube and it can be corrected by special formula or diagramme. The formula can be taken

$$Nu = 1{,}62 \cdot Pe \left(\frac{d}{l_c}\right)^{1/3} \qquad /13/$$

The temperature difference for heat transfer is taken as the arithmetic mean of the terminal values, i.e. $(T_w - T_1) + (T_w - T_2) \cdot \frac{1}{2}$ where T_w is the temperature of the tube, which is taken as constant. For viscous liquids /viscous oils/ the experimental values of h are greater then those given by equation /13/. This is due to the large variation of viscosity with temperature. With correction introduced /like it was made for turbulent flow/ we can use the equation

$$Nu = 1{,}86 \cdot RePr \left(\frac{d}{l_e}\right)^{1/3} \left(\frac{\mu}{\mu_s}\right)^{0{,}14} \qquad /14/$$

For forced convection in a circular tube with laminar flow also the formula

$$\overline{Nu} = 1,61 \left(RePr \frac{d}{l_e} \right)^{1/3} \qquad /15/$$

is valid. Here $\overline{Nu}$ is the mean value of Nusselt number on the length l_e , i.e.

$$\overline{Nu} = \frac{1}{l_e} \int_0^{l_e} Nu \, dl_e$$

and the condition $Re < 2200$ has to be performed.
Forced convection across a single cylinder or tube in temperature range to 1073 K with Reynolds numbers from 10^3 to 10^5:

$$Nu = 0,26 \; Re^{0,6} Pr^{0,3} \qquad /16/$$

For gases /taking Pr as 0,74/ this reduces to

$$Nu = 0,24 \; Re^{0.6} \qquad /16\ a/$$

For very low values of Re /from 0,2 to 2.10^2/ with liquids the experimental data are better approximated by

$$Nu = 0,86 \; Re^{0,43} Pr^{0,3} \qquad /17/$$

In each of eq. /14/ - /17/ the physical properties are ment at the mean film temperature T taken as the average of the surface temperature T_w and the mean fluid temperature

$$T_m = \frac{T_1 + T_2}{2} \qquad /17\ a/$$

Forced convection at right angles to tube bundles is designated by dependence heat transfer on geometrical arrangement of tubes /in-line or staggered/ and on the number of rows.

$$Nu = 0,33 \; C_h Re_{max}^{0,6} \; Pr^{0,3} \qquad /18/$$

where C_h is the correction factor tabelled in specialized guide books. For example, in the first row the heat transfer coefficient is higher then in the next ones, in more then 8^{th} row is quite stabilized; in bundles with staggered tubes in higher then in bundles with in-line ones. Reynolds number Re_{max} is based on the maximum velocity through the bundle, the characteristic geometric dimension is the outside tube diameter. Convection heat transfer in rectangular channels: For turbulent flow /Re > 10^4/ is recommended the equation /9/. The characteristic dimension to use is the hydraulic mean diameter $d_e = \frac{4a.b}{2\ a+b} = 2\ \frac{ab}{a+b}$ where a,b : the sides of the channel cross-section. For flat channels /if b >>a/ $d_e \doteq 2a$, where a is the less internal dimension of the channel.

For annular section between concentric tubes the hydraulic mean diameter $d_e = d_2 - d_1$ recommended. For viscous region in ducts of non-circular sections formula /14/ or /15/, under using the hydraulic mean diameter d_e, is acceptable.

Convection heat transfer on a plane plate, parallel to fluid flow depends on character of the film flow. If boundary layer is turbulent /it is for gases often fulfilled/, two equations are useful:

$$Nu = 0{,}029\ Re^{0{,}8} Pr^{0{,}4} \qquad /19\ a/$$

$$\overline{Nu} = 0{,}036\ Re^{0{,}8} Pr^{0{,}4} \qquad /19\ b/$$

where Nu, $\overline{Nu}$ are the local and mean Nusselt number respectively. Characteristic dimension is the distance x from the leading edge of the plate. Note, that the transferred heat must be calculated by integration due to Nu → ∞ at the distance.

Convection heat transfer to spherical particles:

$$Nu = 2 + \beta''\ Re^n Pr^m \qquad /20/$$

where values of β'', n and m are listed in tables in dependence on Re and Pr. The value 2 represents the pure conduction heat transfer to sphere in quiet liquid or gas, equation /20/ is valid for forced convection. If the influence of natural convection takes place, equation of $Nu = \Psi(Re, Pr, Gr)$ - type is recommended. For tubes the eq./9/ is modified to equation

$$\overline{Nu} = 0{,}74\ Re^{0{,}2}\ (\overline{Gr\ Pr})^{0{,}1}\ \overline{Pr}^{0{,}2} \qquad /21/$$

$$1 > 50\ d$$

for horizontal tubes and channels

note, that the effect of forced and natural convection cannot be added. Natural convection is dependent on buoyancy effect and the heat transfer would be expected to follow a relation

$$Nu = f\,(Gr.Pr) \qquad /22/$$

For a wide spektrum of shapes of bodies exchanging heat with liquids or gases the equation

$$Nu = C'(Gr \cdot Pr)^n \qquad /23/$$

is recommended.

Values of C' lists the next table:

Nature of surface	Characteristic dimension	C' Gr.Pr range $<2.10^9$ $n=0,25$	$>2.10^9$ $n=0,33$
Horizontal or vertical cylinders	Diameter	0,47	0,10
Vertical planes or vertical cylinders of large diameter	Heigh	0,56	0,12
Horizontal planes facing upwards	Mean length of side	0,54	0,14
Horizontal planes facing downwards	Mean length of side	0,25	not reached

Fluid layers of thickness x between two surfaces transfer heat from a hot $/T_1/$ surface across to a parallel cold $/T_2/$ one. Natural convection in a fluid layer causes raising thermal conductivity of fluid, so that the actual heat $/Q_k/$ transferred is given by relations

$$Q = Q_k \quad /\text{for } (Gr.Pr) \leq 10^3/$$

$$Q = Q_k .0,15\,(Gr \cdot Pr)^{0,25} \quad /\text{for } 10^4 < Gr.Pr < 10^6/ \qquad /24/$$

$$Q = Q_k .0,15\,(Gr.Pr)^{1/3} \quad /\text{for } Gr.Pr > 10^6/ \qquad /25/$$

where Q_k is the rate at which heat would be transferred by pure thermal conduction between the surfaces,

$$Q = \frac{k}{x}(T_1 - T_2) = \frac{k}{x}\,\Delta T \qquad /26/$$

The characteristic dimension to be used for the Grashof number is x /distance between the planes/, so that

$$\frac{Q}{Q_k} = \frac{h.\Delta T}{\frac{k}{x}\,\Delta T} = \frac{hx}{k} \qquad /26\ a/$$

corresponds with Nusselt number.
Heat transfer coefficients to condensing and boiling water:
The description of heat transfer coefficients for condensation or boiling by formula is much more complicated than this for convection, due to great number of impacting factors.

This is the reason, that here only few approximate data may be presented.

Especially, in case of condensation, the heat transfer coefficient is removed and replaced by heat flux density $U_o \Delta T$ on the steam-side surface of a given apparatus - a steam condenser. The transferred heat may be expressed by equation

$$Q_c = U_o.\Delta T.F$$

where ΔT is temperature difference between cooling water and tube wall, F is the outside tube surface.
As most illustrative the two equations can be shown:

$$U_o = \left(0{,}00522 + \frac{1}{47{,}2.\left(\frac{U'}{0{,}305}\right)^{0{,}8}}\right)^{-1} \; Wm^{-2}K^{-1} \qquad /27\ a/$$

for the old tube, and

$$U_o = \left(0{,}00227 + \frac{1}{47{,}2.\left(\frac{U'}{0{,}305}\right)^{0{,}8}}\right)^{-1} \; Wm^{-2}K^{-1} \qquad /27\ b/$$

for the clean tube of condenser, where U' ms^{-1} is the velocity of cooling water inside the tube. The more precise values have to be listed from specialized guidebooks for detailed defined apparatus.

Analogous situation is in estimation of heat transfer coefficient for boiling water or other liquids. The best appreciation is to be taken from firms manufacturing steam boilers. The most illustrative value characterizing intensity of boiling is the specific heat flux q Wm^2 transferred to boiling liquid, which is generally very high /about of order 10^6 Wm^{-2}/. The most essential is also the fact, that the temperature difference between the wall and condensing saturated steam or boiling water is very little. It is very difficult to measure it precisely and to determin the heat transfer coefficient.

Mass transfer

In the air conditioning and in the cooling-tower practice is well known the combined heat and mass trabsfer between air and water streams or droplets. By immediate contact the air is saturated by water vapor and the water evaporates. The latent heat of vaporization is taken off from the air, which is cooled. The water is cooled too. The more detailed insight in this phenomenon is to get in specialized guide-books.

Finned surfaces

In order to increase heat transfer amount which depends on the product h.F there are two alternative ways: to raise the velocity of fluid or to enlarge the heat transfer surface F. In gaseous fluids in which the heat transfer coefficients have little values, appart from raising gas velocity, the enlarging of surface F by arranging fins is advantageous. Due to lot of shapes of fins there are many formulas to calculate heat exchange coefficient which are treated in specialized books and experimental reports.

Pressure drop

Both internal and external convective heat and heat-mass transfer is connected with momentum and energy losses, which are manifested by pressure drop in the heat exchange device, i.e. negative pressure difference between input and output of the device. Pressure drop represents costs, which results from price of power spent by pumps, fans, exhaustors, blowers compressors and other machines required to force the fluid through heat exchange device. It exist many formulas to predict pressure drop for variously shaped active parts of heat exchangers and to complex them into the summary pressure loss of an exchanger so that it is impossible to refer them here in detail. Let us show, however, the only some substantial equations and relations as example. The two general formulas for a pressure drop $\Delta p = \Delta p_t + \Delta p_s$ in a fluid duct are as follows:

$$\Delta p_t = \sum_{m=1}^{M} f_m \cdot \frac{l_m}{d_m} \frac{U_m^2}{2} \cdot \rho_m \qquad \mathrm{Pam}^{-2} \qquad /28/$$

for tubes, and

$$\Delta p_s = \sum_{n=1}^{N} C_n \frac{U_n^2}{2} \cdot \rho_n \qquad \mathrm{Pam}^{-2} \qquad /29/$$

for fittings

where are: f_m a friction coefficient depending of Reynolds number of a flow in the tu' or a fitting

l_m m the length of pipeline

$d_{m,n}$ m the hydraulic mean diameter of a duct /or inner diameter when a circular tube/

$U_{m,n}$ ms^{-1} the characteristic fluid velocity /i.e. mean volumetric velocity/

$\rho_{m,n}$ kgm^{-3} fluid density

m is the index of the pipeline length division

C_n thw coefficient of discharge of duct fittings /i.e. sudden enlargement, sudden contraction, diffusor, confusor, elbow, valve, coupling, etc./ depending on Reynolds number

n index of the tube fittings

The characteristic Reynolds number for a fitting is defined by use of the fluid mean velocity in a prescribed duct section. The formulas given above are then used to estimate and predict pressure drop of a heat exchanger.

5. 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.

If heat transfer coefficients of fluid and thermal conductivity of the wall if it is arranged between both fluids are known, the heat exchange in some heat exchangers can be calculated by use of the equation

$$Q = K.F.\theta \qquad /30/$$

where Q is the exchanged heat flux W
F reference surface of heat exchange m^2
θ mean temperature difference K , and
K is the overall heat transfer coefficient $Wm^{-2}K^{-1}$.

It is familiar, that 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 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.

Here the term fluid is used in the most general sense. The terms of evaporator and condenser imply, that the phase changes occur in fluids. 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 cross-flow type.

The reasons for separating the two streams may be any of the following:

- a pressure difference may exist between the two streams of fluid
- in many, if not most, cases the one stream would contamine the other, if they were permitted to mix
- heat exchangers permit the use of an intermediate fluid better suited than either of the principal media for transporting waste heat through long distances
- 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 ceramic industries. The heat exchanger performing the same function in the stream 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 products of combustion while the other is being cooled by the incoming air. Let us show some simple types of heat exchangers.

Recuperators

The simplest configuration for a heat exchanger is the metalic radiation recuperator which consists of two concentric lengths of metal tubing as shown in figure 3. The inner tube is often fabricated from high temperature materials such as stainless steels.

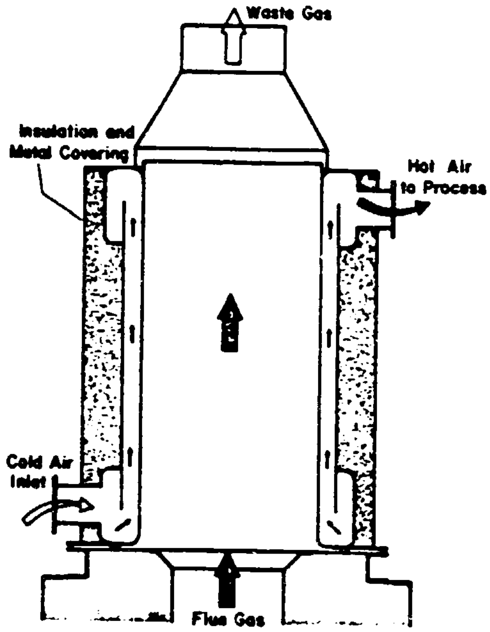

Figure 3. Diagram of metallic radiation recuperator.

A second common configuration for recuperators is called the tube type or convective recuperator. As seen in the schematic diagram of figure 4 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.

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 1540°C and on the preheated air side to 1200°C on an experimental basis and to 800°C on a more or less practical basis. Earlier designs had experienced leakage rates from 8 to 60 percent. The new designs are reported with air preheat temperatures as high as 700°C with much lower leakage rates.

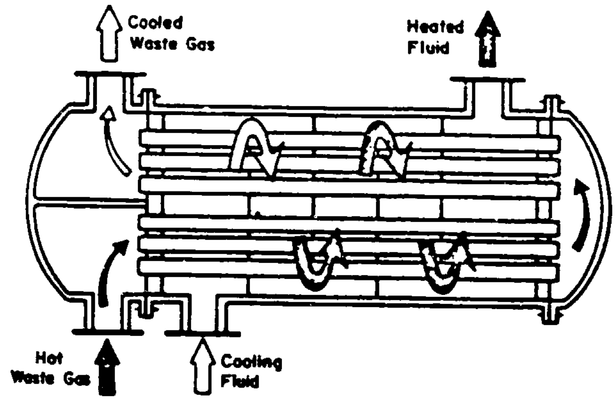

Figure 4. Diagram of convective type recuperator.

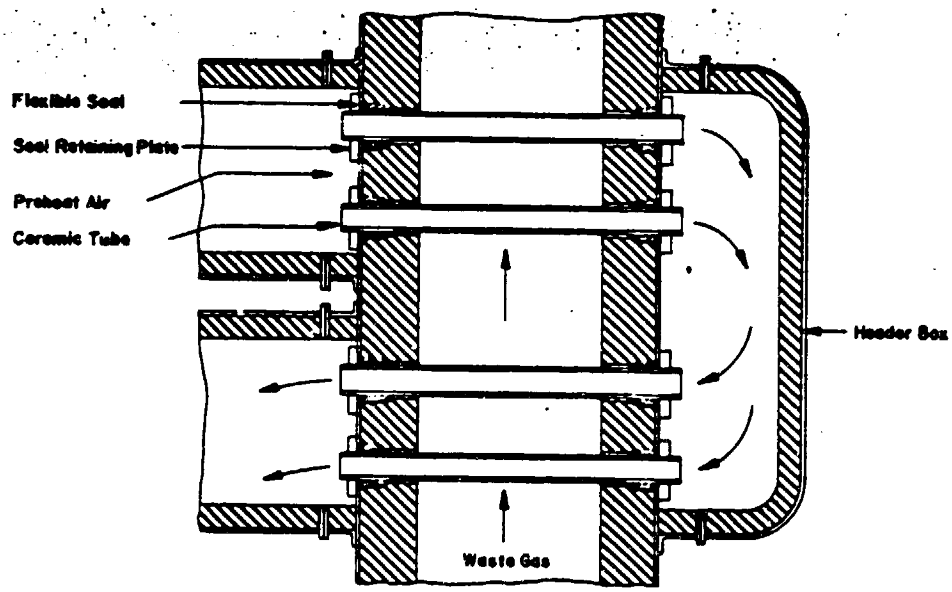

Figure 5. Ceramic recuperator.

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 6. The advantage claimed for this arrangement is the ease of replacing 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.

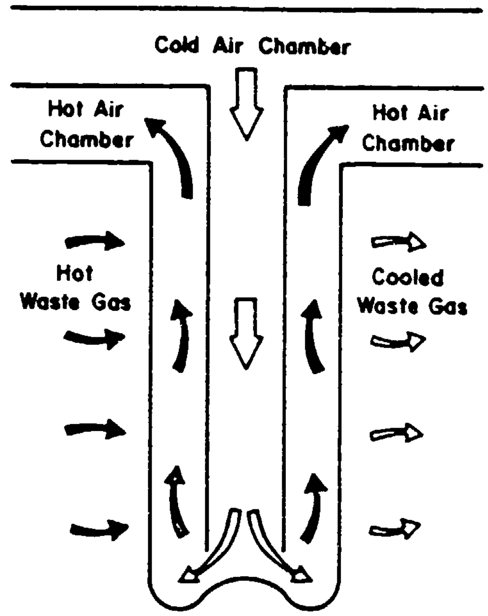

Figure 6. Diagram of vertical tube-within-tube recuperator.

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 7.

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

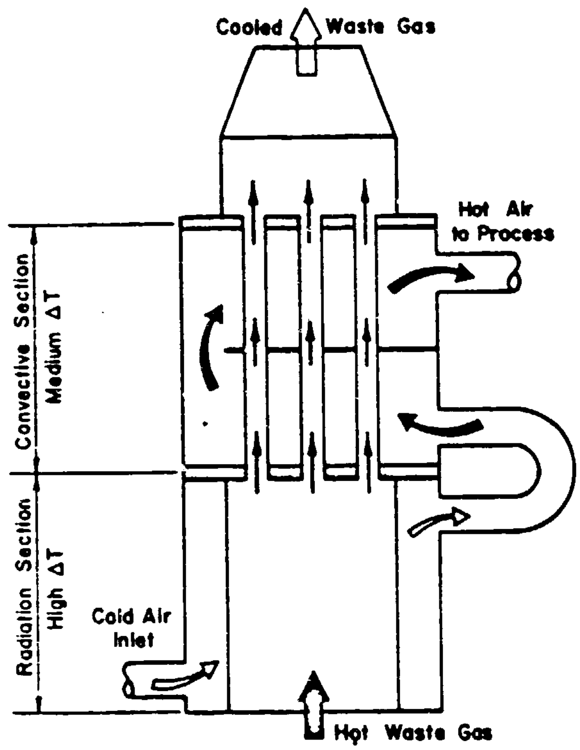

Figure 7. Diagram of combined radiation and convective type recuperator

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 8 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.

Heat wheals.

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 9 is a sketch

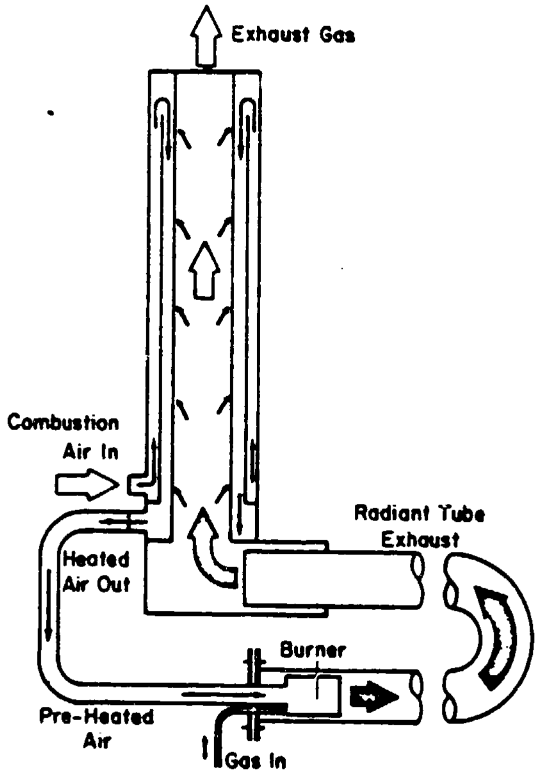

Figure 8. Diagram of a small radiation-type recuperator fitted to a radiant tube burner

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 20 m in diameter with air capacities up to 20 m^3/s. Multiple units can be used in parallel.

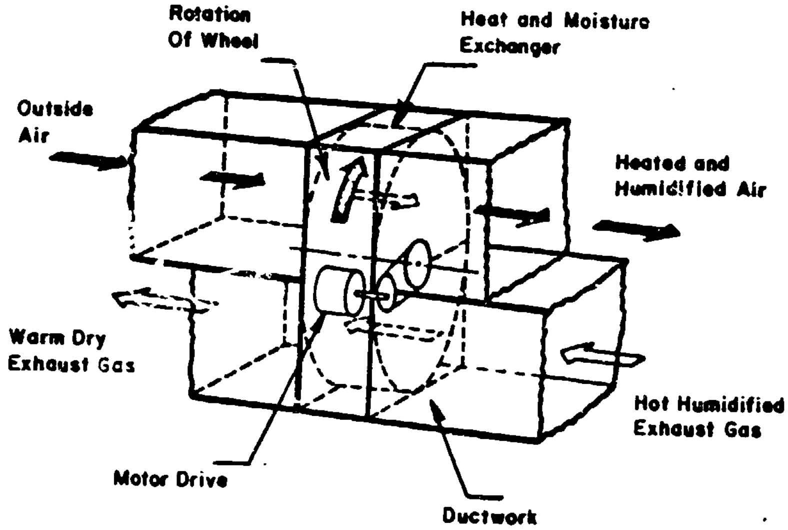

Figure 9. Heat and moisture recovery using a heat wheel type regenerator

Most industrial stack gases contain water vapor, since water vapor is a product of the combustion, of all hydrocarbon fuels,drying, etc. 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. Its latent heat may be a substantial fraction of the sensible energy in the exit gas stream. A hydroscopic material is one such as lithium chloride /LiCl/ which readily absorbs water vapor and it can be concluded that the ratio of water to lithium chloride in $LiCl/H_2O$ is 3/7 by weight. In a hydroscopic 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 10 illustrates the features of an installation using a purge section.

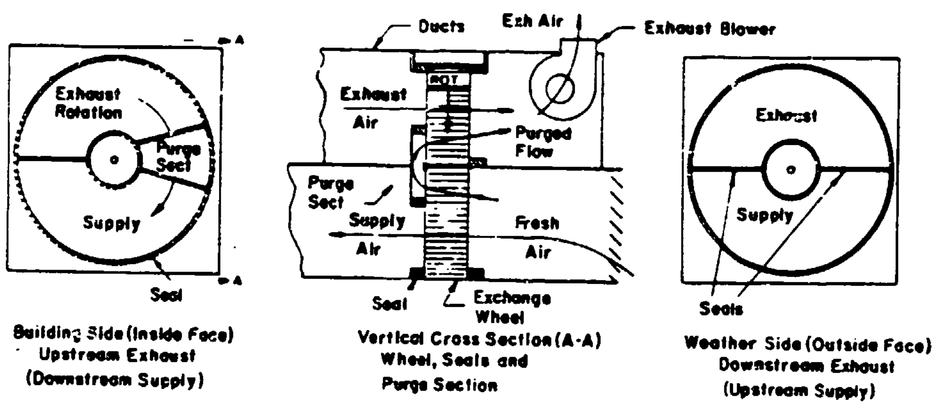

Figure 10. Heat wheel equipped with purge section to clear contaminants from the heat transfer surface

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.

One application of heat wheels is in space heating situations where unusually large quantities of ventilation air are required for health or safety reasons. Another typical applications would be curing or drying ovens and air preheaters in all sizes for industrial and utility boilers in low and moderate temperature environments.

Other type heat exchangers

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.11/ 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.

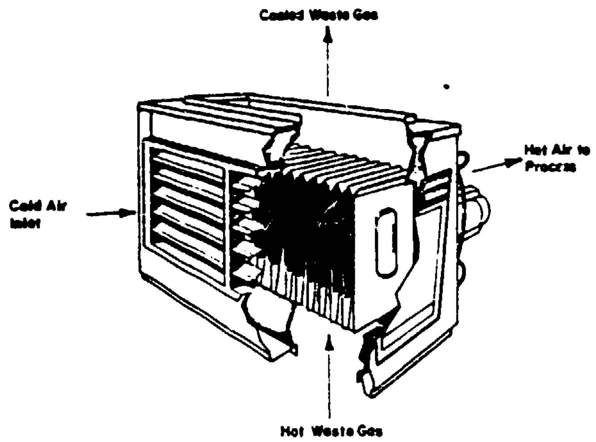

Figure 11. A passive gas to gas regenerator

A list of typical applications follows:

- 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

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 12, 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.

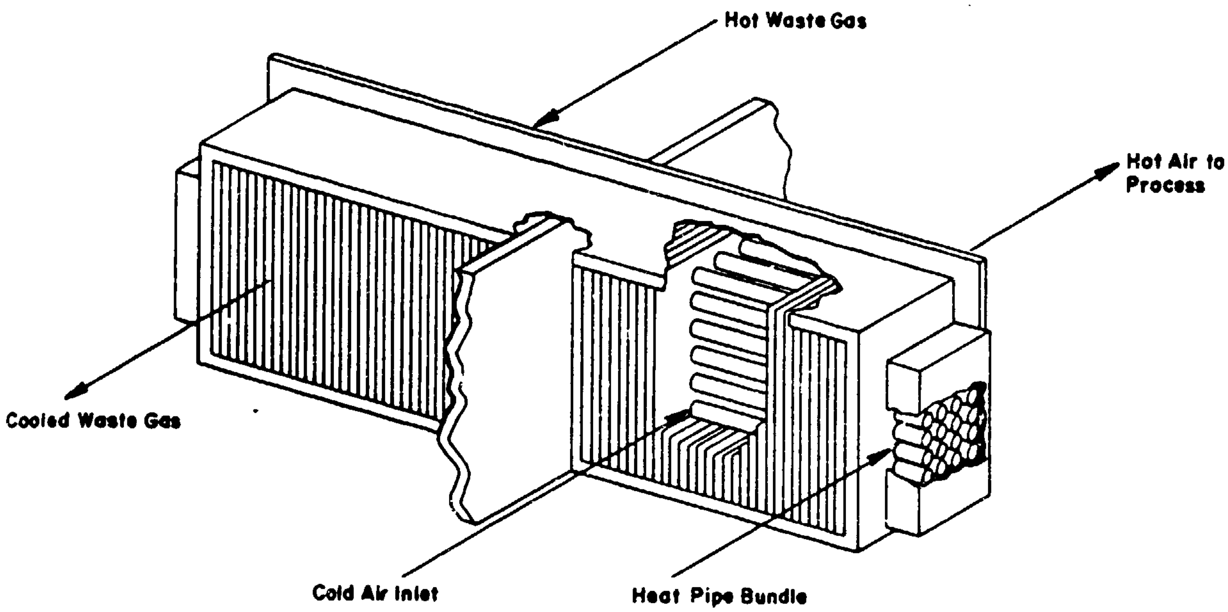

Figure 12. Heat pipe bundle incorporated in gas to gas regenerator

Figure 12 shows how the heat absorbed from hot exhaust gases evaporates the e..trained 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 ·ondenses 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:

- the finned-tube bundle is inherently a good configuration for convective heat transfer in both gas ducts, and
- 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:

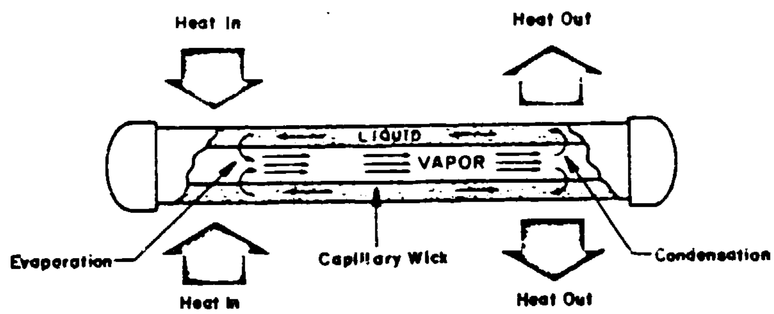

Figure 13. Heat pipe schematic

- 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

When waste heat in exhaust gases is recovered for heating liquids for purpose 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. Figure 14 shows the usual arrangement for the finned-tube-heat exchanger positioned in a duct. The tubes are often connected all in series but can also be arranged in series-parallel bundles to control the liquid side pressure drop. Typical applications are domestic hot water heating, heating boiler feedwater, hot water space heating, absorption-type refrigeration or air conditioning, and heating process liquids.

Waste heat boilers are ordinarily water tube boilers in which the hot exhaust gases from flame-fired furnaces, 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 15 indicated one arrangement that is used, where the exhaust gases pass over the water tubes twice before they are exhausted to the air. If the exhaust

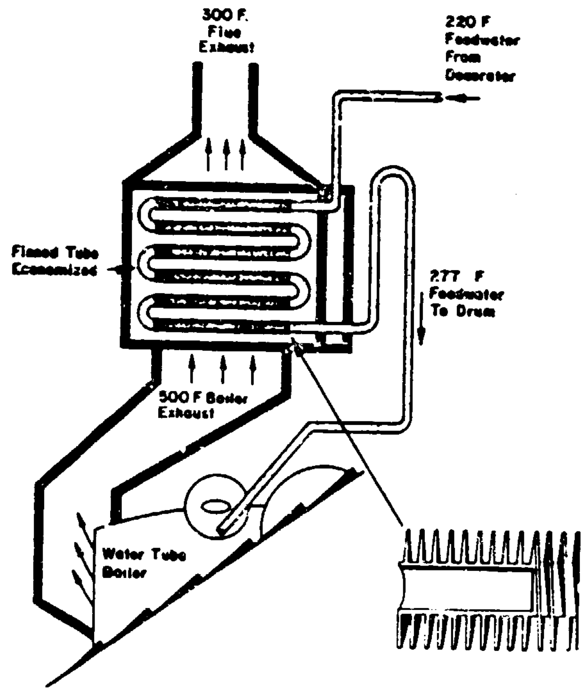

Figure 14. Finned tube gas to liquid regenerator /economizer/

gases are in the medium temperature range, in order to conserve space a more compact boiler can be produced by use of finned tubes which increases the effective heat transfer area on the gas side.

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. Perhaps of greater applicability than the last example are steam used for producing mechanical work or for driving electrical generators. In silicate industries in countries having tropical or subtropical climatic conditions, the mechanical work is of greater importance /for example, to drive air conditioning devices/ than the heat recovered from technologies /perhaps except for conversion sea/drinking water/.

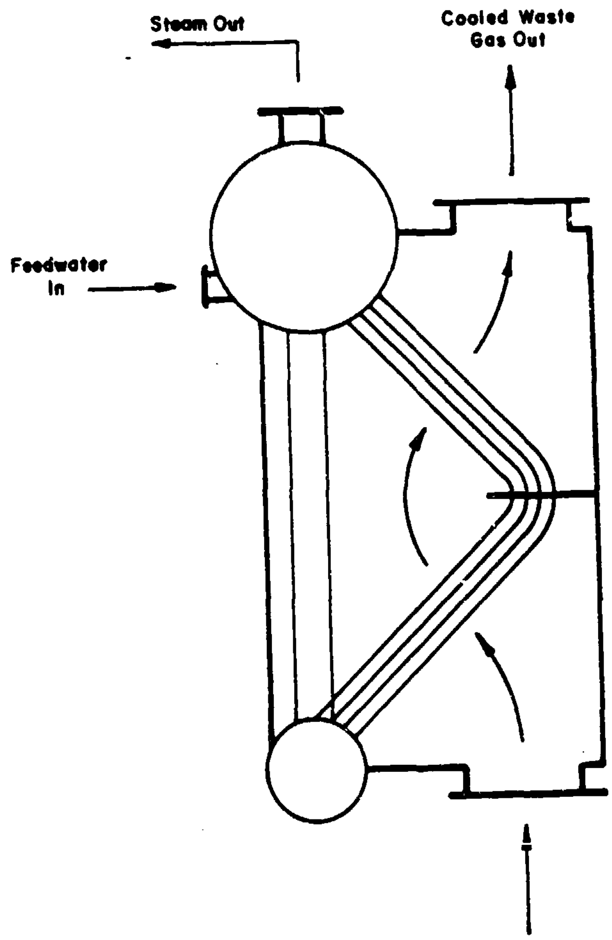

Figure 15. Waste heat boiler

In many cases waste heat to be transferred from a fluid to another one, whose temperature is higher. It is possible to reverse the direction of spontaneous energy flow by the thermodynamic system known as a heat pump. This device consists of two heat exchangers, a compressor and an expansion device. 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 an 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 evaporator where the cycle repeats. The coefficient of performance of an ideal heat pump would be found as

$$\eta_{Therm} = \frac{T_H}{T_H - T_L} = \frac{\text{Ideal transferred heat in condenser}}{\text{Ideal compressor work}}$$

where T_L/K/ is the temperature at which waste heat is extracted from the low temperature medium and T_H/K/ is the high temperature at which heat is given up by the pump as useful energy.

The heat pump was developed as a space heating system where low temperature energy /for instance from the ambient air, sea water, earth/ is raised to the temperature of heating system. In the past, the heat pump has not been applied generally to industrial applications.

The next table presents the collation of a number of significant attributes of the most common types of industrial heat exchangers in order to allow rapid comparisons to be made in selecting competing types of heat exchangers/see page 42/.

Over-All Heat Transfer Coefficient

This coefficient from equation /30/ which is valid for a heat exchanger may be simply expressed in the case when two fluids are separated by a solid simple wall by equation

$$K = \frac{1}{\frac{1}{F_1 h_1} + \frac{1}{F_2 h_2} + \frac{\delta}{k F_\delta}} \qquad /31\ a/$$

or when the wall is finned, by equation

$$K = \frac{1}{\frac{1}{F_1 h_1 E_1} + \frac{1}{F_2 h_2 E_2} + \frac{\delta}{k F_\delta}} \qquad /31\ b/$$

where F_1, F_2 - areas of finned heat transfer surfaces /see figure 16/

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.

h_1, h_2 - heat transfer coefficients
E_1, E_2 - temperature efficiency of finned surfaces
δ - thickness of the separating wall and
k its heat conduction coefficient
F_δ - mean area of the separating wall

The use of equations /31 a/ or /31 b/ can the reader find more detailed in special literature, as well for the reference surface F and the mean temperature difference θ .
As to reference surface F, it is usually taken as the mean area of the separating wall

$$F = \frac{F_1' + F_2'}{2}$$

/see figure 16/.

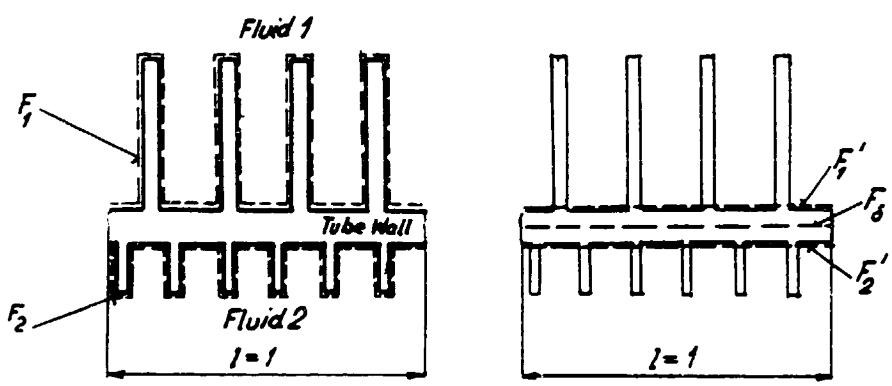

Figure 16. Heat transfer areas of a finned tubes

The mean temperature difference θ is defined either for parallel flow or for counter flow, when it is true /see figure 17/, or for cross - and/or combined flow. In this case the idealized temperature difference for counter-flow /see figure 17 b/ is calculated and using special corrections the mean temperature difference θ is evaluated.

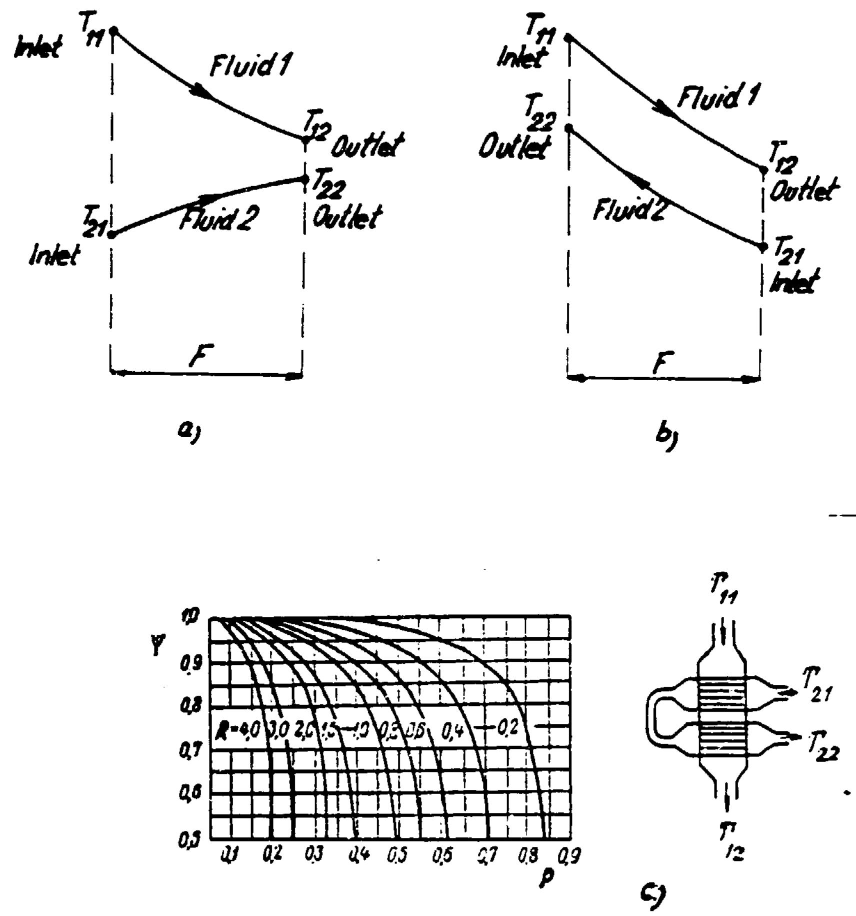

Figure 17. Temperature scheme for parallel- and counter-flow in a heat exchanger

It is as follows:

$$\theta = \frac{\Delta t_2 - \Delta t_1}{\ln \frac{\Delta t_2}{\Delta t_1}} \qquad /32/$$

For parallel-flow : $\Delta t_1 = T_{11}-T_{21}$; $\Delta t_2 = T_{12}-T_{22}$,

and for counter-flow: $\Delta t_1 = T_{11}-T_{22}$; $\Delta t_2 = T_{12}-T_{21}$

For cross-flow and for combined flow the ratios

$$P = \frac{T_{22}-T_{21}}{T_{11}-T_{21}} \quad \text{and} \quad R = \frac{T_{11}-T_{12}}{T_{22}-T_{21}}$$

are to be expressed and a correction factor Ψ /see for example the figure 17 c for the given form of fluids path/ must be found.

Then $\theta_c = \Psi . \theta$ /32 a/

where θ is calculated as the one for counter-flow. This simple directions are valid for fluid-wall-fluid type heat exchangers only. For more sofisticated heat exchangers /for example heat wheel, heat pipe bundles, etc./ the formulas for determining of K- and θ- values have to be modified by special methods. The analogous specialized access must be accepted if the calculation of heat transfer between material and furnace walls using the equation /30/ is desired.

To estimate the simple heat transfer coefficients between fluid and wall and/or the over-all ones in a "fluid-wall-fluid"-types heat exchangers the approximate data listed in the next table may be used:

process, fluids	heat transfer coefficient $Wm^{-2}K^{-1}$		
heating or cooling air	1	-	60
heating or cooling overheated steam	30	-	130
heating or cooling oils	60	-	2000
heating or cooling water	250	-	12000
boiling water convection	600	-	55000
film condensing steam	5000	-	19000
droplet condensing steam	5000	-	150000

heat exchange	over-all h.t.coefficient $Wm^{-2}K^{-1}$		
gas gas	10	-	30
gas water	20	-	50
water water	800	-	1600
condensing steam → water	2000	-	3200
condensing steam → oil	160	-	400
condensing steam → boiling oil	400	-	600

6. Waste Heat Recovery

The energy balance of some silicate industry plants is nearly in equilibrium state, i.e. the only little portion of the balanced energy is lost in surroundings and/or exhausted in air.

This is not fully the case in glass technology, where the developed heat recovery devices can be equipped. Let us show waste heat recovery devices particularly the heat exchanger appliances, in silicate industries in some simple examples.

Cement rotary kiln - two stage dry process

The diagram shown in figure 18 presents gas and law material temperatures of the individual preheater stages which are located before a rotary kiln. In the preheater, which is a noncoventional kind of heat exchanger, the raw material is preheated by waste gases to the degree of calcination. The hot clinker falls off into a clinker cooler, where the secondary combustion air is preheated. As primary combustion air the free air is used. The stack gas from preheater is used for drying raw materials in a drier.

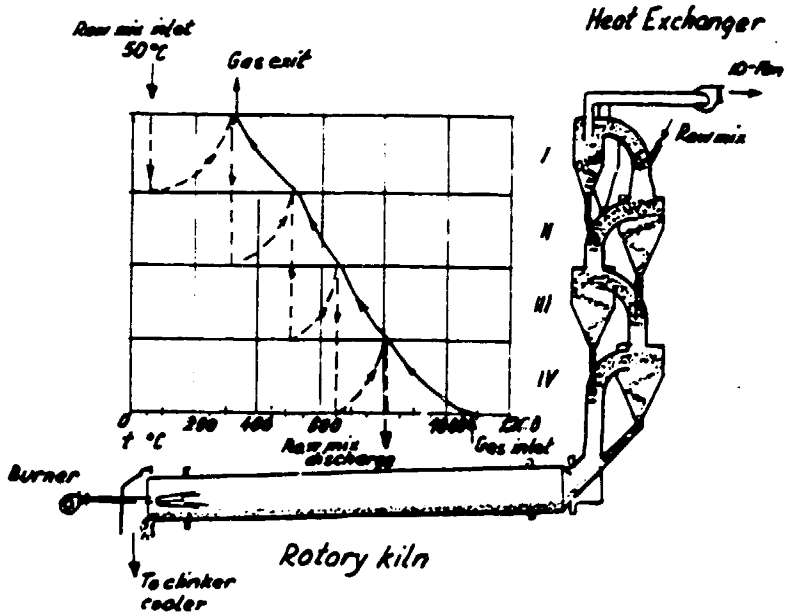

Figure 18. Waste heat recovery in cement rotary kiln by preheater

The example of a furnace having a nearly equilibrium heat balance is the small section tunnel kiln for bisque firing of wall tile, where the wall tiles are placed in fireclay setters to fill cars for passage the kiln. Some opportunities for heat recovery gives the schematic diagram of such kiln showed in figure 19.

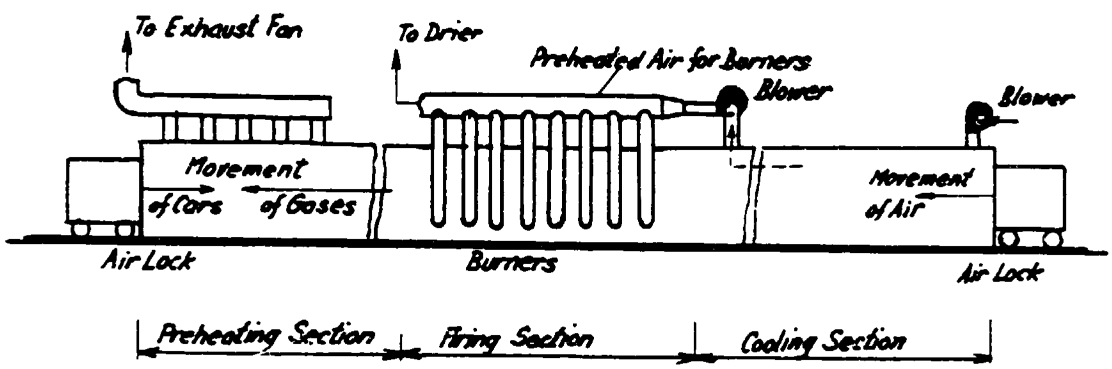

Figure 19. Scheme of an open-fired tunnel kiln

It is obvious, that the amount of heat accumulated in ware is in cooling section transferred to cooling air and exploited to increase the firing temperature by preheated primary and secondary combustion air. On the other side the cars with ware are preheated by flow of flue gases in the preheating section. The degree of exploitation of the heat capacity of flue gases depends on pressure relations in all three sections, i.e. depends on volume rates of air or gases transported by blowers and by exhaust fan. This determines also the acces of preheated air which can be exploited in drier and the amount and temperature of the stack air. This thermal energy can be recovered in an additional heat exchanger for various purposes, dependent on its temperature, which is about 250-400°C. To improve the waste heat utilization, the improvement of the kiln insulation is necessary and the decision has to be made, if the waste heat would be utilized as heat or be converted to mechanical or electrical energy. If the heat is desired, the outlet temperature of exhaust gases can be lower and the degree of exploitation of flue gases in the preheating section will be higher. If the heat is to be converted into mechanical or electrical energy, the exhaust gas temperature is recommended to be higher. The correct relations ensures the careful made energy balance and the analyse of the economic efficiency of the designed energy-saving device.

The next example indicated the principle of use hydroscopic heat wheel-type heat exchanger to utilize the waste heat of a drier. As shown schematically in figure 20, the

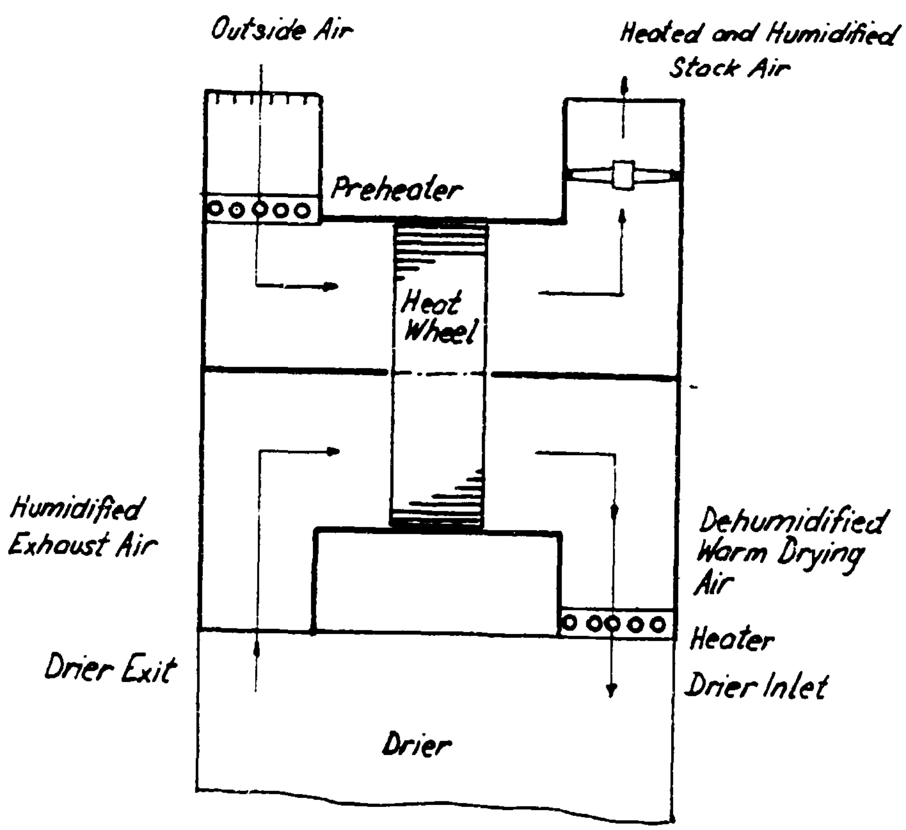

Figure 20. Heat wheel with a drier

humidified exhaust air flowing from drier outlet gives up part of its water vapor to the wheel. Dehumidified and with decreased or increased temperature / it depends on the degree of preheating outside air/ returns back to drier inlet. The water absorbed in wheel is given up to the incoming outside air flowing through the other half of wheel. In this manner the humidified drying air may be circulated without to be exhausted and wholy substituted by heated outside air and the heat wheel is theoretically the most suitable to recover the low temperature level waste heat.

As the last case let us show the recovering of the medium temperature level waste heat of a glass plant. A glass melting tank, equipped with ceramic brick regenerators and with a small packaged boiler for heating steam is retrofitted with an economizer to heat the boiler feedwater /see figure 21/. The total steam produced during the heating season differed from that for the remainder of the year. The combustion burner compensates and smooths the variable heat input on the tank exhaust gases side. The simple installed economizer reduced necessary heat from fuel combustion and secured warm water supply during the season of normal boiler shutdown or of its subdued work.

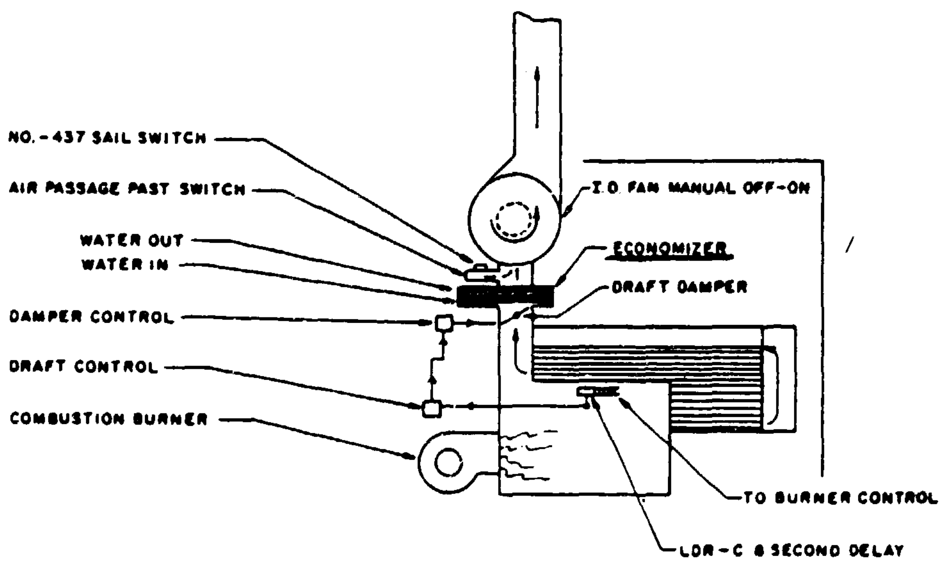

Figure 21. Gas fired boiler with economizer installed

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