ENERGY ANALYSIS

WITH A POCKET CALCULATOR

BY

G.A.PATTERSON

2nd Edition

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"Success of the individual in the achievement of his immediate aims depends on his conscious insight into causal connections." — *New Studies in Philosophy,* F. A. Hayek

"To be is to be related."
— *Science and Sanity,* Count Alfred Korzybski

FOREWORD

There is great current interest, prompted by accelerating costs, to devise improved energy systems. Without attempting to rationalize specific proposals, we should be guided by two viewpoints:

Is it feasible?
Is it economic?

By feasible, we mean, simply, can it be done under practical conditions? For example, solar energy for night usage needs a storage system. Solar energy for transportation needs also a storage system which must be portable. Costs of the storage system, its usable life, and its weight, can render the concept unfeasible.

By economic, we mean, is this the best way, the low cost way? In a completely free maket the low cost system will prevail. The test is often related to inherent efficiency of the process. For example, the reciprocating steam engine is less efficient than the diesel engine and has disappeared from use as a railroad or ship engine.

The economic balance can be shifted to the less efficient process by subsidy of government grants or by use of "waste" products. For example, a lumber mill may use wood chips to power a steam boiler. Wood chips are themselves valuable, however, for making composition lumber and their use for fuel may be uneconomic.

The guide posts of feasibility and economics can only be judged by current technology. There are many promising and intriguing concepts for future use. Among these are fusion power from the atom, electrohydromagnetic processes, and most exotic of all, energy released by combustion of common earth.

To a considerable degree, the tests of feasibility and economics depend upon proper application of the laws of nature to the task. The intent of this book is to identify these laws as they apply to energy and to show how they are related and must be used.

> "Progress in thinking is progress toward simplicity."
> — Siegmund George Warburg,
> Merchant-Banker

INTRODUCTION

What are the *sources* of energy and how do they compare on an economic basis?

What are the *forms* of energy and how are they controlled and used?

The answers to these questions form the basis for this book and give us the guides we need to select the most economical fuel and to use it most efficiently.

The analysis of energy is based upon easily understood effects:

> Energy flows always from hot to cold unless acted upon by external forces.
>
> Energy flows only by
> *Conduction* - through an unmoving substance
> *Convection* - through a fluid (a liquid or gas)
> *Boiling* - a combined conduction/convection process with formation of bubbles
> *Radiation* - through space by electromagnetic waves

This book describes the various natural laws which govern these effects, tells how they are related, and gives examples and step-by-step solutions to each problem. The programmable pocket calculator is used to provide rapid solutions as well as to identify the exact data and methods to be used.

Energy analysis is basic to the design of efficient structures and machines. By learning to control the flow of energy we can add to our comfort, reduce our expenditures, and take satisfaction in having enlarged our understanding of the universe.

BASIC CONCEPTS OF ENERGY ANALYSIS

FOURIER'S PRINCIPLE: ENERGY FLOWS FROM HOT TO COLD. THE RATE OF ENERGY FLOW IS INVERSELY PROPORTIONAL TO THE RESISTANCE OF THE MEDIUM.

Baron Jean Baptiste Joseph Fourier (1768 - 1830), a French mathematician and physicist, established this principle in 1822.

MAXWELL'S EQUATIONS: ENERGY CAN BE PROPAGATED AND ABSORBED IN THE APPARENT ABSENCE OF A MEDIUM BY ELECTROMAGNETIC WAVES CHARACTERIZED BY AMPLITUDE AND FREQUENCY.

James Clerk Maxwell (1831 - 1879), a Scottish physicist, related etheric phenomena (light, radiation energy, heat, magnetism, radio and cosmic waves) by laws which he devised in 1864.

DIMENSIONLESS NUMBERS: THE UNITS AND QUANTITIES OF A MATHEMATICAL OPERATION ARE DETERMINED BY THE UNITS AND QUANTITIES USED IN THE OPERATION. UNITS WHICH COMBINE TO PRODUCE NONDIMENSIONAL NUMBERS CAN BE USED TO GENERALIZE NATURAL PHENOMENA. CONVERSELY, EQUATIONS WHICH ARE NOT NON-DIMENSIONAL ARE NON-GENERAL.

Osborne Reynolds devised the first well known dimensionless number in 1882 to describe fluid flow phenomena. Lord Raleigh and E. Buckingham independently in 1915 devised rules for such analysis. A pioneer work was given in 1922 by P.W. Bridgman of Yale University.

THE COBURN FACTOR: THERE IS AN ANALOGY, A RELATION, IN FLUID FLOW BETWEEN THE FRICTIONAL RESISTANCE AND THE TRANSFER OF ENERGY BETWEEN TWO FLUIDS OF DIFFERENT TEMPERATURE.

This relation was noted by A.P. Colburn and T. H. Chilton in a chemistry work published in 1934.

BOUNDARY LAYER: A BODY IMMERSED IN A FLOWING FLUID IS BOUNDED BY A FILM OR BOUNDARY LAYER. THE VELOCITY AND THICKNESS OF THE FILM IS ZERO AT THE LEADING EDGE "STAGNATION POINT". THE THICKNESS AND VELOCITY INCREASES WITH INCREASING DISTANCE AFT AND OUTWARD OF THE STAGNATION POINT UNTIL IT RESUMES FREE STREAM VELOCITY.

Ludwig Prandtl developed the boundary layer theory in 1909 during experimental investigations of airfoils at Gottingen, Germany.

CONCISE DEFINITIONS

HEAT — Energy in transit from a particular body.

BTU — BRITISH THERMAL UNIT, the energy needed to raise the temperature of a one pound body by one degree Fahrenheit at 60 deg. F.

CONDUCTIVITY — The rate at which energy will flow from a particular body expressed as BTU/sec-ft-deg F.

R VALUE — The inverse of conductivity for a particular thickness and unit area, sec-deg. F/BTU.

LAMINAR FLOW — Fluid flow in streamlines with gradual change of direction.

TURBULENT FLOW — Fluid flow in chaotic patterns with vortices.

FREE CONVECTION — Fluid flow induced by density variations caused by temperature differences.

FORCED CONVECTION — Fluid flow induced by pressure differences.

NUCLEATE BOILING — The periodic formation and escape of vapor bubbles with transfer of energy.

FILM BOILING — The formation of a vapor film between a liquid and a heated surface at very high energy transfer rates.

BURN OUT — An unstable intermediatary condition between nucleate boiling and film boiling with low energy transfer rates. At such a condition the wall may be vaporized or weakened.

CONTENTS

ENERGY ANALYSIS ELEMENTS

The Temperature Scale

The invention of the thermometer is credited to Galileo in 1592 and was improved in 1700 by Gabriel Fahrenheit who used mercury as the fluid and devised the scale given his name. On the Fahrenheit scale fresh water at sea level freezes at 32 degrees and boils at 212 degrees.

The Centigrade (cent, one hundreth), or Celsius, scale was proposed in 1742 by Anders Celsius. On this scale water freezes at 0 degrees and boils at 100 degrees.

In comparing the two scales, it can be seen that for the Fahrenheit scale the range of degrees for freezing to boiling is 180 degrees, while for the Celsius scale it is 100 degrees. This is a ratio of 1.8 degrees C for each degree F. There is also a base change of 32 degrees between the two systems.

$$\text{Deg C} = (\text{deg F}-32)/(1.8\ \text{deg F}/\text{deg C})$$

$$\text{or, for example,}(100\ {}^{o}\text{F}\ -32)/1.8 = 37.8\ {}^{o}\text{C}$$

$$\text{Deg F} = \text{deg C} \times (1.8\ \text{deg F}/\text{deg C}) + 32$$

$$\text{or, for example,}(37.8\ {}^{o}\text{C} \times 1.8) + 32 = 100.\ {}^{o}\text{F}$$

The effect of temperature change on gas volume was noted in 1805 by two French chemists, Gay-Lussac and Charles. They found the volume change was proportional to the temperature plus a constant. In effect, they discovered the absolute zero reference value.

Gas volume at constant temperature is directly proportional to pressure, as found by Robert Boyle in 1700.

Rankine and Kelvin in 1900 did experimental work at very low temperatures and developed scales given their names:

$$\text{Deg R} = \text{deg F} + 460.$$

$$\text{Deg K} = \text{deg C} + 273.$$

Vapor Pressure and Boiling Point

Vapor pressure is the aggregate result of molecular motion measured at the gas interface. It increases with temperature and persists at all temperatures above absolute zero.

Vapor pressure is related to molecular weight; the smaller and lighter molecules have higher vapor pressure. Substances composed of several types of molecules, such as petroleum fuels, have a vapor pressure which can be calculated by special methods.

Vapor pressure determines the boiling point, or the temperature at which a fluid boils.

Water boils at 212 °F at sea level.
That is to say, the vapor pressure of water
at 212 °F is equal to 14.7 psi, sea level pressure

Saturated temperature is the temperature at which the substance boils at a specified pressure. For water at sea level pressure, the saturated temperature is 212 deg F.

Reid Vapor Pressure is a measure of the vapor pressure of the more volatile fuels. The sample is heated to 100 deg F in a container one-fourth full of liquid. The true vapor pressure is approximately 3 percent higher because of the partial evaporation of the fuel.

Surface Finish

Energy transfer is significantly affected by surface effects, such as oxide coatings, oily films, chemical scale and deposits. Surface texture and reflectivity play an important part.

An oil film can greatly increase the temperature at which boiling begins. Light oil is sometimes added to open water reservoirs to reduce the loss by evaporation.

Boiling depends on the existance of gas or vapor bubbles found in crevices at the solid-liquid interface. Small surface scratches are nearly always present and can increase the rate of boiling. Very smooth, deaerated surfaces delay the onset of boiling. It is in fact possible to exert negative pressure of several thousand psi without boiling under laboratory conditions.

These factors contribute to the difficulty of precise definition of heat transfer phenomena. Conditions are idealized and simplified for laboratory experiment in order to improve the repeatibility of results. For example, gases may be removed from the liquids, surfaces may be chemically cleaned and ground smooth and polished. Such results may not be the same as for field tests.

The States of Matter

Matter is composed of molecules which can be arranged rigidly in the case of the solid, semirigidly for the liquid, and freely for the gas. Solids have a specific crystalline arrangement peculiar to the type of substance. There are a certain number of molecules to each crystal. Various crystalline arrangements are possible. for the same substance. At higher pressures the crystals condense to smaller volume. At the other extreme, gases can become very diffuse at low pressures, becoming ionized to enter a plasma state with new electrical values.

The triple point identifies the melting/freezing point. At this temperature the substance can exist as a liquid, solid, or gas. Below the triple point the substance can sublime directly from a solid to a gas. Above the triple point, the substance can change from a liquid to a gas or the opposite, and a mixture of liquid and gas can be present. A mixture of gas and liquid is defined by quality, where zero quality is pure liquid and 100 percent is pure gas. The change of state is determined by the environment pressure and the addition or subtraction of energy.

The critical point identifies the highest temperature at which the liquid state can exist. There is a corresponding minimum pressure known as the critical pressure. The two are known as the critical points.

Change of state can be gradual, as in the warming of water, or precipitious, as in the formation of ice or steam. Heat transfer experiments have determined the quantity of energy needed to change the temperature of a pound of a substance, or to melt, freeze or vaporize it. Data for various substances are given in Appendix II.

Phase Change

Melting and freezing are reverse processes which occur at the freezing point with the addition or subtraction of the heat of fusion.

Freezing of a liquid proceeds by excluding the impurities, that is, the pure substance freezes first, excluding in the process the dissolved materials. This effect is of great use for purification processes, such as the preparation of fresh water from sea water.

Similarly, melting of pure and impure substances proceeds at different rates. The melting of an ice cube is an example. During freezing, the outer layer of water is cooled first and freezes the purest water. Inner layers of water freeze last and contain the most dissolved substances. This can be observed by the clarity of outer ice and the opaquity of inner ice. Old ice tends to be entirely opaque because of the condensation and absorption of gases from the environment.

The preferential freezing of the pure substance is used in the metallurgical process of zone melting. A bar of the substance is heated along its length by induction heating in a progressive fashion. Impurities are driven to the far end of the bar. The impurities are then removed by cutting off the end of the bar.

Evaporation and condensation are reverse processes of the change from the gas to the liquid state. The change proceeds at constant temperature with energy added for evaporation and subtracted for condensation.

Evaporation proceeds from the liquid-gas interface. It is encouraged by increasing the area of the liquid-gas interface, either by greater surface area or by agitation of the liquid. Evaporation of the liquid proceeds until the partial pressure of the evaporated gas equals the partial pressure of the liquid (its vapor pressure) or until the liquid is exhausted.

The Sources of Energy

Energy is produced by changes in the composition or arrangement of matter.

Solar energy is the result of thermonuclear processes. The very great mass of the sun produces high pressures at its center, crushing together the atoms of which all matter is composed. This rearrangement causes high temperatures and the energy evolved is transmitted to us by electromagnetic waves, or radiation, a portion of which we percieve as light.

Thermal energy can be generated by friction, compression, radiation and chemical processes. The latter include room temperature oxidation in which the heat produced may be unnoticed. If the oxidation occurs sufficiently rapidly and in a concentrated form, then combustion results. See Engine Thermodynamics, Ref. 14.

Energy analysis is concerned primarily with the conservation of energy or with its dissipation. Conservation would apply to the insulation of a machine or structure to preserve the required temperature with the least expenditure of fuel. Dissipation of energy would apply to maintaining a desired temperature under a condition in which waste heat was produced, as for example, use of a radiator to avoid excessive cylinder and bearing temperatures. Accordingly, energy analysis is concerned primarily with insulators and radiators.

Heating Value

The unit of energy is the British Thermal Unit (BTU). It is defined as the energy needed to change the temperature of one pound of water by one degree Fahrenheit.

An alternate unit of energy is the Gram-Calorie. It is the energy needed to change the temperature of one gram of water by one degree Celsius. Also in use is the Kilogram-Calorie. If no units are given, the Calorie refers to the gram-calorie.

There are 454 grams per pound and 1.8 degrees F per degree C. The BTU is equal to 252 Gram-Calories.

There is an equivalence beteeen thermal processes and mechanical processes, such as friction and gas compression as compared to combustion processes. This was investigated by Benjamin Thompson (Count Rumford) during the American Revolution and later by James Joule in 1845. The equivalence, called the Joule, is 778.16 foot-pounds/BTU.

Heating value is found by burning the fuel in a container submerged in water and insulated, with a measurement made of the temperature rise. Water is formed in this process by the products of combustion. The heating value is known as the "higher heating value" if the vapor is not condensed, and as the "lower heating value" if the water is condensed.

Heating value of a fuel can be estimated if the hydrogen/carbon ratio of the substance is known, since these are usually the principal constituents of common fuels. Approximate values are 53,000 BTU/lb for hydrogen, and 14,000 BTU/lb for carbon.

Internal Energy

Energy analysis is concerned with the change of internal energy of a substance for a phase change from gas to liquid or liquid to solid. These changes occur at constant temperature. The change in internal energy is the amount needed to rearrange the molecules into the new form of greater or lesser mobility. It is a precipitious change as compared to the gradual process of warming or cooling a solid, liquid or gas. There is a quantum of energy, measured by units of BTU/lb, to produce the phase change for a particular substance. Phase change is described by various common names:

Less Mobility	Phase Change	More Mobility
	Solid-Vapor	Heat of Sublimation
Heat of Fusion	Solid-Liquid	Heat of Melting
Heat of Condensation	Liquid-Gas	Latent Heat, Heat of Evaporation

The temperature at which phase change occurs is controlled by the pressure. Above the critical temperature of the substance no phase change occurs and the substance is a gas.

At a temperature not equal to the phase change temperature the energy which is added or subtracted from the substance causes a change in the temperature of the substance. The extent of the change in temperature is determined by the amount of energy and the specific heat of the substance, see below.

Specific Heat

A change in temperature can occur at constant pressure or at constant volume. The energy needed to produce this change in temperature is known as specific heat, measured in BTU/lb-°F. It has the notation c_p for constant pressure and c_v for constant volume. The ratio of these specific heats is given the notation k, equal to c_p/c_v.

Specific heat varies slightly with temperature and is nearly constant with pressure. An exception is near the critical point at which the specific heat varies widly.

Thermal Conductivity

Thermal conductivity is a measure of the ability of a body to conduct heat within itself from a higher temperature source to a lower temperature sink. The body may be of a uniform substance or a composite of several materials. It may be a uniform mixtures or made of layers of fifferent substances. It may be solid or there may be void spaces.

The thermal conductivity factor, k, is determined by test and is expressed in units of

$$\text{BTU-ft/sec-ft}^2\text{-deg F} = \text{BTU/sec-ft-deg F}$$

Atmospheric Properties and Ram Effects

Atmospheric pressure is greatest at sea level and decreases exponentially with altitude. Atmospheric temperature similarly decreases with altitude but remains constant for a considerable height above 35,000 feet up to 80,000 feet. For a particular altitude there is wide variation in specific temperature caused by local weather conditions. Standards have been established for general usage. The Standard Day has a sea level temperature of 59 deg F, the Cold Day -65 deg F, and the Hot Day 100 deg F. The actual atmosphere can contain "temperature inversion layers" in which the temperature increases, rather than decreases, with altitude.

A gas becomes heated when compressed and cooled when it is expanded. An object passing through the air compresses the air ahead of it causing an increase in temperature and pressure. These are known as "ram effects".

ENERGY ANALYSIS METHODOLOGY

The Modes of Energy Transfer

It is the first law of scientific reasoning that nature can not act at a distance; there must be a continuity of the process throughout its action.

This continuity is most apparent in the case of energy transfer by conduction. As Fourier showed in 1822, there is an exponential dimunition of temperature along a solid conductor from the source of energy to the energy sink.

Energy transfer by convection deals with a fluid (gas or liquid) medium rather than a solid one. The energy imparted to the fluid reduces its density causing it to move within itself. This creates currents which modify the energy transfer process. If these currents are natural (produced only by gravity) the process is called natural convection. If the currents are caused by external forces (such as a pump) the process is called forced convection.

Energy transfer by boiling is commonly used without appreciation of the complexity of the process involved. The intensity of boiling is controlled by the temperature difference of the boiling fluid and the fluid container at the location where the heat is applied. At low temperature differences, there is convective heat transfer. At somewhat higher temperature differences, the onset of nucleate boiling, with small bubbles, occurs. At increasing temperature difference bubbles become larger. At very high temperature differences, there comes an abrupt transition to film boiling. Boiling bubbles are no longer formed but the entire heating surface becomes an undulating wave of gas film.

Energy transfer by radiation does, to our incomplete perception and understanding, appear to be action at a distance. It does, however, follow the immutable laws of nature discovered by James Clerk Maxwell in 1864. (His work led to the discovery of radio waves.) Radiation transfers energy through a vacuum by what is, in fact, the interchange of energy and matter. The study of radiation has shown us the cosmic scope of energy relations: the exponentially increasing energy of specific vibrations and wave lengths which we recognize as heat, light, radio waves and cosmic waves.

Heat Exchangers

Heat exchangers facilitate the flow of energy. In specific application they are known as coolers or heaters. By location, we may hear of an intercooler, as between two compressors.

The heat exchanger allows the transfer of energy between a hot fluid and a cold fluid. Either or both fluids can be liquid or gas. Important design parameters are cost, size, weight, flow rates, fluid pressure drop and temperature differences.

Insulation

Insulation controls the flow of energy. It increases the time to reach an equilibrium temperature. Insulation achieves its purpose by use of materials and methods of construction which reduce the rate of energy flow. Insulation used between two bodies at nearly room temperature is concerned principally with conduction and convection.

Conduction insulation is obtained by use of materials with low conductivity, as for example, wood in place of steel, and by reducing the area of conduction. A hollow wood door has less area of contact between the two sides and provides better insulation. Conduction can be reduced if a solid material is composed of layers which are slightly separated. Separation can consist merely of oxidation or powder.

Conduction insulation for high temperatures includes graphite and bricks made of fire clay or alumina. Lower temperatures use asbestos.

Space reentry vehicles are subject to very high heat loads at the leading edge by compression of the air. Evaporating or ablative materials may be used for these applications.

Convection insulation is obtained by reducing the mobility of the environmental fluid, usually air. Principal convective insulators are glass wool, fiberboard and foamed polyurethane. It is essential that void spaces be present; if compressed too tightly the materials become more conductive.

At high vacuum conditions the principal mode of energy transfer is by radiation. Sir James Dewar, a Scottish chemist, invented the storage bottle used to store low temperature liquidified gases such as helium and hydrogen. The dewar, as it is called, consists of two glass or metal bottles between which is drawn a high vacuum. (The familiar Thermos bottle.) Multiple radiation surfaces are sometimes used to further reduce energy transfer.

Low temperature insulation is known as cryogenic insulation from the Greek kyros, icy cold. Foamed materials are very useful for such applications, especially if the temperatures are such that the gasses in the bubbles become condensed to form a vacuum. Polyurethane foam is formed by mixing liquids to produce a gas and foaming bubbles. The mixture dries to leave foam skeletons encapsulating the gas. The size and density of the foam can be controlled by regulating the pressure during the formation.

Cryogenic insulation is subject to "cryopumping" in which the surrounding atmosphere and its contaminants, especially water, can be condensed. This can build up deposits between the insulation and the insulated surface causing the insulation to spall off and the conductivity to increase. Moisture seals are required to prevent this occurance.

The Energy Balance

The energy of a body or system is evidenced by

1. Its temperature
2. Its molecular arrangement (liquid, solid or gas)
3. Its chemical arrangement (oxidized or ionized)
4. Its physical state (compressed or distended)

Two bodies brought into contact tend to reach a level of equal energy as limited by time and the degree of contact. Basic energy transfer study is concerned primarily with steady-state relations. That is to say, ones for which the rate of energy transfer is constant. The true processes can be approximated by a series of steady state solutions at progressively converging temperatures and for suitable time intervals.

Energy cannot be created of destroyed (omitting nuclear processes) but can become unavailable, as when useful energy is lost as useless friction. Heat can be transmitted by radiation which includes light and radio waves.

Energy added to a body without changing its structure or temperature must be transferred to a third body. This is the principle of entropy, the conversion of energy to work, as an example. The empty kettle can no longer transfer energy to the water and must add energy to the kettle itself. The empty kettle burns up. The energy balance can be expressed by the equation:

Heat loss = Heat Gain

Analysis Processes

Energy analysis processes are a synthesis of experiment and reasoning. The investigator observes the process of boiling:

> "Aha! Boiling deals with the formation of bubbles!"
>
> He sets up a formula relating the geometry and the materials to the environment. He measures the temperature difference and the heat flux to obtain a set of data points. He now compares the results of his test with the prediction of his equation to test its validity, range of acceptability, and its suitability for other fluids and materials. Usually, he will add an experimental constant to account for unknown effects.

All energy analysis calculations are necessarily incomplete and approximate. There are many methods which can be employed to obtain a satisfactory answer to a particular problem.

Energy analysis is facilitated by ignoring effects known to be minor from preliminary calculations, experience or intuition. For example, radiation between two bodies at room temperature can be ignored in most practical cases.

Energy analysis, as well as most analysis, is best approached by determining the proper order of magnitude of the answer. For instance, is the true rate of energy transfer close to 1 BTU/second, 10 BTU/second, or 1000 BTU/second? Once the problem has been approximated to this extent it can either be refined to the necessary accuracy or disregarded as trivial. The uncertainty of the data should be considered in regard to the validity of the answer.

The key to successful energy analysis is the simplification of the problem and the isolation of the effect to be studied from the rest of the universe. In the Roman maxim: Divide and Conquer. Narrow the scope of the investigation to obtain a first order approximation. Avoid complex solutions. The simplest answer is the most to be trusted.

Dimensionless Numbers

Energy analysis makes extensive use of dimensionless numbers. These are simply combinations of measurements which form a ratio. For example, we might speak of the fineness of a needle, F, measured by the length, L, and the width, W. Then, F= L/W. Measuring both length and width in the same units, as inches or feet, we see that the units cancel so that the fineness ratio, F, is a pure number. That is, it has no units.

One of the earliest and most useful dimensionless numbers was devised by Osborne Reynolds in 1882 for the study of fluids. The Reynolds Number relates length, velocity, density and viscosity. He found that his criterion would identify whether the flow would be laminar (streamlines) or turbulent. The transition for flow in a tube, based on the tube diameter, occurs at a RN of 3,000. The transition for flow past an aircraft wing, based on the chord of the wing, occurs at a RN of 500,000.

Buckingham and Raleigh in 1915 independently derived rules for formulating dimensionless numbers. These laws form the basis for theoretical analysis and provide insight into natural processes. For example, consider the relation between mass and energy.

Mass, M = Weight/Gravity = W/g = $\mathrm{lb}/(\mathrm{ft}/\mathrm{sec}^2)$
$= \mathrm{lb\text{-}sec}^2/\mathrm{ft}$
Energy, E = ft-lb

To relate these, we can divide Energy by Mass:
Number = $(\mathrm{ft\text{-}lb})/(\mathrm{lb\text{-}sec}^2/\mathrm{ft}) = (\mathrm{ft}/\mathrm{sec})^2$

We recognize this as the Einstein Equation, $E = Mc^2$ where c represents the speed of light.

Equations in which the product of the input units do not match the output units are dimensionsally invalid. Equations should be dimensionally analyzed before use.

The Prandtl Number was devised by Ludwig Prandtl while investigating aerodynamics in Gottingen, Germany in 1905. He also developed the concept of the boundary layer surrounded by the ideal gas.

The Nusselt Number was devised by W. Nusselt, Germany, 1909. It is the basis for correlations of convective energy flow.

The Grashof Number is used in convective energy flow where needed for gravity and bouyancy effects. The number can be expressed in terms of the mass density of the liquid and vapor.

REYNOLDS NUMBER

$Rn = DV\rho/\mu$

where D = Diameter of tube, feet
V = Velocity of fluid, ft/se

ρ = Mass Density, $(lb/ft^3)/(32.2\ ft/sec^2)$ or slugs/cu.ft.

μ = Viscosity, $lb\text{-}sec/ft^2$

PRANDTL NUMBER

$Pr = g\ c_p\,\mu/k$

where g = Gravity constant, 32.2 ft/sec^2

c_p = Specific heat at constant pressure, BTU/lb-deg F

μ = Viscosity, $lb\text{-}sec/ft^2$

k = Thermal Conductivity, BTU/sec-ft-deg F

NUSSELT NUMBER

$Nu = h\ L/k$

where k is thermal conductivity as above and

L = Length, ft

h = Heat transfer coefficient, BTU/sec-ft^2-deg F

GRASHOF NUMBER

$Gr = g\,\beta\,(dT)\ L^3\ \rho^2/\mu^2$

where β = Coefficient of volume expansion, ft^3/ft^3-deg F

dT = Temperature difference, deg F

ENERGY ANALYSIS PROCESSES AND EXAMPLES

CONDUCTION

Conduction is the direct transfer of energy from one body to another through a conducting medium. The rate of conduction was experimentally observed by Joseph Fourier in 1822. His equation both defines the experimental value of conductivity for the material and predicts the rate of energy conduction for specific conditions.

Conduction is limited to solid bodies and to liquids at low heat flux rates. This is because heat addition to a gas or a liquid causes convective currents.

Conduction frequently occurs across a wall bounded by a liquid or gas subject to convection. The heat transfer rates will depend upon the film coefficients at the wall surfaces as well as the conduction.

CONDUCTION FORMULAE

HEAT FLUX

$$Q = U\ A\ dT$$

where Q = Heat flux, BTU/sec
U = Overall transfer coefficient, BTU/sec-ft^2-deg F
A = Heat transfer area, ft^2
dT = Temperature difference, deg F

HEAT TRANSFER COEFFICIENT

$$U = k/dL$$

where k = Thermal conductivity, BTU/sec-ft-deg F
dL = Length of heat transfer through body, ft

FOURIER EQUATION

$$Q = -k\ A\ (dT/dL)$$

HEAT TRANSFER COEFFICIENT IN COMPOSITE MATERIALS

$$1/U = dL_1/k_1 + dL_2/k_2 + dL_3/k_3 + \ldots..$$

ADJUSTMENT FOR UNEQUAL AREAS OF COMPOSITES

$$U_x = (\text{fraction of area occupied}) \times (k_x/dL_x)$$

PROCEDURE FOR CONDUCTION

See
APPENDIX I. Program 1. Wall Component Conduction, Single Element
Program 2. Wall Average Conduction, Multiple Element

1. Find heat transfer coefficient for each component, U_x

$$U_x = (A_x/A) \times (k_x/dL_x)$$

2. Find composite heat transfer coefficient, U

$$1/U = 1/U_1 + 1/U_2 + 1/U_3 + \dots$$

$$U = \text{inverse of } (1/U)$$

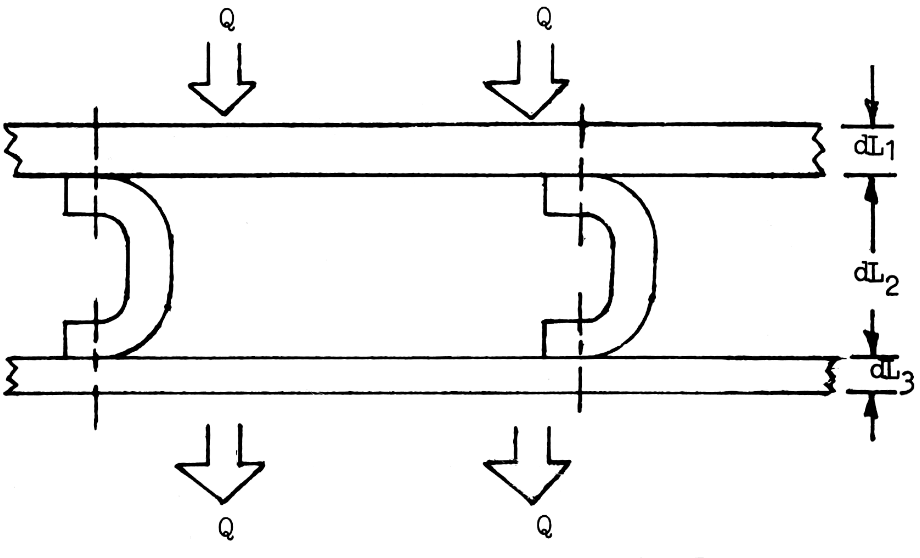
3. Find heat transfer rate, Q

$$Q = U\ A\ dT$$

Example 1. CONDUCTION, SANDWICH STRUCTURE

DATA:

Material	Conductivity, k BTU/sec-ft-deg F	Length, dL ft	Area, A ft^2
Asbestos	2.0×10^{-5}	0.025	1.0
Steel	7.2×10^{-3}	0.13	0.04
Aluminum	3.24×10^{-2}	0.02	1.0



Example 1. ILLUSTRATION OF SANDWICH STRUCTURE

PROCEDURE: See Program 1. COMPONENT CONDUCTION, SINGLE ELEMENT
Program 2. COMPONENT CONDUCTION, MULTIPLE ELEMENT

$$U_1 = (1/1) \text{ x } (2.0 \text{ x } 10^{-5})/(0.025)$$

$$= 8.0 \text{ x } 10^{-4} \text{ BTU/sec-ft}^2\text{-deg F}$$

$$U_2 = (0.04/1) \text{ x } (7.2 \text{ x } 10^{-3})/(0.13)$$

$$= 2.2 \text{ x } 10^{-3} \text{ BTU/sec-ft}^2\text{-deg F}$$

$$U_3 = (1/1) \text{ x } (3.24 \text{ x } 10^{-2})/(0.02)$$

$$= 1.62 \text{ BTU/sec-ft}^2\text{-deg F}$$

$$1/U = 1/(8.0 \text{ x } 10^{-4}) + 1/(2.2 \text{ x } 10^{-3} + 1/1.62$$

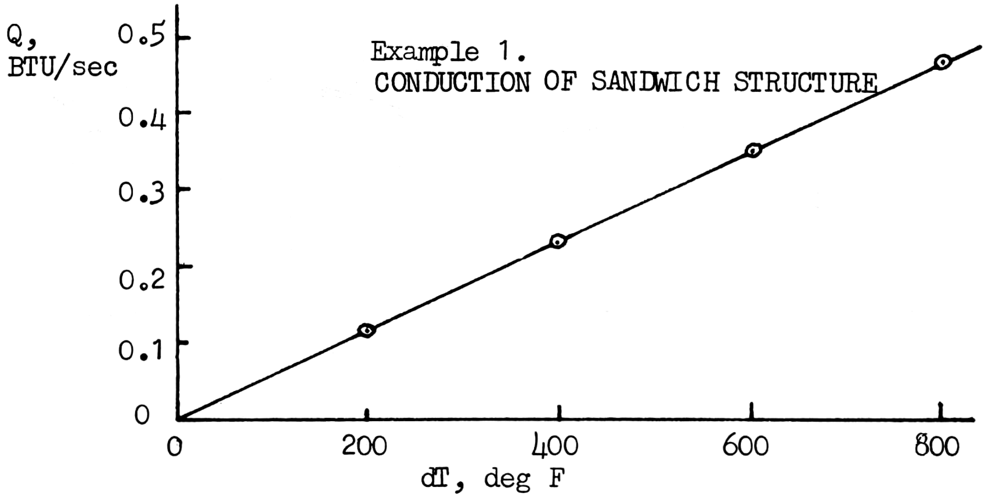
$$= 1250. + 455. + 0.62$$

$$= 1705.$$

$$U = 5.86 \text{ x } 10^{-4} \text{ BTU/sec-ft}^2\text{-deg F}$$

TABULATED RESULTS:

dT	Q
0.0 deg F	0.0 BTU/sec
200.	0.12
400.	0.23
600.	0.35
800.	0.47



Example 1.
CONDUCTION OF SANDWICH STRUCTURE

CONVECTION

Convection heat transfer is a modified conduction process with fluid in motion. Free convection is gravity induced flow; forced convection is fluid motion induced by forces other than gravity. Gravity force acts on the fluid to make it move as a result of the lowered density of the heated portion.

Convection can result in either laminar or turbulent flow. Convective currents can be observed in still air by the smoke rising from a cigarette. The smoke rises initially undisturbed as a laminar jet; at some inches above the tip the smoke becomes turbulent and the smoke particles diffuse. Convective currents can also be observed by diffraction of light in clear water heated from below.

The rate of energy transfer in convective films can be correlated by use of the Grashof Number, Gr, and the Prandtl Number, Pr. The transition from laminar to turbulent flow occurs at a value of the product of the Grashof Number and the Prandtl Number equal to 1×10^9.

FREE CONVECTION FORMULAE

PRANDTL NUMBER

$$Pr = g\, c_p \mu / k$$

where g = Gravity constant, 32.2 ft/sec^2
c_p = Specific Heat at constant pressure, BTU/lb-deg F
μ = Viscosity, $lb\text{-}sec/ft^2$
k = Thermal conductivity, BTU/sec-ft-deg F

GRASHOF CONSTANTS

$$C_4 = g \beta \rho^2 / \mu^2$$

where β = Coefficient of thermal expansion, $ft^3/(ft^3\text{-}deg\ R)$
ρ = Mass density, $(lb/ft^3)/(32.2\ ft/sec^2)$

$$C_3 = C_4\, L^3$$

where L= Length of vertical plate, ft

GRASHOF NUMBER

$$Gr = C_3\, dT$$

where dT = Temperature difference between wall and fluid, deg F

TEST FOR TURBULENCE

If Gr Pr is more than 1×10^9 flow is turbulent

If Gr Pr is less than 1×10^9 flow is laminar

LAMINAR FLOW, FREE CONVECTION, VERTICAL FLAT PLATE

1. Test for Equation Validity

   Gr Pr must exceed 1×10^4

2. Find average Nusselt Number, Nu

   $Nu = 0.508\ Pr^{0.5}\ Gr^{0.25} / (0.952 + Pr)^{0.25}$

3. Find heat transfer coefficient, h

   $h = Nu/(L/k)$

4. Find heat flux, Q/A, $BTU/sec\text{-}ft^2$

   $Q/A = h\ dT$

TURBULENT FLOW, FREE CONVECTION, VERTICAL FLAT PLATE

1. Find heat transfer coefficient, h

   $h = 0.23\ k\ C_4^{0.33}\ P_r^{0.33}\ dT^{0.33}$

2. Find heat flux, Q/A, $BTU/sec\text{-}ft^2$

Example 2. FREE CONVECTION, VERTICAL FLAT PLATE LAMINAR

DATA:

$L = 1.0$ ft

$dT = 5.0$ deg F

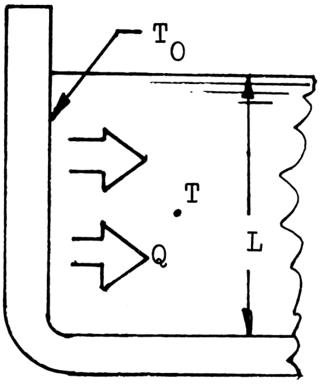
$k = 9.44 \times 10^{-5}$ BTU/sec-ft-deg F

$\beta = 0.08 \times 10^{-3}$ (1/deg R)

$\rho = 62.4 / 32.2 = 1.94$ slugs/ft^3

$\mu = 2.34 \times 10^{-5}$ lb-sec/ft^2

$c_p = 1.0$ BTU/lb-deg F



CALCULATION RESULTS:

See Program 3. Free Convection, Vertical, Laminar

$Pr = 32.2 \times 1.0 \times (2.34 \times 10^{-5})/(9.44 \times 10^{-5})$

$= 7.98$

$C_4 = 32.2 \times (0.08 \times 10^{-3}) \times (1.94)^2/(2.34 \times 10^{-5})^2$

$= 1.77 \times 10^7$

$C_3 = (1.77 \times 10^7) \times (1.0)^3 = 1.77 \times 10^7$

$Gr = (1.77 \times 10^7) \times (5.0) = 8.85 \times 10^7$

$Gr\ Pr = (8.85 \times 10^7) \times 7.98 = 7.06 \times 10^8$

$Nu = 0.508\ (7.98)^{1/2}\ (8.85 \times 10^7)^{1/4}\ /(0.952 + 7.98)^{1/4}$

$= 80.29$

$h = 80.29 \times (9.44 \times 10^{-5})/1.0$

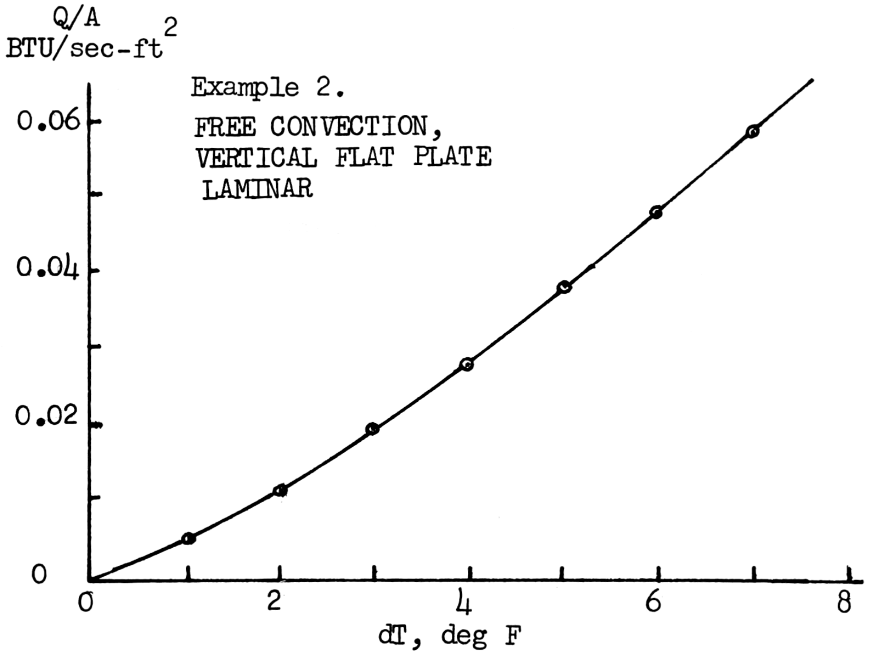
$= 0.0076$ BTU/sec-ft^2-deg F

$Q/A = 0.0076 \times 5.0 = 0.0379$ BTU/sec-ft^2

TABULATED RESULTS:

Example 2 Free Convection, Vertical Flat Plate, Laminar

dT deg F	Q/A BTU/sec-ft^2	Gr Pr
8.	----	1.12×10^9
7.	0.0577	9.79×10^8
6.	0.0476	8.39×10^8
5.	0.0379	7.08×10^8
4.	0.0287	
3.	0.0200	
2.	0.0121	
1.	0.0051	



Example 3. FREE CONVECTION, VERTICAL FLAT PLATE, TURBULENT

DATA:

Length of plate, $L = 4.0$ ft

Temperature of air, $T_{air} = 80.$ deg F

Temperature of plate, $T_{plate} = 0.$ deg F

Air film properties:

$Pr = 0.72$ $\qquad$ $\mu = 3.45 \times 10^{-7}$ lb-sec/ft^2

$c_p = 0.239$ BTU/lb-deg F

$k = 3.77 \times 10^{-6}$ BTU/sec-ft-deg F

$\beta = 1/(460 + \text{avg. temp.})$

$= 1/500. = 2.0 \times 10^{-3}$ deg F^{-1}

$\rho = P/(gRT)$

$= (14.7 \times 144)/(32.2 \times 53.3 \times 500) = 0.0025$ slugs/ft^3

CALCULATION RESULTS: See Program 4. Free Convection, Vertical

$C_4 = 32.2 \times (2.0 \times 10^{-3}) \times 0.0025)^2/(3.45 \times 10^{-7})^2$

$= 3.38 \times 10^6$

$C_3 = (3.38 \times 10^6) \times (4.0)^3 = 2.16 \times 10^8$

$Gr = (2.16 \times 10^8) \times (80 - 0) = 1.73 \times 10^{10}$

$Gr\ Pr = (1.73 \times 10^{10}) \times 0.72 = 1.24 \times 10^{10}$

(turbulent flow;

(exceeds transition value of 1×10^9)

$h = 0.23\ k\ (C_4\ dT)^{0.33}\ Pr^{0.33}$

$= 0.23 \times (3.77 \times 10^{-6}) \times (3.38 \times 10^6)^{0.33}$

$\times (80-0)^{0.33} \times 0.72^{0.33}$

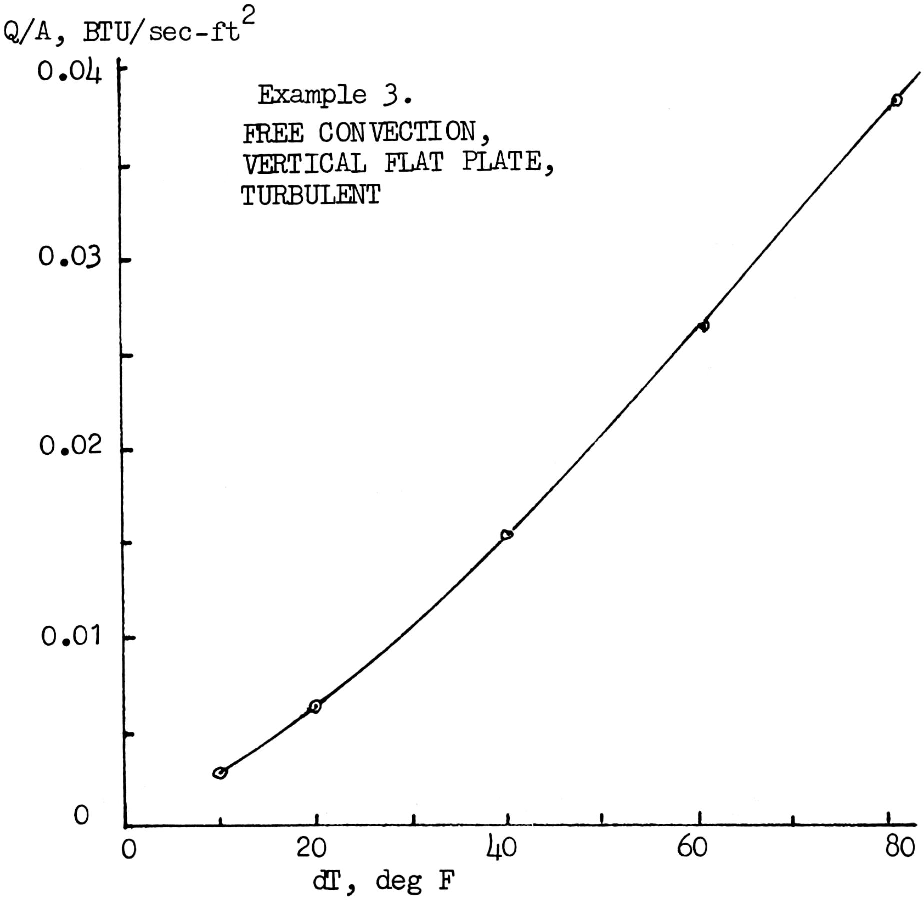
$= 4.72 \times 10^{-4}$ BTU/sec-ft^2-deg F

$Q/A = (4.72 \times 10^{-4}) \times (80-0) = 0.0378$ BTU/sec-ft^2

TABULATED RESULTS:

(Example 3 Free Convection, Vertical Flat Plate, Turbulent)

dt deg F	Q/A BTU/sec-ft^2	Gr Pr
10.	0.00237	1.56×10^9
20.	0.0060	3.11×10^9
30.	0.0102	
40.	0.0150	
50.	0.0202	
60.	0.0257	
80.	0.0377	



FORCED CONVECTION IN ROUND TUBES

Forced convection in round tubes can be separated into laminar and turbulent cases. For the laminar case at a transition Reynolds Number of 3000, based on the tube diameter, the Nusselt Number is equal to 4.36 for round tubes with uniform heat flux. For the turbulent case, Rn above 3000, the Nusselt Number is a function of the Reynolds and Prandtl Numbers, see Ref. 2.

The complete solution of heat transfer in tubes must include coefficients for the wall itself and for exterior film coefficients.

Fluid pressure drop is an important consideration in the design of heat transfer equipment. Pressure drop can be calculated by the methods of Ref. 3, Basic Fluid System Analysis. The fluid properties which cause pressure drop, that is, friction losses, likewise cause heat transfer. There is an analogy between friction and heat transfer, known as the "j factor", and has been investigated by Colburn, Ref. 4. The j factor is approximately one-eight the f factor used in computing friction loss.

CORRELATION OF FORCED CONVECTION IN ROUND TUBES

LAMINAR FLOW, Rn less than 3000.

$Nu = 4.36 = h\,D/k \qquad f = 64/Rn$

where h = Heat transfer coefficient, BTU/sec-ft^2-deg F

D = Tube flow diameter, ft

k = Thermal conductivity, BTU/sec-ft-deg F

TURBULENT FLOW, Rn more than 3000.

$Nu = 0.023\ Rn^{0.8}\ Pr^{0.4} \qquad f = 0.184\ Rn^{-0.20}$

where $Nu = h\,D/k \qquad Pr = g\ c_p \mu/K \qquad Rn = D\ V \rho/\mu$

FORCED CONVECTION, FLAT PLATE, LAMINAR

This condition applies for a Reynolds Number of less than 500,000 and a Prandtl Number of greater than 0.5. It is the boundary layer value that is determinant rather than the free stream. The condition is encountered even at very high speed flight since the leading edge has a "stagnation point" of zero velocity. The velocity in the boundary layer increases with the distance downstream until it reaches a turbulent value and is eventually reaches the free stream condition. Experimental data show that the heat flux can be correlated in terms of the Prandtl Number and the Reynolds Number, see Ref. 1.

CORRELATION OF FORCED CONVECTION, FLAT PLATE, LAMINAR

$$Nu = 0.664\ Pr^{1/3}\ Rn^{1/2}$$

where Nu = Nusselt Number, $h\ (L/k)$

Pr = Prandtl Number, $g\ c_p \rho / k$

Rn = Reynolds Number, $L\ V \rho / \mu$

PROCEDURE FOR PROBLEM SOLUTION

1. Test for Equation Validity

    1a. Reynolds Number must be less than 500,000.

    1b. Prandtl Number must be more than 0.50

2. Find Nusselt Number by use of Rn and Pr as above.

3. Find heat transfer coefficient, h

    $h = Nu/(L/k)$

4. Find heat flux, Q/A

    $Q/A = h\ dT$

Example 4. FORCED CONVECTION, LAMINAR, FLAT PLATE

DATA: $L = 1.5$ ft

$k = 9.44 \times 10^{-5}$ BTU/sec-ft-deg F

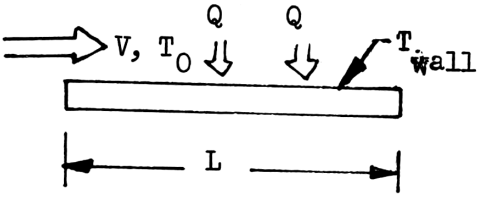
$V = 4.0$ ft/sec

$\rho = 1.94$ slugs/ft^3

$\mu = 2.34 \times 10^{-5}$ lb-sec/ft^2

$dT = 100.$ deg F

$c_p = 1.0$ BTU/lb-deg F



CALCULATION RESULTS: See Program 5. FORCED CONVECTION, LAMINAR, FLAT PLATE

$Rn = 1.5 \times 4.0 \times 1.94/(2.34 \times 10^{-5})$

$= 4.97 \times 10^5$

$Pr = 32.2 \times 1.0 \times (2.34 \times 10^{-5})/(9.44 \times 10^{-5})$

$= 7.98$

$Nu = 0.664 \times (7.98)^{1/3} \times (4.97 \times 10^5)^{1/2}$

$= 935.$

$h = 935./(1.5/(9.44 \times 10^{-5}))$

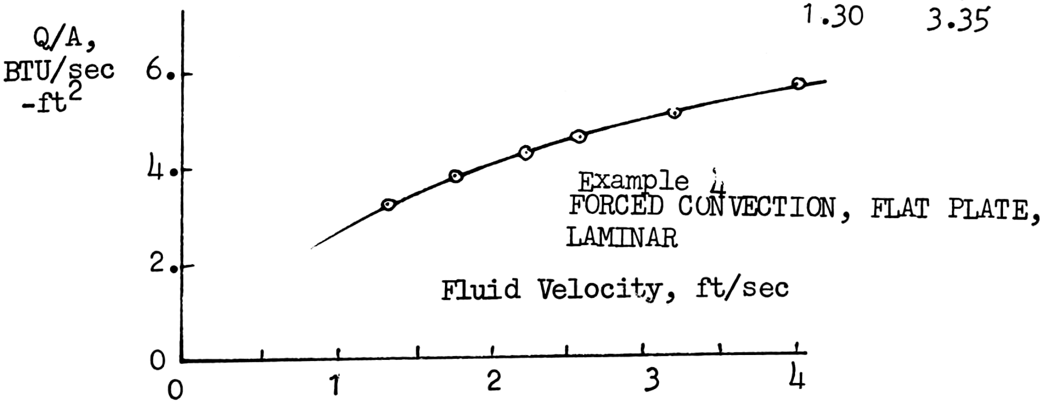
$= 0.0588$ BTU/sec-ft^2-deg F

$Q/A = 0.0588 \times 100.$

$= 5.88$ BTU/sec-ft^2

TABULATED RESULTS:

V ft/sec	Q/A BTU/sec-ft^2
4.0	5.88
3.26	5.30
2.66	4.79
2.17	4.32
1.77	3.90
1.30	3.35



PRESSURE DROP IN SMOOTH, ROUND TUBES

$$dP = (f\ (L/D) + K)) \times (1/2)\ \rho V^2$$

where dP = Pressure drop, lb/ft^2

L = Length of pipe, ft

D = Diameter (inside) of pipe, ft

K = Fitting loss factors

ρ = Mass density, $(lb/ft^3)/(32.2\ ft/sec^2)$

V = Velocity of fluid, ft/sec

COLBURN j FACTOR

$$j = St\ Pr^{2/3}$$

St = Stanton Number, $= Nu/(Rn\ Pr) = h/(c_p\ G)$

where G = Specific flow rate, $(lb/sec)/ft^2$

also, by experiment it is found that, approximately

$$j = 0.023\ Rn^{-0.20}$$

therefore by comparison to the friction factor

$$j = f \times (0.023/0.184)$$

$$j = f/8$$

PROCEDURE FOR FORCED CONVECTION IN ROUND TUBES

See
APPENDIX I. Program 6. Forced Convection, Round Tubes,
Laminar and Turbulent

1. Find Reynolds Number, Rn

$$Rn = D\ V\rho/\mu$$

$$= dW/(25.25\ D\mu)$$

where dW = Flow rate, lb/sec

2. Test for laminar/turbulent flow

If Rn less than 3000, flow is laminar, go to Step 3.

If Rn more than 3000, flow is turbulent, go to Step 4.

3. Find heat transfer coefficient, h, laminar flow

$$Nu = 4.36 = h\ D/k \qquad f = 64/Rn$$

$$h = 4.36 \times (k/D)$$

(Go to Step 5)

4. Find heat transfer coefficient, h, turbulent flow

$$Pr = g\ c_p\mu/k \qquad f = 0.184\ Rn^{-0.20}$$

$$Nu = 0.023\ Rn^{0.023}\ Pr^{0.4}$$

$$h = Nu \times (k/D)$$

5. Find pressure drop

$$G = dW/(0.7854\ D^2)$$

$$V = G/(g\rho)$$

$$q = (1/2)\ \rho\ V^2$$

where q = dynamic pressure, lb/ft^2

$$dP = (f\ (L/D) + K)\ q$$

6. Find Colburn j factor

$$St = h/(c_p\ G)$$

$$j = St\ Pr^{2/3}$$

EXAMPLE 5. FORCED CONVECTION IN ROUND TUBES

DATA:

$D = 0.0254$ ft

$\mu = 0.00008$ lb-sec/ft^2

$k = 3.6 \times 10^{-5}$ BTU/sec-ft-deg F

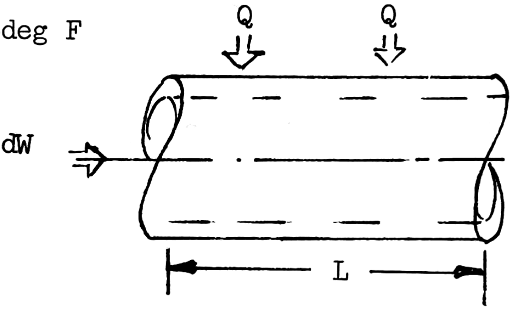
$c_p = 0.855$ BTU/lb-deg F

$dW = 0.30$ lb/sec

$g\rho = 48.0$ lb/ft^3

$L = 100.$ ft

$K = 3.0$ (fitting loss factor)



CALCALATION: See Program 6. FORCED CONVECTION, ROUND TUBES

ENERGY TRANSFER

$Pr = 32.2 \times 0.855 \times (0.00008)/(3.6 \times 10^{-5}) = 61.18$

$Rn \quad 0.30/(25.25 \times 0.254 \times 0.00008) = 5847.$

Flow is turbulent

$Nu = 0.023 \times (5847)^{0.8} \times (61.18)^{0.4}$

$= 123.$

$h = 123. \times (3.6 \times 10^{-5})/0.0254$

$= 0.174$ BTU/sec-ft^2-deg F

PRESSURE DROP

$G = 0.30/(0.7854 \times (0.0254)^2)$

$= 592.$ lb/sec-ft^2

$V = 592./48.0$

$= 12.33$ ft/sec

$f = 0.184 \times (5847)^{-0.20}$

$= 0.0325$

$q = (1/2) \times (48.0/32.2) \times (12.33)^2$

$= 113.31 \ \mathrm{lb/ft^2}$

$dP = (0.0325 \times (100./0.0254) + 3.0) \times 113.31$

$= 14509. \ \mathrm{lb/ft^2}$ or $100. \ \mathrm{lb/in^2}$

COLBURN j FACTOR

$St = 0.174/(0.855 \times 592.$

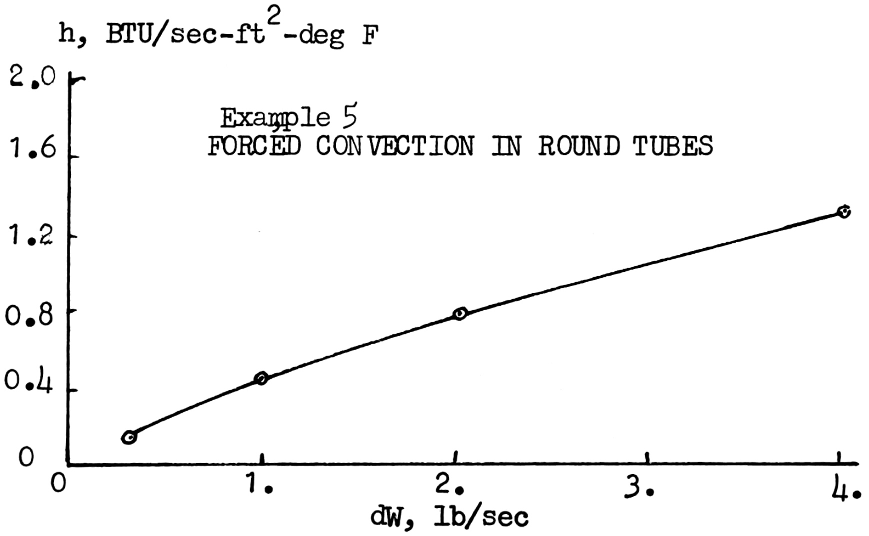
$= 0.000343$

$j = (0.000343) \times (61.18)^{2/3}$

$= 0.00532$

TABULATED RESULTS

dW lb/sec	h BTU/sec-ft^2-degF	dP psi
0.30	0.174	100.
1.0	0.457	
2.0	0.795	
4.0	1.385	


Example 5
FORCED CONVECTION IN ROUND TUBES

FORCED CONVECTION, FLAT PLATE, HIGH VELOCITY

This condition applies especially to aerodynamic heating of aircraft and missiles during supersonic flight at high altitude. For that purpose equations are given below for computing the ambient temperature and pressure at altitude.

Energy transfer in a high velocity flow must consider the kinetic energy of the fluid. This is done by computing the theoretical "stagnation temperature" that would result by bringing the fluid slowly and frictionlessly to rest, then reducing the temperature to account for losses.

The temperature which would result at the plate with no transfer of energy by the plate is known as the "adiabatic temperature" and is computed as above but with the addition of a recovery factor, r.

The temperature of the fluid at the boundary layer should be used to evaluate the fluid properties. This requires a knowledge of the actual wall temperature which is usually unknown. It may, therefore, be necessary to estimate the wall temperature, compute the temperature, and repeat the calculation with improved fluid values. The equation for computing boundary layer temperature is given by Ref. 5.

FORMULAE FOR FORCED CONVECTION, FLAT PLATE, HIGH VELOCITY

AMBIENT TEMPERATURE AND PRESSURE

$$T_0 = 519.69\ e^{-0.007888\ H}$$

$$P_0 = 2116.\ e^{-0.0414\ H}$$

where T_0 = Temperature, deg R

P_0 = Pressure, lb/ft^2

H = Altitude, 1000. ft

RAM TEMPERATURE AND RAM PRESSURE

$$T_s = T_0\ (1 + ((\gamma - 1)/2)\ Mn^2) \qquad \text{where} \qquad \gamma = c_p/c_v$$

$$P_s = P_0\ (1 + ((\gamma - 1)/2)\ Mn^2)^{\gamma/(\gamma - 1)} \qquad = 1.4 \text{ for air}$$

where T_s = Stagnation temperature, deg R

P_s = Stagnation pressure, lb/ft^2

Mn = Mach Number = V/V_c

where V_c = Velocity of sound in fluid, ft/sec

$= (\gamma\ g\ R\ T_0)^{1/2}$

R = Gas constant,(53.3 for air)

RECOVERY FACTOR, r

LAMINAR BOUNDARY, Rn less than 500,000, Pr of 0.5 to 5.0

$r = Pr^{1/2}$ (where $Pr = g\, c_p \mu / k$; $Rn = dV \rho / \mu$)

TURBULENT BOUNDARY, Rn more than 500,000, Pr of 0.5 to 2.0

$r = Pr^{1/3}$

ADIABATIC WALL TEMPERATURE

$Taw = T_0 (1 + r((\gamma -1)/2\, Mn^2)$

where Taw = Adiabatic wall temperature, deg R

BOUNDARY LAYER TEMPERATURE

$T_{bl} = T_{fs} + 0.50 (T_{wall} - T_{fs}) + 0.22 (T_{aw} - T_{fs})$

where T_{bl} = Temperature of boundary layer, degR

T_{fs} = Temperature of free stream, deg R

T_{wall} = Temperature of wall, deg R

NUSSELT NUMBER

LAMINAR BOUNDARY

$Nu = 0.332\, Rn^{1/2}\, Pr^{1/3}$

TURBULENT BOUNDARY

$Nu = 0.029\, Rn^{0.8}\, Pr^{1/3}$

where $Nu = h\, D/k$

PROCEDURE FOR FORCED CONVECTION, FLAT PLATE, HIGH VELOCITY

Program 7. Ambient Temperature and Compressible Temp
Program 8. Ambient Pressure and Compressible Pressure
Program 9. Forced Convection, Flat Plate, High Velocity (Parts 1 to 4)

1. Find altitude temperature and pressure and air density

$$T_0 = 519.69^{-0.007888\ H}$$

$$P_0 = 2116.\ e^{-0.0414\ H}$$

$$\rho = P/(gRT)$$

2. Find stagnation temperature and velocity of sound

$$T_s = T_0\ (1+((\gamma\ -1)/2)\ M_n^{\ 2}) \qquad V_c = (k\ g\ R\ T_0)^{1/2}$$

3. Find Prandtl Number and Reynolds Number

$$Pr = g\ c_p \mu/k \qquad Rn = D\ V_c\ \rho/\mu$$

4. Find recovery factor

For laminar flow (Rn less than 500,000)

$$r = Pr^{1/2}$$

For turbulent flow (Rn more than 500,000)

$$r = Pr^{1/3}$$

5. Find adiabatic wall temperature

$$T_{aw} = T_0\ (1\ +\ r\ ((\gamma\ -1)/2)\ Mn^2$$

6. Find boundary layer temperature

$$T_{bl} = T_0 + 0.50\ (T_{wall} - T_0) + 0.22\ (T_{aw} - T_0)$$

7. Find Nusselt Number

For laminar flow

$$Nu = 0.332\ Rn^{1/2}\ Pr^{1/3}$$

For turbulent flow

$$Nu = 0.029\ Rn^{0.8}\ Pr^{1/3}$$

8. Find heat transfer coefficient and heat flux

$$h = Nu/(L/k) \qquad Q/A = h\ (T_{aw} - T_{wall})$$

9. Repeat calculations as needed with improved values for Pr and Rn

EXAMPLE 6. FORCED CONVECTION, FLAT PLATE, HIGH VELOCITY

DATA: H = (35,000 ft/1000) = 35.

Mn = 1.5

L = 10. ft

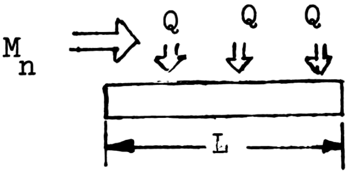
c_p = 0.241 BTU/lb-deg F

k = 6.94×10^{-6} BTU/lb-ft-deg F

μ = 3.97×10^{-7} lb-sec/ft^2

γ = 1.4

R = 53.3



CALCULATION RESULTS: (T_{wall} = 400. deg R)

$T_0 = 519.69\ e^{-0.007888\,(35)}$ = 394. deg R

$P_0 = 2116.\ e^{-0.0414(35)}$ = 496. lb/ft^2

$g\rho$ = 496./(53.3 x 394) = 0.0236 lb/ft^3

ρ = 0.0236/32.2 = 7.33×10^{-4} slugs/ft^3

T_s = $394\ (1 + ((1.4-1)/2\ (1.5)^2))$ = 571. deg R

Pr = $32.2 \times 0.241 \times (3.97 \times 10^{-7})/(6.94 \times 10^{-6})$ = 0.44

V_c = $(1.4 \times 32.2 \times 53.3 \times 394)^{1/2}$ = 973. ft/sec

Rn = $10. \times 973. \times (7.33 \times 10^{-4})/(3.97 \times 10^{-7})$ = 1.80×10^7

Flow is turbulent

r = $(0.44)^{1/3}$ = 0.76

T_{aw} = $394.\ (1 + 0.76\ ((1.4 - 1)/2\ (1.5)^2))$ = 529. deg R

T_{bl} = 394. + 0.50 (400. - 394.) + 0.22 (529. - 394) = 430. deg R

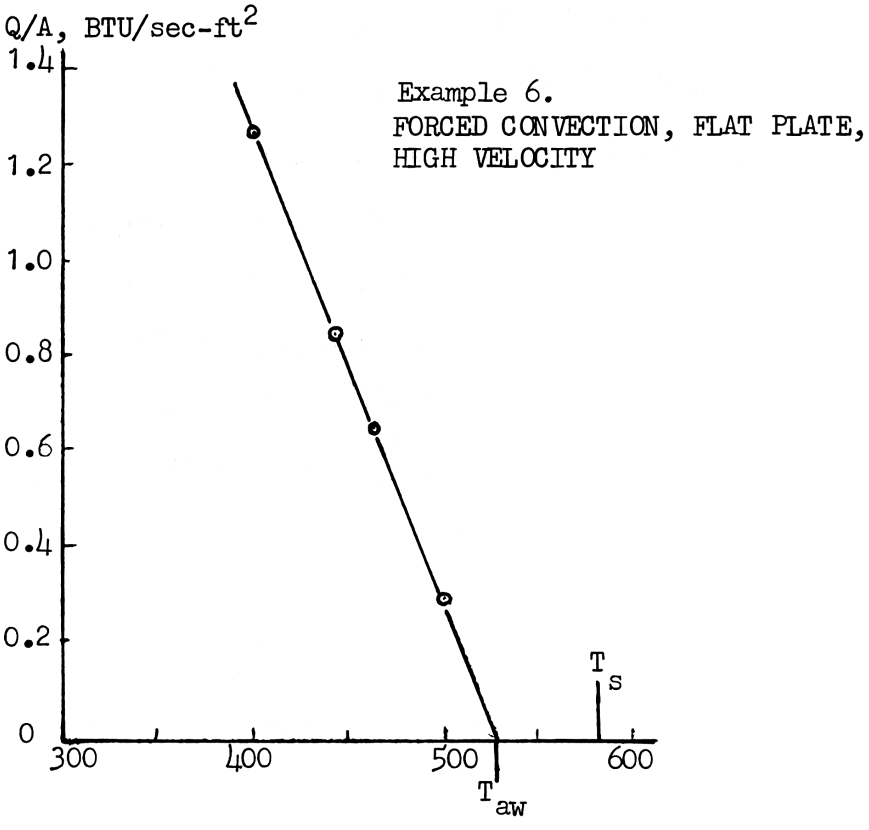
Nu = .0 29 $(1.80 \times 10^7)^{0.8}\ (0.44)^{1/3}$ = 1.41×10^4

h = $(1.41 \times 10^4)/(10/6.94 \times 10^{-6})$ = 0.0098 BTU/sec-ft^2-deg F

Q/A = 0.0098 x (529. - 400.) = 1.26 BTU/sec-ft^2

TABULATED VALUES:

T_{wall} deg R	Q/A BTU/sec-ft^2
400.	1.26
440.	0.869
460.	0.673
500.	0.283
520.	0.088



Example 6.
FORCED CONVECTION, FLAT PLATE,
HIGH VELOCITY

BOILING

NUCLEATE BOILING

Nucleate boiling is characterized by the appearance of small gas bubbles which pervade the liquid. The bubbles originate from surface imperfections, cracks, crevices and tool marks which are not wetted by the liquid because of surface tension.

Nucleate boiling is initiated at wall temperature which is 0 to 10 degrees F above the liquid saturation temperature. The bubbles at the liquid-wall interface grow by absorbing vaporized gas. These bubbles become detached from the wall when sufficiently large because of surface tension and rise to the surface by gravity. The residual gas in the crevice becomes the source for the new bubble. The bubble rises to the surface with the heat transmitted by the wall for its formation.

The Rohsenow equation for nucleate boiling is widely used. It requires experimental determination of a liquid-surface coefficient, see Ref. 6 and 7. Impurities of the surface and liquid, surface finish and other irregularities can change these values.

At a sufficiently high temperature of the wall above the liquid saturation temperature (the driving temperature difference) the bubbles formed at the wall become so numerous that a continuous vapor film is formed. Nucleate boiling proceeds in this way through a transition to film boiling. The transition is accompanied by a low heat flux with possible wall overheat, or burnout. Peak nucleate heat flux can be found by Noyes' equation, see Ref. 8 and 9, which accounts for gravity effects.

ROHSENOW HEAT FLUX EQUATION:

$$Q/A = (c_p\,(T_{wall} - T_{sat})/(h_{fg}\,c_{sf}\,P_r^{1.7}))^3\ g\mu_{liq}\ h_{fg}\ ((g/g_o)(g\rho_{liq} - g\rho_{vap})/\sigma)^{1/2}$$

where Q/A = Heat flux, BTU/sec-ft^2
c_p = Heat capacity of liquid, BTU/lb-deg F
$T_{wall} - T_{sat}$ = Temperature difference between saturated liquid and wall, deg F
h_{fg} = Latent heat of evaporation, BTU/lb
c_{sf} = Surface coefficient, 0.003 to 0.015

P_r = Prandtl Number of liquid = $g\,c_p\,\mu/k$
μ = Viscosity of liquid, lb-sec/ft^2
$g\rho_{liq} - g\rho_{vap}$ = Density of liquid minus that of vapor, lb/cu ft
σ = Surface tension, lb/ft
k = Thermal conductivity, BTU/sec-ft-deg F

NOYES PEAK NUCLEATE HEAT FLUX:

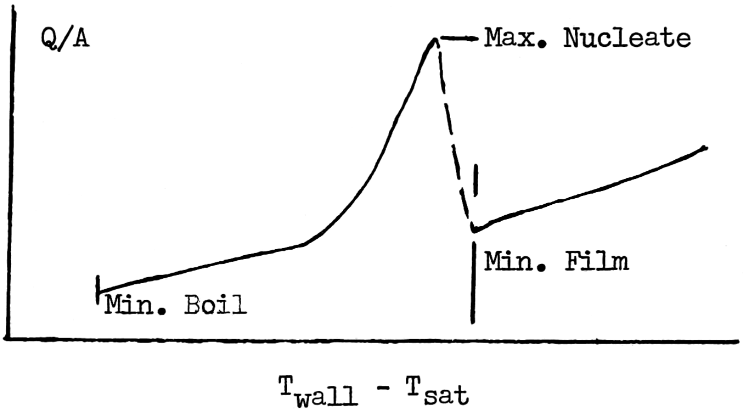
$$(Q/A)_{max} = 0.144\ h_{fg}\ (g\rho_{vap})^{1/2}$$
$$(((g\rho_{liq} - g\rho_{vap})^2/g\rho_{liq})\ g\ g_0\sigma)^{1/4}$$
$$Pr^{-0.245}\ (a/g)^{1/4}$$

where $(Q/A)_{max}$ = Max heat flux of nucleate boiling, BTU/sec-ft^2

a/g = Local gravity to standard gravity ratio

g_0 = Conversion factor; mass to force, 32.2 ft/sec^2

FIGURE 12. CHARACTERISTIC BOILING PLOT



PROCEDURE FOR NUCLEATE BOILING

See APPENDIX I. Program 10. Nucleate Pool Boiling
Program 11. Peak Nucleate Heat Flux

1. Find Prandtl Number of liquid

$$Pr = g\ c_p\mu/k$$

2. Find constant C_1

$$C_1 = ((g/g_0)\ ((g\rho_{liq} - g\rho_{vap})/\sigma))^{1/2}$$

3. Find Q/A

$$Q/A = (c_p\ (T_{wall} - T_{sat})/(h_{fg}\ c_{sf}\ Pr^{1.7}))^3\ g\mu_{liq}\ h_{fg}\ C_1$$

4. Find peak heat flux

$$(Q/A)_{max} = 0.144\ h_{fg}\ (g\rho_{vap})^{1/2}\ \ldots\ \text{etc.,as above}$$

EXAMPLE 7 . NUCLEATE POOL BOILING

DATA:

T_{sat} =(212 deg F + 460 deg F) = 672 deg R

T_{wall} = (672 deg R + 20 deg F) = 692 deg R

h_{fg} = 970. BTU/lb

c_p = 1.0 BTU/lb-deg F

c_{sf} = 0.013

μ = 6.38×10^{-6} lb-sec/ft^2

σ = 0.004 lb/ft

g = 32.2 ft/sec^2

$g\rho_{vap}$ = 0.037 lb/ft^3

$g\rho_{liq}$ = 62.4 lb/ft^3

k = 1.09×10^{-4} BTU/ft-sec-deg F

CALCULATION RESULTS:

$$Pr = 32.2 \times 1.0 \times (6.38 \times 10^{-6})/(1.09 \times 10^{-4}) = 1.88$$

$$C_1 = ((32.2/32.2)((62.4-0.037)/0.004))^{1/2} = 124.$$

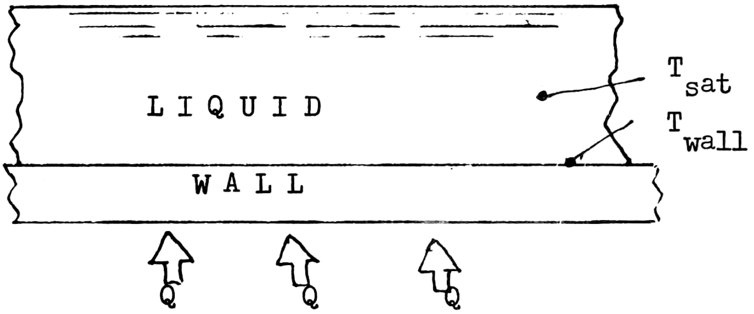
$$Q/A = (1.0 \times (692-672)/(970. \times 0.013 \times 1.88^{1.7}))^3$$
$$\times 32.2 \times (6.38 \times 10^{-6}) \times 970. \times 124.$$
$$= 3.94 \text{ BTU/sec-ft}^2$$

$$(Q/A)_{max} = 0.144 \times 970. \times (0.037)^{1/2} \times((62.4 - 0.037)^2/62.4$$
$$\times 32.2 \times 32.2 \times 0.004)^{1/4} \times 1.88^{-0.245} (1.0)^{1/4}$$
$$= 92.29 \text{ BTU/sec-ft}^2$$

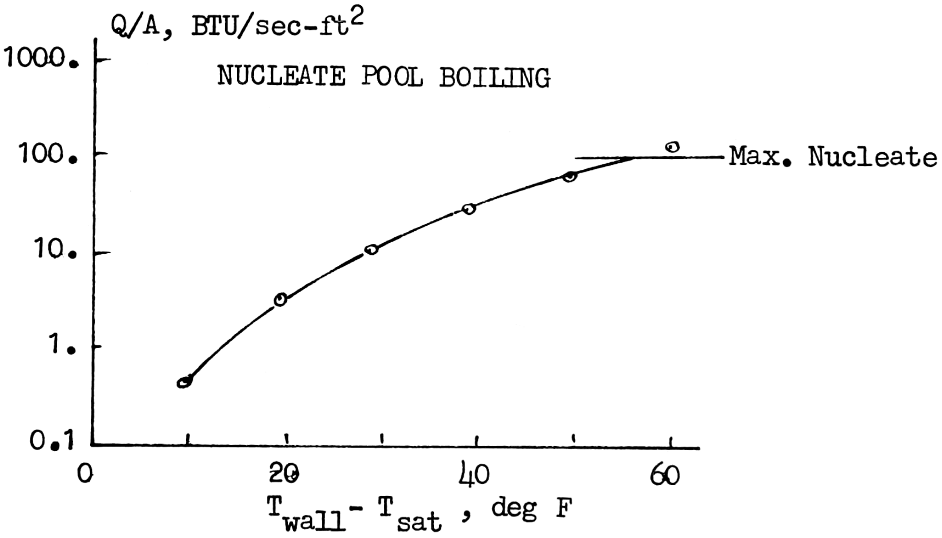
TABULATED RESULTS:

$T_{wall} - T_{sat}$ deg F	Q/A BTU/sec-ft^2
10.	0.47
20.	3.73
30	12.58
40	29.81
50	58.22
60	100.60

$(Q/A)_{max} = 92.$ BTU/sec-ft^2



Example 7. NUCLEATE POOL BOILING



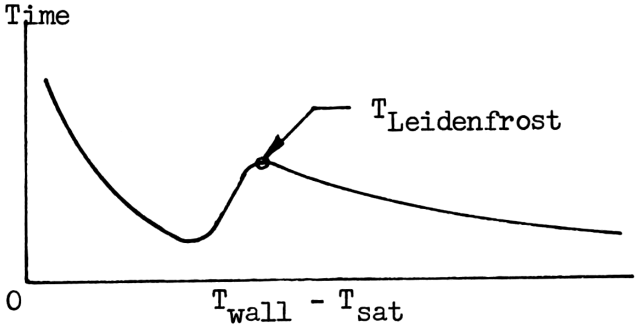
FILM BOILING

Film boiling is characterized by the complete separation of the liquid from the heated surface by the vapor. The rate of heat transfer does not depend on the surface characteristics.

The temperature at which film boiling is attained can be correlated by the Leidenfrost temperature according to the experiments of Baumeister and Simon, Ref. 10. The Leidenfrost point is found by measuring the time for evaporation of a drop of liquid at increasing plate temperatures and occurs at the maximum time following the burnout point. See below.

A generalized correlation of film boiling was made by Clements and Colver, Ref. 11, based on data for water, oxygen, nitrogen and pure mixed hydrocarbons.

CHARACTERISTIC LEIDENFROST POINT



FORMULAE FOR FILM BOILING

See Appendix I. Program 12. Minimum Film Boiling Temp.

ISOTHERMAL LEIDENFROST POINT ESTIMATE

$$T_{Leid} = 1.51\, T_{crit}^{(1-e^{-m})}$$

where T_{Leid} = Leidenfrost point, deg R

T_{crit} = Critical temperature of fluid, deg R

$$m = 0.99\, (\sigma_{lv})^{-1/3}\, (\text{spgr}/\text{molwt})^{4/9}$$

where σ_{lv} = Surface tension, lb/ft

spgr = Specific gravity of wall

molwt = Molecular weight of wall

FILM BOILING EQUATION

See APPENDIX I. Program 13. Film Boiling Heat Flux

$$Nu = 0.94\,(Ra\,\Theta\,Tr^{-2})^{1/4}$$

where Nu = Nusselt Number, $h\,D/k_{vap}$

Ra = Rayleigh Number = $Gr\,Pr$

Gr = Grashof Number, $\rho_v\,g\,(\rho_{liq} - \rho_{vap})\,D^3/\mu_{vap}^2$

Pr = Prandtl Number, $g\,c_{p_{vap}}\,\mu_{vap}/k_{vap}$

Tr = Temperature ratio = T_{sat}/T_{crit}

Θ = Clement and Colver Number, $h_{fg}/(c_{p_{vap}}\,dT)$

h_{fg} = Latent heat of evaporation, BTU/lb

dT = Driving temperature, $T_{wall} - T_{sat}$

PROCEDURE FOR FILM BOILING

See APPENDIX I. Program 12. Minimum Film Boiling Temperature
Program 13 Film Boiling Heat Flux

1. Find minimum temperature for film boiling

$$T_{Leid} = 1.51\,T_{crit}^{(1-e^{-m})}$$

2. Find Nusselt Number

$$Nu = 0.94\,(Ra\,\Theta\,Tr^{-2})^{1/4}$$

3. Find heat flux

$$h = Nu\,k_v/D$$

$$Q/A = h\,dT$$

EXAMPLE 8. FILM BOILING

DATA:

T_{sat} =(212 deg F) 672 deg R

T_{crit} = 1165 deg R

T_{wall} = 860 deg R

$spgr_{wall}$ = 2.71

$molwt_{wall}$ = 26.9

σ_{liq} = 0.004 lb/ft

h_{fg} = 700. BTU/lb

Pr_{vap} = 0.94

$c_{p_{vap}}$ = 0.47 BTU/lb-deg F

μ_{vap} = 3.9×10^{-7} lb-sec/ft^2

ρ_{liq} = 1.86 slugs/ft^3

ρ_{vap} = 8.01×10^{-4} slugs/ft^3

D = 0.10 ft

k_{vap}=9.0×10^{-4} BTU/sec-ft-deg F

CALCULATION RESULTS:

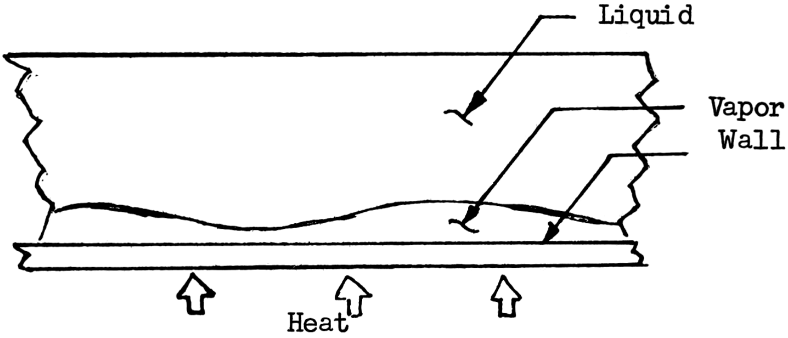
$$m = 0.99\ (0.004)^{-1/3}\ (2.71/26.9)^{4/9} = 2.25$$

$$T_{Leid} = 1.51\ (1165)^{(1-e^{-2.25})} = 836.\ \text{deg R (376 deg F)}$$

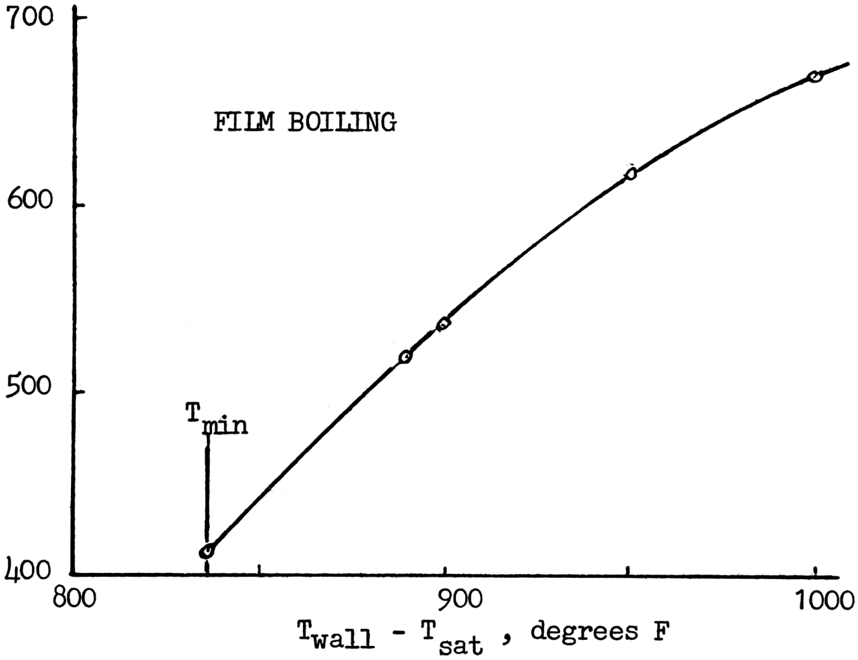
TABULATED RELULTS:

$Ra = 2.96 \times 10^{8}$

T_{wall} deg R	Θ ---	Nu ---	h BTU/sec-ft^2-deg F	Q/A BTU/sec-ft^2
836.	9.08	282.	2.54	415.
890.	6.83	262.	2.36	515.
900.	6.53	259.	2.34	532.
950.	5.36	246.	2.22	617.
1000.	4.54	236.	2.13	670.



Example 8. FILM BOILING



RADIATION

The existence of magnetic fields was demonstrated in 1850 by Michael Faraday using iron filings on a paper above a permanent magnet. When the paper is gently tapped the filings arrange themselves along the lines of force between the two magnetic poles.

A moving magnetic field produces a flow of electricity in a conductor; a flow of electricity in a conductor produces a magnetic field. The two are related according to laws discovered by James Clerk Maxwell, a Scottish physicist in 1864. Electromagnetic waves are characterized by a frequency and a wave length. In order of increasing frequency we find radio waves, infrared, visible light, ultraviolet, X-rays, gamma rays and cosmic rays. Thermal radiation occurs within the infrared to ultraviolet bands.

Thermal emission is a function only of the fourth power of the absolute temperature. A body emits thermal radiation in all directions. The radiation intensity of a given radius can be considered to pass through a sphere of that radius and is accordingly diminished by the square of the radius ratio.

The average solar radiation at the earth's surface at noon on a clear day facing the sun is approximately 0.089 BTU/sec-ft^2. The amount absorbed will be reduced by the surface absorptivity factor. By comparison a pound of petroleum fuel with a typical energy content of 18,000 BTU/lb would provide the energy of 59. square feet of solar panels for one hour.

FORMULAE FOR RADIATION AND PROCEDURE

See APPENDIX I. Program 14. Radiation Gain
Program 15. Radiation Loss

STEFAN-BOLTZMANN LAW

$$e = \sigma T^4$$

where e = Emissive power, BTU/sec-ft^2

σ = Stefan-Boltzmann constant, 4.758×10^{-13} BTU/sec-ft^2-deg R^4

T = Surface temperature, deg R

RADIATION HEAT LOSS

$$Q_{loss} = \sigma e_b A_b (T_b^4 - T_e^4)$$

where Q_{loss} = Heat loss, BTU/sec

e_b = Emissivity of body, 0.0 to 1.0

A_b = Area of radiating body, ft^2

T_b = Temperature of body, deg R

T_e = Temperature of background, deg R

RADIATION HEAT GAIN

$$Q_{gain} = \sigma\, e_b\, a_b\, (r_1/r_2)^2\, A_b\, (T_s^4 - T_b^4)$$

where Q_{gain} = Heat gain, BTU/sec

e_s = Emissivity of source, 0.0 to 1.0

a_b = Absorptivity of body, 0.0 to 1.0

r_1 = Radius of source, feet

r_2 = Radius from center of source to body, feet

A_b = Area of body exposed to radiation, $feet^2$

T_s = Temperature of source, deg R

EXAMPLE 9. RADIATION

DATA:

	SOURCE	SPACE	BODY ENERGY GAIN	 ENERGY LOSS
Emissivity, e	1.0			0.8
Absorptivity, a			0.9	
Surface Temperature,deg R	5000.	460.	500.	500.
Surface Area, ft^2			10.	100.
Radius , ft	0.5		10.	

CALCULATION RESULTS:

$$Q_{gain} = (4.758 \times 10^{-13}) \times 1.0 \times 0.9 \times (0.5/10.)^2 \times 10. \times (5000.^4 - 500.^4)$$

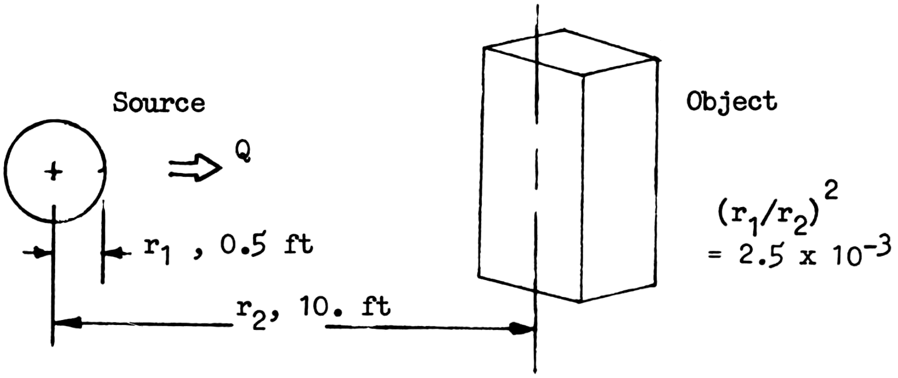
$$= 6.69 \text{ BTU/sec}$$

$$Q_{loss} = (4.758 \times 10^{-13}) \times 0.8 \times 100. \times (500.^4 - 460.^4)$$

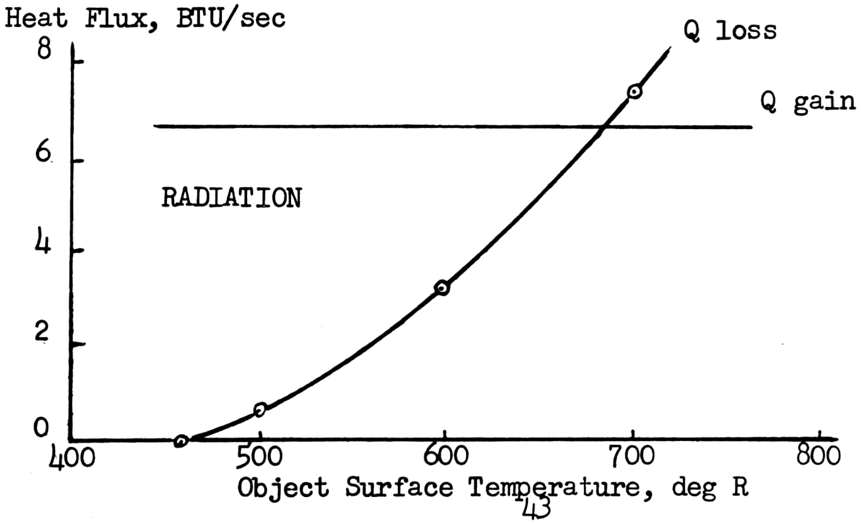
$$= 0.67 \text{ BTU/sec}$$

TABULATED RESULTS:

Body temperature deg R	Heat gain BTU/sec	Heat Loss BTU/sec
460.		0.
500.	6.69	0.67
600.	6.69	3.23
700.	6.69	7.43
3000.	5.82	



Example 9. RADIATION



HEAT EXCHANGER COMPARISON

Heat exchangers transfer energy from one fluid to another. The fluids can be both liquids, both gas or one of each. There may by a simple change of temperature or there may be a change of phase with condensation or evaporation. The fluids may both flow in the same direction (parallel flow) or in opposite directions (counterflow).

The counterflow heat exchanger is preferable if the aim is to heat the cooler fluid. The parallel flow heat exchanger is preferable if the aim is to cool the hotter fluid. The counterflow design is the most efficient.

The fluid may flow through tubes or across fins or plates. The design objective is to obtain the required surface area for each fluid at minimum cost, size and weight. For example, in a water-to-air heat exchanger the air side area will govern the design.

The heat exchanger may contain baffles to obtain several passes of the fluid in a tube-and-shell design. Tubes may contain internal fins or external fins to increase the surface area and to promote turbulence.

Heat exchanger surfaces may be coated with scale from deposits of liquid impurities or from liquid decomposition. Metal oxides can be present. These coatings reduce the heat transfer rate.

The design of heat exchangers is simplified by the NTU (Number of Transfer Units) concept devised by Kays and London, Reference 12. A transfer unit is the product of the heat transfer area and the heat transfer coefficient divided by the flow rate and the specific heat of the fluid least able to absorb the heat, C_{min}. The fluid most able to absorb the heat, by virtue of the greater product of flow rate and specific heat, is known as C_{max}. Efficiency can be determined from values for NTU and the ratio C_{min}/C_{max}.

Efficiency refers to the ratio of the heat transferred compared to the amount of heat that could be transferred in a counterflow heat exchanger of infinite area. At NTU values above 3 the efficiency increases very slowly.

FORMULAE FOR HEAT EXCHANGERS

NTU VALUE

$$N_{tu} = A\,U/C_{min}$$

where N_{tu} = Number of transfer units

A = Heat transfer area, ft^2

U = Heat transfer coefficient, $BTU/sec\text{-}ft^2\text{-}deg\ F$

C_{min} = Minimum product of hot or cold fluid flow rate x specific heat, BTU/sec-deg F

COUNTERFLOW EFFICIENCY

$$\text{eff.} = \frac{1 - e^{(-N_{tu}\,(1-(C_{min}/C_{max}))}}{1 - ((C_{min}/C_{max})\; e^{(-N_{tu}\,(1-(C_{min}/C_{max}))})}$$

PARALLEL EFFICIENCY

$$\text{eff.} = \frac{1 - e^{(-N_{tu}\,(1 + (C_{min}/C_{max})))}}{1 + (C_{min}/C_{max})}$$

HEAT TRANSFERRED BY FLUID C_{min}

$$Q = \text{eff.} \times C_{min} \times (T_{h1} - T_{c1})$$

wehre Q = Heat transfer rate, BTU/sec

T_{h1} = Temperature of hot fluid at inlet, deg F

T_{c1} = Temperature of cold fluid at inlet, deg F

PROCEDURE FOR HEAT EXCHANGER ANALYSIS

See APPENDIX I. Program 16, Heat Exchanger Effic., Parts 1-3.

1. Find C_{min} and ratio C_{min}/C_{max}

$C_h = dW_h\, c_{p_h}$

$C_c = dW_c\, c_{p_c}$.

where dW_h = Hot fluid flow rate, lb/sec

dW_c = Cold fluid flow rate, lb/sec

c_{p_h} = Specific heat of hot fluid, BTU/lb-deg F

c_{p_c} = Specific heat of cold fluid, BTU/lb-deg F

If C_h is less than C_c: $C_{min} = C_h$ and $C_{min}/C_{max} = C_h/C_c$

If C_c is less than C_h: $C_{min} = C_c$ and $C_{min}/C_{max} = C_c/C_h$

2. Find number of transfer units, N_{tu} (see above)

3. Find efficiency (see above)

4. Find heat transfer rate (see above)

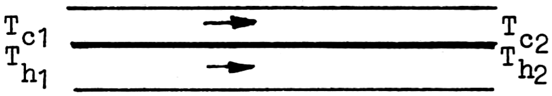
5. Find temperature change for each fluid.

$dT_h = Q/(dW_h\ c_{p_h})$

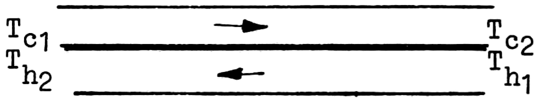
$dT_c = Q/(dW_c\ c_{p_h})$

EXAMPLE of HEAT EXCHANGER PERFORMANCE

DATA:
$dW_c = 2.5$ lb/sec
$dW_h = 1.5$ lb/sec
$c_{p_c} = 0.99$ BTU/lb-deg F
$c_{p_h} = 1.10$ BTU/lb-deg F
$A = 20.0\ ft^2$
$U = 0.25$ BTU/sec-ft^2-deg F



Parallel Flow



Counterflow

CALCULATION RESULTS:

$C_h = (1.5) \times 1.10 = 1.65$ BTU/sec-deg F

$C_c = 2.5 \times 0.99 = 2.48$ BTU/sec-deg F

$C_{min} = C_h = 1.65 \qquad C_{min}/C_{max} = 1.65/2.48 = 0.67$

$N_{tu} = (20.\ ft^2) \times (0.25$ BTU/sec-ft^2-deg F$) / (1.65) = 3.03$

Counterflow efficiency: Example 10

$$\text{eff.} = \frac{1 - e^{(-3.03 \times (1 - 0.67))}}{1 - ((0.67) \times e^{(-3.03 \times (1 - 0.67))})} = 0.84$$

$Q = 0.84 \times 1.65 \times 10.$ deg F $= 5.63$ BTU/sec

$dT_h = (5.63$ BTU/sec$)/((1.5$ lb/sec$) \times (1.10$ BTU/lb-deg F$))$
$= 3.41$ deg F

$dT_c = (5.63$ BTU/sec$)/((2.5$ lb/sec$) \times (0.99$ BTU/lb-deg F$))$
$= 2.27$ deg F

Parallel flow efficiency: Example 11

$$\text{eff.} = \frac{1 - e^{(-3.03 \times (1 + 0.67)}}{1 + 0.67}$$

$$= 0.60$$

$$Q = 0.60 \times 1.65 \times 10.\ \text{deg F} = 4.02\ \text{BTU/sec}$$

$$dT_h = (4.02\ \text{BTU/sec})/((1.5\ \text{lb/sec}) \times (1.10\ \text{BTU/lb-deg F}))$$

$$= 2.44\ \text{deg F}$$

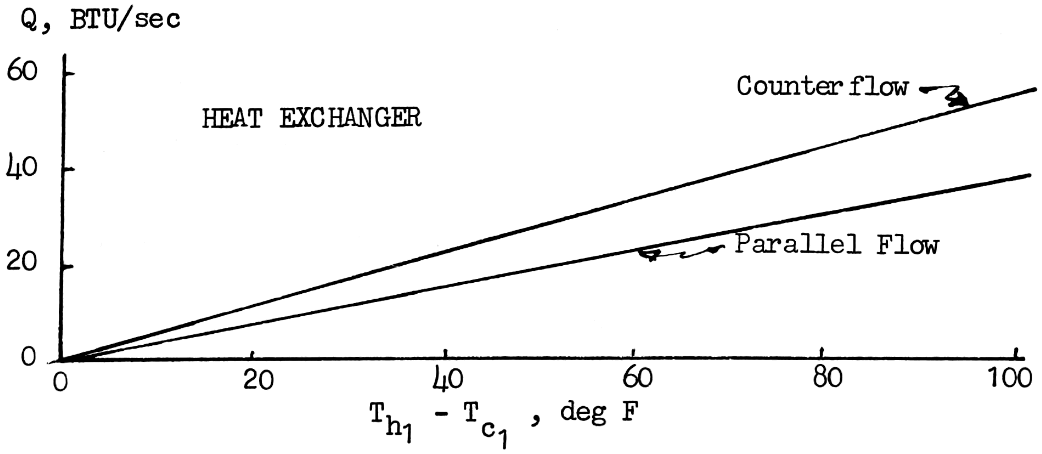
$$dT_c = (4.02\ \text{BTU/sec})/((2.5\ \text{lb/sec}) \times (0.99\ \text{BTU/lb-deg F}))$$

$$= 1.62\ \text{deg F}$$

TABULATED DATA:

$T_{h_1} - T_{c_1}$ deg F	Counterflow Q BTU/sec	Parallel Flow Q BTU/sec
10.	5.63	4.02
20.	11.26	8.04
40.	22.51	16.08
100.	56.28	40.20

Examples 10 and 11. HEAT EXCHANGER PERFORMANCE



ENERGY COSTS

The principal sources of energy for stationary uses are fuel oil, natural gas and coal. Natural gas is the preferred fuel because of a lack of residue or deposits and burning cleanliness.

Prior to the OPEC (Organization of Petroleum Exporting Countries) embargo on oil shipments to the United States in 1975, fuel oil sold for approximately 2 to 5 dollars a barrel (42 gal). Natural gas was sold under twenty year contract for as low as 10 to 20 cents per million BTU, as limited by Federal regulation.

Spring 1981 fuel oil is priced at approximately one dollar per gallon with crude petroleum approximately forty dollars per barrel. Natural gas, which is still controlled in price by Federal regulation, has been allowed to sell for approximately two dollars per million BTU, with efforts being made by procucers to raise the price to six dollars to bring it into equality on a BTU basis with fuel oil.

Natural gas is used to heat over half of all American homes. It is also used extensively for industrial applications. Wellhead petroleum usually contains a large amount of natural gas and if a pipeline is not available the gas is burned or "flared" to dispose of it. Overseas natural gas is made available by shipping as a liquid in refrigerated vessels at temperatures on the order of -200 deg F.

Crude petroleum ordinarily contains sulfur as an undesirable element. Those oils with little sulfur are called "sweet" and those with much sulfur "sour". The latter have a distinctive rotten egg smell. High sulfur compounds are restricted from burning in many localities and sell at a lower marked price.

Coal is the product of a succession of organic processes and is found in various compositions and conditions. Most primitive and least decomposed is peat or lignite. The intermediate grade, which comprises most coal, is soft, bituminous or steam coal. The cleanest and highest energy coal is hard, anthracite or metallurgical coal. It is preferred that coal by washed and dried before burning. Coal can be transported by pipeline by mixing with water. Coal can be used and burned as a fluid by grinding and mixing with air under pressure.

The state of the art of solar energy devices is indicated by a press release of January 1981 by Energy Conversion Devices Inc. They report tests of a direct conversion silicon cell with an efficiency of 6.6 percent. The Department of Energy has a goal to develop by 1986 a solar cell with a cost of $2.20 per peak watt. A peak watt is defined as one produced by a solar cell exposed to the sun at the equator at high noon normal to its rays at sea level on a clear day.

ENERGY UNIT COSTS

FUEL TYPE	UNIT COST (nominal)	HEATING VALUE BTU/lb	COST PER UNIT $/mm BTU	$/Therm
Lignite	$6.60/ton	8,300.	0.40	0.04
Coal	$32./ton	13,500.	1.20	0.12
Natural gas (LNG) (methane)	$2.00/1000 ft^3 (0.047 lb/ft^3)	21,000.	2.03	0.20
Bottle gas (LPG) (propane, butane)	$0.51/gallon (4.24 lb/gal)	20,000.	6.01	0.60
Fuel oil	$ 1.00/gallon 6.4 lb/gal)	19,000.	8.22	0.82
Wood	$ 90./cord (128 ft^3/cord) (30. lb/ft^3)	8,000.	2.93	0.29
Charcoal	$ 0.13/lb	14,000.	8.75	0.88
Electric Power	$0.015/kilowatt-hr $0.050/kilowatt-hr	(1 kw = 3413 BTU/hr)	4.39 14.65	0.44 1.47

(one Therm = 100,000 BTU)

HOME HEATING ENERGY LOSSES, Example 12

Energy analysis methods can be applied to estimate home heating energy costs and the benefits to be expected by use of insulation. A typical home has been defined for illustration:

Floor and attic area	2000. ft^2
Outer wall area	1440. ft^2
Window area (single pane)	432. ft^2
Outer door area (solid wood, 1")	65. ft^2
Chimney minimum flow area	1.5 ft^2
Chimney height	12. ft
Water heater area	39. ft^2
Heating duct area	0.6 ft^2
Duct length	100. ft

The cost of energy has been assumed at 2.00 dollars per million BTU. This includes a furnace efficiency of approximately 75 percent.

Heating duration, 3 months 7.78×10^6 sec
Heating duct flow at 10 percent of time = 1.30×10^6 sec

Inside temperature	80. deg F
Outside temperature	0. deg F
Attic temperature	0. deg F
Subfloor temperature	40. deg F

On this basis the energy costs would be as follows:

SUMMARY OF HOME HEATING ENERGY COSTS

Location	Construction	Energy Cost Uninsulated	Insulated
Ceiling	Plaster on Wood Lath 2.4" thickness	$ 2300.	$ 45. (+ 4" glass wool)
Floor	Wood, 1-1/4"	754.	23. (+ 4" glass wool)
Wall	Brick, 4" + 1/2" plaster	566.	115. (+ 1" fiberboard)
	Wood Frame, board & plaster	103.	20. (+ 4" foam)
Window	Single pane	104.	19. (double pane)
Chimney	Damper open	305.	--- (closed)
Water Heater		26.	2.
Door	Wood, solid	3.	
Heating Duct		2.	

CEILING ENERGY LOSS

UNINSULATED CEILING,
Plaster on wood lath, 3/4 inch thick

$L_1 = 0.0625$ ft

$k_1 = 5.79 \times 10^{-5}$ BTU/sec-ft-deg F

$U = k_1/L_1 = (5.79 \times 10^{-5})/0.0625 = 9.26 \times 10^{-4}$ BTU/sec-ft^2-sec

$Q = U\ A\ dT\ dt$

$= (9.26 \times 10^{-4}) \times (2000.\ \text{ft}^2) \times (80.\ \text{deg F}) \times (7.78 \times 10^6\ \text{sec})$

$= 1.15 \times 10^9$ BTU

Energy cost = $(1.15 \times 10^9$ BTU$) \times (2.00$ \$/mm BTU$)$ = \$2300.

INSULATED CEILING
4 inch glass wool batt added

$L_2 = 0.33$ ft

$k_2 = 6.11 \times 10^{-6}$ BTU/sec-ft-deg F

$1/U = L_1/k_1 + L_2/k_2$

$= 0.0625/(5.79 \times 10^{-5}) + 0.33/(6.11 \times 10^{-6}) = 55088.$

$U = 1.82 \times 10^{-5}$ BTU/sec-ft^2-deg F

$Q = U\ A\ dT\ dt$

$= (1.82 \times 10^{-5}) \times (2000.) \times (80.\ \text{deg F}) \times (7.78 \times 10^6\ \text{sec})$

$= 2.27 \times 10^7$ BTU

Energy cost = $(2.27 \times 10^7$ BTU$) \times (2.00$ \$/mm BTU$)$ = \$45.

FLOOR ENERGY LOSS

UNINSULATED FLOOR
Wood, 1-1/4" thick

$L_1 = 0.104$ ft

$k_1 = 6.39 \times 10^{-5}$ BTU/sec-ft-deg F

$U = k_1/L_1 = (6.39 \times 10^{-5})/0.104 = 6.14 \times 10^{-4}$ BTU/sec-ft^2-deg F

$Q = U\ A\ dT\ dt$

$= (6.14 \times 10^{-4}) \times (2000.\ \text{ft}^2) \times (40.\ \text{deg F}) \times (7.78 \times 10^6\ \text{sec})$

$= 3.82 \times 10^8$ BTU

Energy cost = $(3.82 \times 10^8$ BTU$) \times (2.00$ \$/mm BTU$)$ = \$ 754.

INSULATED FLOOR
4" glass wool batt added

$L_2 = 0.33$ ft

$k_2 = 6.11 \times 10^{-6}$ BTU/sec-ft-deg F

$1/U = L_1/k_1 + L_2/k_2$

$= 0.104/(6.39 \times 10^{-5}) + 0.33/(6.11 \times 10^{-6}) = 55637.$

$U = 1.80 \times 10^{-5}$ BTU/sec-ft^2-deg F

$Q = U\ A\ dT\ dt$

$= (1.80 \times 10^{-5}) \times (2000.\ \text{ft}^2) \times (40.\ \text{deg F}) \times (7.78 \times 10^6\ \text{sec})$

$= 1.12 \times 10^7$ BTU

Energy cost = $(1.12 \times 10^7$ BTU$) \times (2.00$ \$/mm BTU$)$ = \$ 23.

BRICK WALL ENERGY LOSS

UNINSULATED BRICK WALL
1.Common brick, 4" thick
2.Plaster, gypsum-sand, 1/2" thick

L_1 = 0.33 ft

$k_1 = 1.16 \times 10^{-4}$ BTU/sec-ft-deg F

L_2 = 0.0417 ft

$k_2 = 1.29 \times 10^{-4}$ BTU/sec-ft-deg F

$1/U = L_1/k_1 + L_2/k_2$

$= 0.33/(1.16 \times 10^{-4}) + 0.0417/(1.29 \times 10^{-4}) = 3167.$

$U = 3.16 \times 10^{-4}$ BTU/sec-ft^2-deg F

Q = U A dT dt

$= (3.16 \times 10^{-4}) \times (1440\ \text{ft}^2) \times (80\ \text{deg F}) \times (7.78 \times 10^6\ \text{sec})$

$= 2.83 \times 10^8$ BTU

Energy cost = $(2.83 \times 10^8) \times (2.00$ \$/mm BTU) = \$ 566.

INSULATED BRICK WALL
3. Cellulose fiberboard, 1" thick

L_3 = 0.0833 ft

$k_3 = 6.67 \times 10^{-6}$ BTU/sec-ft-deg F

$1/U = L_1/k_1 + L_2/k_2 + L_3/k_3$

$= 3167. + 0.0833/(6.67 \times 10^{-6}) = 15656.$

$U = 6.39 \times 10^{-5}$ BTU/sec-ft^2-deg F

Q = U A dT dt

$= (6.39 \times 10^{-5}) \times (1440\ \text{ft}^2) \times (80.\ \text{deg F}) \times (7.78 \times 10^6\ \text{sec})$

$= 5.73 \times 10^7$ BTU

Energy Cost = $(5.73 \times 10^7$ BTU) x (2.00 \$/mm BTU) = \$ 115.

WOOD FRAME WALL ENERGY LOSS

UNINSULATED WOOD FRAME WALL

1. Inside panel, Plaster on wood lath, 3/4" thick
   2" x 4" studs between panels
2. Outside panel, Wood siding, 2" thick

INSIDE PANEL

$L_1 = 0.0625$ ft

$k_1 = 5.79 \times 10^{-5}$ BTU/sec-ft-deg F

air film $h = 2.98 \times 10^{-4}$ BTU/sec-deg F-ft^2
(from example of "Flat Plate, vertical, turbulent")

$$1/U = 1/h + L_1/k_1 + 1/h$$

$$= 1/(2.98 \times 10^{-4}) + 0.0625/(5.79 \times 10^{-5}) + 1/(2.98 \times 10^{-4})$$

$$= 7790.$$

$$U = 1.28 \times 10^{-4} \text{ BTU/sec-ft}^2\text{-deg F}$$

OUTSIDE PANEL

$L_2 = 0.167$ ft

$k_2 = 6.39 \times 10^{-5}$ BTU/sec-ft-deg F

$$1/U = 1/h + L_2/k_2 + 1/h$$

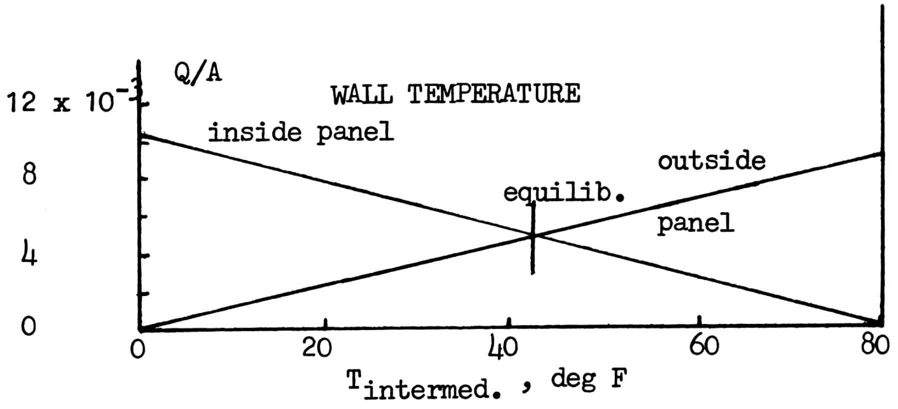
$$= 1/(2.98 \times 10^{-4}) + 0.167/(6.39 \times 10^{-5}) + 1/(2.98 \times 10^{-4})$$

$$= 9325.$$

$$U = 1.07 \times 10^{-4} \text{ BTU/sec-ft}^2\text{-deg F}$$

Heat flux, Q/A = U dT for inside panel must equal that for outside panel with inside temperature of 80. deg F and 0. deg F outside.

Intermediate Temperature deg F	Q/A Inside Panel $BTU/sec\text{-}ft^2$	Q/A Outside Panel $BTU/sec\text{-}ft^2$
0.	10.2×10^{-3}	0.
20.	7.68×10^{-3}	2.14×10^{-3}
40.	5.12×10^{-3}	4.28×10^{-3}
60.	2.56×10^{-3}	6.42×10^{-3}
80.	0.	8.56×10^{-3}



The above plot indicates an intermediate temperature of approximately 45 degrees F and a wall heat flux of 0.0046 $BTU/sec\text{-}ft^2$

$Q = (U\ dT) \times A \times dt$

$= (0.0046\ BTU/sec\text{-}ft^2) \times (1440\ ft^2) \times (7.78 \times 10^6\ sec)$

$= 5.15 \times 10^7\ BTU$

Energy cost = $(5.15 \times 10^7\ BTU) \times (2.00\ \$/mm\ BTU)$ = $ 103.

INSULATED WOOD FRAME WALL

3. Polyurethane foam between panels, 4" thick

L_3 = 0.33 ft

$k_3 = 3.94 \times 10^{-6}$ BTU/sec-ft-deg F

$$1/U = 1/h + L_1/k_1 + L_3/k_3 + L_2/k_2 + 1/h$$

$$= 1/(2.98 \times 10^{-4}) + 0.0625/(5.79 \times 10^{-5}) + 0.33/(3.94 \times 10^{-6}) + 0.167/(6.39 \times 10^{-5}) + 1/(2.98 \times 10^{-4})$$

$$= 90805.$$

$U = 1.10 \times 10^{-5}$ BTU/sec-ft^2-deg F

$$Q = U\ A\ dT\ dt$$

$$= (1.10 \times 10^{-5}) \times (1440.\ \text{ft}^2) \times (80.\ \text{deg F}) \times (7.78 \times 10^6\ \text{sec})$$

$$= 9.86 \times 10^6\ \text{BTU}$$

Energy cost = $(9.86 \times 10^6) \times (2.00$ \$/mm BTU) = \$ 20.

WINDOW ENERGY LOSS

UNINSULATED WINDOW

Single thickness glass, 1/8" thick

L_1 = 0.0104 ft

$k_1 = 1.22 \times 10^{-5}$ BTU/sec-ft-deg F

See "Free Convection, Vertical Plate"

Grashof constant, $C_4 = 3.38 \times 10^6$

Grashof constant, $C_3 = C_4 L^3 = (3.38 \times 10^6) \times (6.0 \text{ ft})^3$

$= 7.30 \times 10^8$

Grashof Number = C_3 dT = $(7.30 \times 10^8) \times 80.$ deg F

$= 5.84 \times 10^{10}$

Flow is turbulent, $h = 4.65 \times 10^{-4}$ BTU/sec-ft^2-deg F

$1/U = 1/h + L_1/k_1 + 1/h$

$= 1/(4.65 \times 10^{-4}) + 0.0104/(1.22 \times 10^{-5}) + 1/(4.65 \times 10^{-4})$

$= 5154.$

$U = 1.94 \times 10^{-4}$ BTU/sec-ft^2-deg F

Q = U A dT dt

$= (1.94 \times 10^{-4}) \times 432.$ ft^2) x (80. deg F) x (7.78×10^6 sec)

$= 5.22 \times 10^7$ BTU

Energy cost = (5.22×10^7 BTU) x (2.00 \$/mm BTU) = \$ 104.

INSULATED WINDOW

Double pane

2. Air space, 1" thick

$L_2 = 0.0833$ ft

$k_2 = 3.77 \times 10^{-6}$ BTU/sec-ft-deg F

$$1/U = 1/h + L_1/k_1 + L_2/k_2 + L_1/k_1 + 1/h$$

$$= 1/(4.65 \times 10^{-4}) + 0.0104/(1.22 \times 10^{-5}) + 0.0833/(3.77 \times 10^{-6}) + 0.104/(1.22 \times 10^{-5}) + 1/(4.65 \times 10^{-4})$$

$$= 28101.$$

$U = 3.56 \times 10^{-5}$ BTU/sec-ft^2-deg F

$$Q = U\ A\ dT\ dt$$

$$= (3.56 \times 10^{-5}) \times (432.\ \text{ft}^2) \times (80.\ \text{deg F}) \times (7.78 \times 10^6\ \text{sec})$$

$$= 9.57 \times 10^6\ \text{BTU}$$

Energy **cost** = $(9.57 \times 10^6$ BTU$) \times (2.00$ \$/mm BTU$)$ = \$ 19.

CHIMNEY HEAT LOSS, DAMPER OPEN

Specific weight of outside air

$$g\rho_{out} = P/(RT)$$

$$= (14.7 \text{ lb/in}^2 \times 144)/(53.3 \times (460. + 0.))$$

$$= 0.0863 \text{ lb/ft}^3$$

$$g\rho_{in} = (14.7 \text{ lb/in}^2 \times 144)/(53.3 \times (460 + 80.))$$

$$= 0.0735 \text{ lb/ft}^3$$

Head of air causing flow in chimney

$$H = \text{Height} \times (g\rho_{out} - g\rho_{in})/g\rho_{in}$$

$$= (12.\ \text{ft}) \times (0.0863 - 0.0735)/0.0735$$

$$= 2.09 \text{ ft}$$

Air flowrate, volume

$$dQ = c_d\, A\, (2g\, H)^{1/2}$$

$$= (0.8) \times (1.5 \text{ ft}^2) \times (2 \times 32.2 \text{ ft/sec}^2 \times 2.09 \text{ ft})^{1/2}$$

$$= 13.9 \text{ ft}^3\text{/sec}$$

Air flowrate, weight

$$dW = dQ \times g\rho_{in}$$

$$= (13.9 \text{ ft}^3\text{/sec}) \times (0.0735 \text{ lb/ft}^3)$$

$$= 1.02 \text{ lb/sec}$$

Energy loss

$$E = (dW\ dt) \times c_p \times dT$$

$$= (1.02 \text{ lb/sec}) \times (7.78 \times 10^6 \text{ sec}) \times (0.24 \text{ BTU/lb-deg F}) \times (80 \text{ deg F})$$

$$= 1.52 \times 10^8 \text{ BTU}$$

Energy cost = $(1.52 \times 10^8 \text{ BTU}) \times (2.00 \text{ \$/mm BTU})$ = \$ 305.

DOOR ENERGY LOSS

UNINSULATED OUTSIDE DOORS

Four doors, solid wood, 1" thick

$L_1 = 0.0833$ ft)

$k_1 = 6.39 \times 10^{-5}$ BTU/sec-ft-deg F

air film $h = 2.98 \times 10^{-4}$ BTU/sec-ft^2-deg F
(from example of "Flat Plate, Vertical, Turbulent")

$$1/U = 1/h + L_1/k_1 + 1/h$$

$$= 1/(2.98 \times 10^{-4}) + 0.0833/(6.39 \times 10^{-5}) + 1/(2.98 \times 10^{-4})$$

$$= 31581.$$

$U = 3.17 \times 10^{-5}$ BTU/sec-ft^2-deg F

$$Q = U\ A\ dT\ dt$$

$$= (3.17 \times 10^{-5}) \times (65.\ \text{ft}^2) \times (80.\ \text{deg F}) \times (7.78 \times 10^{6}\ \text{sec})$$

$$= 1.28 \times 10^{6}\ \text{BTU}$$

Energy cost = (1.28×10^6 BTU) x (2.00 \$/mm BTU) = \$ 3.

WATER HEATER ENERGY LOSS

UNINSULATED WATER HEATER

1. Glass liner, 1/10"
2. Steel case, 0.06"

$L_1 = 0.0083$ ft

$k_1 = 1.22 \times 10^{-5}$ BTU/sec-ft-deg F

$L_2 = 0.0050$ ft

$k_2 = 7.18\ 10^{-3}$ BTU/sec-ft-deg F

air film $h = 4.65 \times 10^{-4}$ BTU/sec-ft^2-deg F
(See example, "Flat Plate, Vertical, Turbulent")

$$1/U = 1/h + L_2/k_2 + L_1/k_1$$

$$= 1/(4.65 \times 10^{-4}) + 0.0050/(7.18 \times 10^{-3}) + 0.0083/(1.22 \times 10^{-5})$$

$$= 2832.$$

$U = 3.53 \times 10^{-4}$ BTU/sec-ft^2-deg F

$$Q = U\ A\ dT\ dt$$

$$= (3.53 \times 10^{-4}) \times 39.\ \text{ft}^2 \times 120\ \text{deg F} \times (7.78 \times 10^{6}\ \text{sec})$$

$$= 1.29 \times 10^{7}\ \text{BTU}$$

Energy cost = $(1.29 \times 10^{7}$ BTU$) \times (2.00$ \$/mm BTU$)$ = \$ 26.

INSULATED WATER HEATER

3. Glass wool batt, 2"

$L_3 = 0.167$ ft

$k_3 = 6.11 \times 10^{-6}$ BTU/sec=ft-deg F

$$1/U = 1/h + L_3/k_3 + L_2/k_2 + L_1/k_1$$

$$= 2832. + 0.167/(6.11 \times 10^{-6}) = 30164.$$

$U = 3.32 \times 10^{-5}$ BTU/sec-ft^2-deg F

$$Q = U\ A\ dT\ dt = (3.32 \times 10^{-5}) \times 39. \times 120. \times 7.78 \times 10^{6}$$

$$= 1.21 \times 10^{6}\ \text{BTU}$$

Energy cost = $(1.21 \times 10^{6}$ BTU$) \times (2.00$ \$/mm BTU$)$ = \$ 2.

HEATING DUCT ENERGY LOSS

UNINSULATED HEATING DUCT

Duct energy loss results from convective air film;
Find Grashof Constants and Grashof Number

Grashof $C_4 = 3.38 \times 10^6$ (See Example of Free Convection)

Grashof $C_3 = C_4\ L^3 = (3.38 \times 10^6) \times (0.5 \text{ ft})^3 = 4.23 \times 10^5$

Grashof Number $= C_3\ dT$

$= (4.23 \times 10^5) \times (80. \text{ deg F}) = 3.38 \times 10^7$

Flow is laminar. See "Free Convection, Vertical Plate, Laminar"

Use Program 3. air properties: $Pr = 0.72$

$L = 0.5$ ft

$k = 3.77 \times 10^{-6}$ BTU/sec-ft -deg F

$L/k = 1.33 \times 10^5$

$h = 0.0002$ BTU/sec-ft^2-deg F

$Q/A = 0.0174$ BTU/sec-ft^2 (for 80 degrees F diff.)

Energy loss $= (Q/A) \times A \times dt$

$= (0.0174 \text{ BTU/sec-ft}^2) \times (50. \text{ ft}^2) \times (1.30 \times 10^6 \text{ sec})$

$= 1.13 \times 10^6$ BTU

Energy cost $= (1.13 \times 10^6 \text{ BTU}) \times (2.00 \text{ \$/mm BTU}) = \$ 2.$

HOME COOLING ENERGY LOSSES, Example 13

Home cooling energy losses can be developed from the material generated by the study of home heating in the above examples. Only the insulated case is considered. Temperatures and time have been adjusted for the change from winter to summer conditions.

Cooling duration, 6 months	1.56×10^7 sec
Inside temperature	70. deg F
Outside temperature	95. deg F
Attic temperature	110. deg F
Subfloor temperature	90. deg F

Refrigeration power, electric
Energy rate 2.0 ¢/kwh = 6.00 $/mm BTU
Mechanical efficiency = 50. percent

On this basis the cooling costs would be as follows:

LOCATION	ENERGY COST
CEILING	$ 272.
FLOOR	134.
WOOD FRAME WALLS	75.
WINDOWS	197.
TOTAL	$678.

INSULATED CEILING COOLING ENERGY

$Q = U\ A\ dT\ dt$

$= (1.82 \times 10^{-5}\ \mathrm{BTU/sec\text{-}ft^2\text{-}deg\ F}) \times (2000.\ \mathrm{ft}^2)$

$\times (30.\ \mathrm{deg\ F}) \times (1.56 \times 10^{7}\ \mathrm{sec})$

$= 2.27 \times 10^{7}\ \mathrm{BTU}$

Energy cost = $(2.27 \times 10^{7}\ \mathrm{BTU}) \times (6.00\ \$/\mathrm{mm\ BTU})/(0.5\ \mathrm{effic.})$

= \$ 272.

INSULATED FLOOR COOLING ENERGY

$Q = U\ A\ dT\ dt$

$= (1.80 \times 10^{-5}) \times (2000.\ \mathrm{ft}^2) \times (20.\ \mathrm{deg\ F}) \times (1.56 \times 10^{7}\ \mathrm{sec})$

$= 1.12 \times 10^{7}\ \mathrm{BTU}$

Energy cost = $(1.12 \times 10^{7}\ \mathrm{BTU}) \times (6.00\ \$/\mathrm{mm\ BTU})/(0.5\ \mathrm{effic.})$

= \$ 134.

INSULATED WOOD FRAME WALLS COOLING ENERGY

$Q = U\ A\ dT\ dt$

$= (1.10 \times 10^{-5}) \times (1440.\ \mathrm{ft}^2) \times (25.\ \mathrm{deg\ F}) \times (1.56 \times 10^{7}\ \mathrm{sec})$

$= 6.22 \times 10^{6}\ \mathrm{BTU}$

Energy cost = $(6.22 \times 10^{6}\ \mathrm{BTU}) \times (6.00\ \$/\mathrm{mm\ BTU})/(0.5\ \mathrm{effic.})$

= \$ 75.

INSULATED WINDOWS COOLING ENERGY

$Q = U\ A\ dT\ dt$

$= (9.70 \times 10^{-5}) \times (432.\ \mathrm{ft}^2) \times (25.\ \mathrm{deg\ F}) \times (1.56 \times 10^{7}\ \mathrm{sec})$

$= 1.64 \times 10^{7}\ \mathrm{BTU}$

Energy cost = $(1.64 \times 10^{7}\ \mathrm{BTU}) \times (6.00\ \$/\mathrm{mm\ BTU})/(0.5\ \mathrm{effic.})$

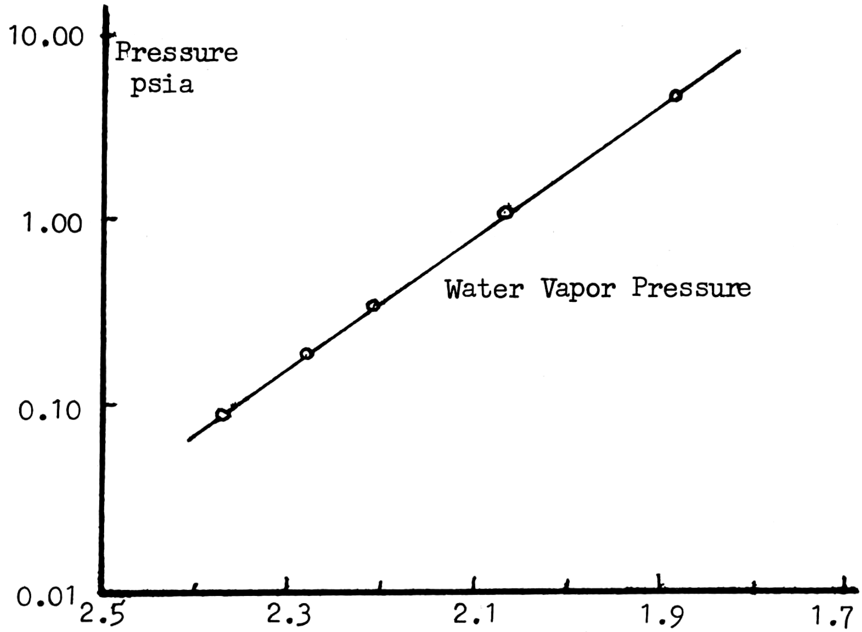
= \$ 197.

WATER VAPOR PRESSURE

Selected values for water vapor pressure as listed by the Handbook of Chemistry and Physics, 48th Edition are listed below:

Temperature deg C	Temperature deg F	Vapor Pressure psia
0	32.	0.0886
10	50.	0.1780
20	68.	0.3386
40	104.	1.0661
70	158.	4.516

These values can be plotted to form a straight line on ordinates of log pressure versus the reciprocal of the temperature ratio, based on the critical point of 1165 deg R.



The plot can be expressed by a linear equation of the form:

$$\log P = C_1 + C_2 \left(1/(T/T_c)\right)$$

The coefficients can be found by setting values for two cases and solving simultaneously:

Case 1: $\log 0.2471 = C_1 + C_2\ (2.245)$ at 59 deg F

Case 2: $\log 4.516 = C_1 + C_2\ (1.885)$ at 158 deg F

Solve Case 1 for C_1

$$C_1 = \log 0.2471 - C_2\ (2.245)$$

Insert C_1 expression in Case 2

$$\log 4.516 = \log 0.2471 - C_2\ (2.245) + C_2\ (1.885)$$

Solve for C_2

$$\begin{aligned} C_2 &= (\log 0.2471 - \log 4.516)/(2.245 - 1.885) \\ &= -3.505 \end{aligned}$$

Solve for C_1

$$C_1 = \log 0.2471 - (-3.505)\ (2.245) = 7.262$$

State final equation

$$P = \log^{-1}(7.262 - 3.505\ (1/(T/T_c)))$$

For convenience, the equation can be solved by use of Program 17 in the Appendix.

HUMIDITY

Humidity is a measure of the water vapor content in the atmosphere. The amount is controlled by the temperature and the availability of water.

Humididy is usually measured by a sling psychrometer which consists of two thermometers mounted together on an axis and rotated by hand in the air. One bulb is dry; the other is wet. The wet bulb is covered with a damp cloth. At saturation, 100 percent humidity, both bulbs give the same reading. At less than saturation, the wet bulb gives a lower reading.

The principal of the psychrometer (psychro, Gr., cold) is that the temperature is reduced by an amount equivalent of the energy needed to saturate the originally unsaturated air.

The ratio of water vapor to dry air at the wet bulb is governed by Dalton's Law of Partial Pressures: The constituents occupy the same volume in proportion to their molecular weights and partial pressures.

The total pressure of the atmosphere at sea level is 14.69 psia at sea level. The partial pressure of the atmosphere at sea level is this value minus the partial pressure of the water vapor.

DALTON'S LAW: $W = (M_v/M_a)(P_v/P_a)$

where W = Humidity Ratio, water vapor to air weight
M_v = Molecular weight of vapor, 18.0
M_a = Molecular weight of air, 28.96
P_v = Pressure of vapor
P_a = Pressure of air

The wet bulb of the psychrometer accomplishes an adiabatic, that is, a constant energy process. The total energy of the incoming air is partially converted to evaporation of additional water with a reduction in temperature.

The energy of the system is expressed by the function of enthalpy, which is a measure of energy based on temperature and pressure. We can make an energy balance, where the energy of the entering air and vapor plus the energy of the added water equals the energy of the leaving air and water.

$$H_{a1} + H_{v1} + H_w = H_{a2} + H_{v2}$$

where H_{a1} = Enthalpy of entering air, BTU/lb

H_{v1} = Enthalpy of entering vapor, BTU/lb

H_w = Enthalpy of added water, BTU/lb
H_{a2} = Enthalpy of leaving air, BTU/lb
H_{v2} = Enthalpy of leaving vapor, BTU/lb

Change of enthalpy for a gas or vapor can be found by the simple expression

$$dH = c_p \, dT$$

where dH = Change of enthalpy, BTU/lb
dT = Change of temperature, Deg F
c_p = Specific heat at constant pressure,
0.24 for air, BTU/lb-deg F
0.44 for water vapor, BTU/lb-deg F

Enthalpy change for the added water vapor is found by values for the energy change with change of state from a liquid to a vapor.

The enthalpy of saturated water can be found by the equation:

$$H_l = 925. - 9.8 \, (1166. - T)^{0.7}$$

where H_l = Enthalpy of liquid, BTU/lb
T = Temperature, deg R

The enthalpy of saturated water vapor is:

$$H_g = 1202. - 0.3 \, (860. - T)$$

where H_g = Enthalpy of vapor, BTU/lb
T = Temperature, deg R

The energy balance for the adiabatic process of the psychrometer can now be expressed as:

$$c_{pa} \, (T_1 - T_2) + c_{pv} \, (T_1 - T_2) \, W_1 = (W_2 - W_1)(H_g - H_l)$$

solving for W_1 (See Program 17)

$$W_1 = W_2 - (T_1 - T_2)(c_{pa} - W_1 \, c_{pv})/(H_g - H_l)$$

The partial pressure of the entering air water vapor can now be found:

$$P_{v1} = W_1 \, P_{a1} \, (M_a / M_v)$$

The humidity of the undisturbed air is then simply the ratio of the actual entering vapor pressure to the saturated vapor pressure.

$$\phi = P_{v1} \, / P_{1\text{-sat}}$$

EXAMPLE 14 DETERMINE HUMIDITY FROM PSYCHROMETER READINGS

DATA: Dry bulb temperature, 80. deg F
Wet bulb temperature, 64. deg F

PROCEDURE:

1. Find water vapor pressure at 64 and 80 deg F. (Use Program 17).

   $P = 0.2947$ psi at 64 deg F and 0.5015 at 80 deg F

2. Find humidity ratio of leaving air. (Program 19)

   $$W_2 = (M_v/M_a)(P_v/P_a)$$
   $$= (18.0/28.96)(0.2947/(14.69-0.2947))$$
   $$= 0.0124 \text{ lb water vapor/lb dry air}$$

3. Find enthalpy of saturated liquid water at 64 deg F (Program 20)

   $$H_1 = 925 - 9.8(1166-(460+64))^{0.7} = 20.3 \text{ BTU/lb}$$

4. Find enthalpy of saturated water vapor at 64 deg F (Program 20)

   $$H_g = 1202 - 0.3(860 - (460 + 64)) = 1101.2 \text{ BTU/lb}$$

5. Find humidity ratio at dry bulb temperature. (Use Program 18.)

   $$W_1 = 0.0088 \text{ lb vapor/lb dry air}$$

6. Find water vapor partial pressure at dry bulb condition. (Program 21)

   $$P_{v1} = W_1 P_{a1} (M_a/M_v)$$
   $$= 0.0088 \times (14.69 - P_{v1}) \times (28.96/18)$$
   $$= 0.2080 - 0.0142 P_{v1}$$
   $$P_{v1} = 0.2080/(1 + 0.0142) = 0.2051 \text{ psi}$$

7. Find relative humidity, ϕ, of undisturbed air. (Program 21)

   $$\phi = P_{v1}/P_{v1\text{-sat}}$$
   $$= 0.2051/0.5015$$
   $$= 0.409 \text{ or } 41. \text{ percent}$$

HEATING OF MOIST AIR

When moist air is heated without the addition of moisture the humidity is reduced. If it is desired to increase the moisture this can be done by passing the air through a humidifier, such as a water spray.

Calculations of the energy required for heating air are based on the weight of air to be processed. The basic requirement for heating air is usually that the volume of the space to be heated be supplied with a change of air at the rate of perhaps one-half to three times per hour. This rate may be specified by law in the case of public buildings.

The volume flow rate can be converted to weight flow rate by determining the specific weight of the air and water vapor using the perfect gas law.

PERFECT GAS LAW: $g\rho = P/(RT)$

where
$g\rho$ = Specific weight, lb/cuft
P = Pressure, lb/sqft
T = Temperature, deg R
R = Gas Constant, 85.78 for water
53.3 for air

The water content may be expressed as grains, with 7000 grains per pound.

EXAMPLE 15. HEATING OF MOIST AIR

DATA: The space to be heated is 5,000 sqft by 9 ft high.
The air is to be changed 2 times per hour.
The entering air is 60 deg F at 90 percent humidity.
The air is to be heated to 120 deg F without adding moisture.

PROCEDURE:

1. Find volume to be heated.

   Vol. = 9 x 5000 = 45,000. cuft

2. Find volume flow rate at 120 deg F.

   $dv/dt_{120} = 45{,}000. \times 2 = 90{,}000.$ cuft/hr

3. Find vapor pressure of water at 60 deg F and 120 deg F. (Use Program 17.)

   At 60 deg F, $P_v = 0.2567$ psi
   At 120 deg R, $P_v = 1.6664$ psi

4. Find specific weight of water vapor at 60 deg F per pound of air.

$$\begin{aligned} g\rho &= \phi\ P/RT \\ &= 0.90 \times (0.2567 \times 144)/(85.78 \times (460 + 60)) \\ &= 0.000746 \text{ lb/cuft} \end{aligned}$$

5. Find humidity at 120 deg F.

$$\phi_{120} = (0.90 \times 0.2567)/1.6664 = 0.138 \text{ or } 14 \text{ percent}$$

6. Find specific weight of air at 60 deg F and 120 deg F.

$$\begin{aligned} g\rho_{60} &= (14.69 - (0.90 \times 0.2567)) \times 144/(53.3 \times (460+60)) \\ &= 0.0751 \text{ lb/cuft} \end{aligned}$$

$$\begin{aligned} g\rho_{120} &= (14.69 -(0.138 \times 1.6664)) \times 144/(53.3 \times (460+120)) \\ &= 0.0674 \text{ lb/cuft} \end{aligned}$$

7. Find humidity ratio, R, of water vapor at 60 deg F.

$$R_{60} = 0.000746/0.0751 = 0.0099 \text{ lb water/lb air}$$

8. Find volume flow rate of air at 60 deg F.

$$\begin{aligned} dv/dt_{60} &= 90{,}000 \text{ cuft/hr} \times (0.0674/0.0751) \\ &= 80{,}772. \text{ cuft/hr} \end{aligned}$$

9. Find weight flow rate of air.

$$\begin{aligned} dw/dt &= 80{,}772 \text{ cuft/hr} \times (0.0751 + 0.000746) \text{lb/cuft} \\ &= 6126. \text{ lb/hr} \end{aligned}$$

10. Find change of enthalpy of dry air.

$$\begin{aligned} dH_a = w\ c_p\ dT &= 0.24 \times (120-60) \times 6126 \\ &= 88{,}214. \text{ BTU/hr} \end{aligned}$$

11. Find change of enthalpy of water vapor.

$$\begin{aligned} dH_v = w\ c_p\ dT &= 0.44 \times (120-60) \times 0.0099 \times 6126 \\ &= 1601. \text{ BTU/hr} \end{aligned}$$

12. Find total change of enthalpy (and energy for heating).

$$dh/dt = 88{,}214. + 1601. = 89{,}815. \text{ BTU/hr}$$

COOLING AND DEHUMIDIFYING MOIST AIR

When moist air is cooled the humidity increases. At the saturation point the humidity becomes 100 percent and subsequent cooling requires that the excess moisture be removed by condensation. An appreciable quantity of heat is extracted in the condensation process.

Cooling of moist air without condensation is referred to as removal of "sensible heat", the heat that can be sensed or felt. Condensation heat is known as "latent heat". The ratio of the sensible heat to the total heat (sensible plus latent) is known as the Sensible Heat Factor, SHF.

Cooling capacity is measured by the Standard Ton of Refrigeration. A Ton of cooling is equivalent to the rate of melting one ton of ice at 32 deg F in one day. The heat of melting is 144 BTU/lb. The Ton is therefore 144 BTU/lb x 2000 lb/ 24 hr = 12,000 BTU/hr.

The value of the latent heat is a function of temperature and the ratio of temperature to the critical temperature. For water, the equation becomes:

$$H_{fg} = 42(1166 - T)^{1/2}$$

where H_{fg} = Latent heat, BTU/lb
T = Temperature, deg R

EXAMPLE 16. COOLING AND DEHUMIDIFYING MOIST AIR

DATA: Air at 80 deg F and 55 percent humidity at 2000 cfm is cooled to 58 deg F and 80 percent humidity. Find the required cooling capacity in Tons.

PROCEDURE:

1. Find vapor pressure at the two temperatures. (Use Program 17)

$$P_{v-80} = 0.5015 \text{ psi}$$

$$P_{v-58} = 0.2394 \text{ psi}$$

2. Find Humidity Ratio at initial temperature.

$$R_{80} = \emptyset\,(P_v/(P_{atm} - P_v))(M_v/M_a)$$
$$= 0.55 \times 0.5015/(14.69-(0.55 \times 0.5015))(0.6215)$$
$$= 0.0119 \text{ lb/lb}$$

3. Find dry air specific weight at initial temperature

$$g\rho = P/RT = 144(14.69-(0.55 \times 0.5015))/(53.3 \times (460+80))$$
$$= 0.0721 \text{ lb/cuft}$$

4. Find dry air specific weight at final temperature.

$$g\rho = 144(14.69-(0.80 \times 0.0.2394))/(53.3 \times (460+58)$$
$$= 0.0756 \text{ lb/cuft}$$

5. Find Humidity Ratio at final temperature.

$$R_{58} = 0.80 \times 0.2394/(14.69-(0.80 \times 0.2394))(0.6215)$$
$$= 0.0082 \text{ lb/lb}$$

6. Find weight flow rate at initial temperature.

Air: $dw/dt_{80-a} = 2000 \text{ cuft/min} \times 0.0721 \text{ lb/cuft}$
$= 144.2$ lb/min

Vapor: $dw/dt_{80-v} = 0.0119 \times 144.2 = 1.72$ lb/min

7. Find enthalpy change of moist air, the sensible heat.

Air: $dH/dt_a = c_p \, dT \, (dw/dt)$

$= 0.24 \times (80-58) \; (144.2) = 761.4$ BTU/min

Vapor: $dH/dt_v = 0.44 \times (80-58) \; (1.72 = 16.6$ BTU/min

Total sensible heat = $761.4 + 16.6 = 778.0$ BTU/min
or $\times$ 60 = 46,681. BTU/hr

8. Find enthalpy change for condansing water, latent heat.

$$H_{fg} = 42(1166-(460+58))^{1/2} = 1069. \text{ BTU/lb}$$

$dH/dt_c = 1069 \text{ BTU/lb} \times 144.2 \text{ lb/min} \times (0.0119-0.0082)$
$= 570.35$ BTU/min
or $\times$ 60 = 34,221 BTU/hr

9. Find Sensible Heat Ratio, SHR.

$$SHR = 46{,}681/(46{,}681 + 34{,}221) = 0.58 \text{ or } 58 \text{ percent}$$

10. Find cooling capacity in Tons.

$$C = (46{,}681 + 34{,}221)/12{,}000$$
$$= 6.74 \text{ Tons}$$

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APPENDIX I.
CALCULATOR PROGRAMS FOR ENERGY ANALYSIS

1 Wall Component Conduction, Single Element
2 Wall Average Conduction, Multiple Element
3 Free Convection, Vertical Flat Plate, Laminar
4 Free Convection, Vertical Flat Plate, Turbulent
5 Forced Convection, Flat Plate, Laminar
6 Forced Convection, Round Tubes
7 Ambient Temperature and Compressible Temperature
8 Ambient Pressure and Compressible Pressure
9 Forced Convection, Flat Plate, High Velocity
10 Nucleate Pool Boiling
11 Peak Nucleate Pool Boiling
12 Minimum Film Boiling Wall Temperature
13 Film Boiling Heat Flux
14 Radiation Heat Gain
15 Radiation Heat Loss
16 Heat Exchanger Efficiency
17 Water Vapor Pressure
18 Humidity Ratio at Dry Bulb Temperature
19 Humidity Ratio at Wet Bulb Temperature
20 Enthalpy of Air or Water Vapor
21 Vapor Pressure at Dry Bulb Temperature and Humidity

Programs are in duplicate for Algebraic
and Reverse Polish Notation (RPN)

Algebraic logic is our conventional approach to equation solution with sets of brackets to indicate the order of execution. For example: ((3 x 7) DIVIDE 4)= , produces 3 x 7 = 21, 21 DIVIDE 4 = 5.25. This method is used by Texas Instruments in the SR-56 and the TI-57 and by Sharp in the Sharp 5100. Most other makers of calculators use this system.

RPN logic makes use of several calculation registers so that several values can be keyed in at one time to be followed by instructions which will be executed one at a time on the current visible register and the preceeding hidden register. For example: 7, ENTER, 3, X, produces 21. Then, 4, DIVIDE produces 5.25. This method is used by Hewlett-Packard in the HP-25 and subsequent models.

RPN PROGRAM 1.
WALL COMPONENT CONDUCTION, SINGLE ELEMENT

Loc.	Name	Value	Dimension
0	A_x/A	1.0	--- (Area ratio)
1	k_x	3.6×10^{-5}	BTU/sec-ft-deg F
2	L_x	0.1	feet

```
00    ---
01    RCL 1
02    RCL 0
03    X
04    RCL 2
05    DIVIDE
06    STOP           U_x , 3.6 x 10^-4 BTU/sec-ft^2-deg F
```

RPN PROGRAM 2.
WALL AVERAGE CONDUCTION, MULTIPLE ELEMENT

Loc.	Name	Value	Dimension
0	U_1	3.6×10^{-4}	BTU/sec-ft^2-deg F
1	U_2	0.40	" " " "
2	U_3	1.26×10^{-3}	" " " "
3	U_4	10.0	" " " "

```
00    ---
01    RCL 0
02    1/x
03    STO 7
04    RCL 1
05    1/x
06    STO + 7
07    RCL 2     (GOTO 13 if Loc. 2 and 3 are zero)
08    1/x
09    STO + 7
10    RCL 3     (GOTO 13 if Loc 3 is zero)
11    1/x
12    STO + 7
13    RCL 7
14    1/x
15    STOP      U, 2.79 x 10^-4 BTU/sec-ft^2-deg F
```

RPN PROGRAM 3.

FREE CONVECTION, VERTICAL FLAT PLATE, LAMINAR

Loc.	Name	Value	Dimension
0	C_1	0.508	---
1	C_2	0.952	---
2	C_3	1.77×10^7	see note
3	dT	5.	deg. F
4	P_r	7.9	---
5	L/k	10593.	ft/(BTU/sec-ft-deg F)
6	---		
7	scratch	---	

Note: $C_3 = g\beta\rho^2 L^3/\mu^2$

$G_r P_r < 1 \times 10^9$

```
00    ---
01    RCL 4
02    RCL 1
03    +
04    1
05    ENTER
06    4
07    DIVIDE
08    Y^x
09    STO 7
10    RCL 2
11    RCL 3
12    X
13    1
14    ENTER
15    4
16    DIVIDE
17    Y^x
18    RCL 7
19    DIVIDE
20    STO 7
21    RCL 4
22    1
23    ENTER
24    2
25    DIVIDE
26    Y^x
27    STO X 7
28    RCL 0
29    STO X 7
30    RCL 7       N_u, 80.29
31    RCL 5
32    DIVIDE      h, 7.6 x 10^-4
33    RCL 3         BTU/sec-ft^2
34    X                     -deg F
35    STOP        Q/A, 0.0379
                    BTU/sec-ft^2
```

RPN PROGRAM 4
FREE CONVECTION, VERTICAL FLAT PLATE, TURBULENT

Loc.	Name	Value	Dimension
0	C_1	0.23	---
1	C_2	0.33	---
2	C_4	3.38×10^6	see note
3	dT	20.	deg F
4	P_r	0.72	---
5	k	3.72×10^{-6}	BTU/sec-ft-deg F
6	scratch	---	
7	scratch	---	

Note: $C_4 = g\beta\rho^2/\mu^2$

$G_r = C_4 L^3 dT$

$G_r P_r > 1 \times 10^9$

```
00    ---                     09    X
01    RCL 0                   10    RCL 1
02    RCL 5                   11    Y^x
03    X                       12    STO X 6
04    STO 6                   13    RCL 6
05    RCL 2                   14    STOP       h, 2.98 x 10^-4
06    RCL 4                   15    RCL 3        BTU/sec-ft^2
07    X                       16    X                   -deg F
08    RCL 3                   17    STOP       Q/A, 0.0060
                                                 BTU/sec-ft^2
```

RPN PROGRAM 5
FORCED CONVECTION, FLAT PLATE, LAMINAR

Loc.	Name	Value	Dimension
0	P_r	7.9	---
1	L	1.5	ft
2	k	9.44×10^{-5}	BTU/sec-ft-deg F
3	V	4.0	ft/sec
4	ρ	1.94	slugs/ft^3 = $(lb/ft^3)/32.2$ ft/sec^2
5	μ	2.34×10^{-5}	lb-sec/ft^2
6	$dT.C_1$	100.664	deg F. ---
7	scratch	---	

```
00    ---                            16    Y^x
01    RCL 1                          17    STO X 7
02    RCL 3                          18    RCL 6
03    X                              19    FRAC
04    RCL 4                          20    STO X 7
05    X                              21    RCL 1
06    RCL 5                          22    RCL 2
07    DIVIDE                         23    DIVIDE
08    STOP     Rn, 497,435.          24    RCL 7      N_u
09    SQRT     (Must be less         25    INVERT
10    STO 7    than 500,000)         26    DIVIDE     h, 0.0587
11    RCL 0                          27    RCL 6         BTU/sec-ft^2
12    1                              28    INTEGER              -deg F
13    ENTER                          29    X
14    3                              30    STOP       Q/A, 5.87
15    DIVIDE                                             BTU/sec-ft^2
```

RPN PROGRAM 6.

FORCED CONVECTION, ROUND TUBES, LAMINAR AND TURBULENT

Loc.	Name	Value	Dimension
0	Dia.	0.0254	feet
1	C_1	25.25	---
2	μ	0.00008	$lb\text{-}sec/ft^2$
3	$C_2.C_3$	3000.023	---
4	C_4	4.36	---
5	k	3.6×10^{-5}	BTU/sec-ft-deg F
6	scratch	---	
7	P_r	1.9	---

```
00   INPUT dW    0.01 laminar      17   .
01   RCL 1       0.30 turb.        18   8
02   DIVIDE       lb/sec           19   Y^x
03   RCL 0                         20   STO 6
04   DIVIDE                        21   RCL 7
05   RCL 2                         22   .
06   DIVIDE                        23   4
07   RCL 3       R_n, 195. laminar 24   Y^x
08   IF (x<y)       5847. turb.    25   RCL 6
09   GO TO 16                      26   X
10   RCL 5                         27   RCL 3
11   RCL 4                         28   FRAC
12   X                             29   X
13   RCL 0                         30   RCL 5
14   DIVIDE     h, laminar         31   X
15   GO TO 34      0.0062          32   RCL 0
16   ROLL       BTU/sec-ft^2-deg F      DIVIDE      h, turb., 0.043
                                   33   STOP      BTU/sec-ft^2-deg F
```

RPN PROGRAM 7.

AMBIENT TEMPERATURE AND COMPRESSIBLE TEMPERATURE

Loc.	Name	Value	Dimension
0	T_{sl}	519.69	deg R.
1	Height	25.	1000 feet
2	C_1	-0.007888	---
3	C_2	460.	conversion
4	c_p/c_v	1.4	---
5	M_n	2.0	---
6	scratch	---	
7	scratch	---	

```
00   ---
01   RCL 2
02   RCL 1
03   X
04   e^x
05   RCL 0
06   X
07   STO 6
08   STOP       T_o, 426 deg. R
09   RCL 4
10   1
11   -
12   2
13   DIVIDE
14   STO 7
15   RCL 5
16   SQUARE
17   STO X 7
18   1
19   STO + 7
20   RCL 7
21   RCL 6
22   X
23   STOP       T_s, 768. deg R
24   RCL 3
25   -
26   STOP       T_s, 308. deg. F
```

RPN PROGRAM 8

AMBIENT PRESSURE AND COMPRESSIBLE PRESSURE

Loc.	Name	Value	Dimension
0	C_1	2116.	lb/ft^2
1	C_2	-0.0414	---
2	Height	20.	1000 feet
3	c_p/c_v	1.4	---
4	M_n	1.0	---
5	scratch	---	
6	scratch	---	
7	C_3	144.	conversion

00	---	
01	RCL 1	
02	RCL 2	
03	X	
04	e^x	
05	RCL 0	
06	X	P_o, 924. lb/ft^2
07	STO 5	
08	RCL 3	
09	1	
10	-	
11	2	
12	DIVIDE	
13	RCL 4	
14	SQUARE	
15	X	
16	1	
17	+	
18	STO 6	
19	RCL 3	
20	1	
21	-	
22	RCL 3	
23	INVERT	
24	DIVIDE	
25	RCL 6	
26	INVERT	
27	Y^x	
28	RCL 5	
29	X	
30	STOP	P_s, 1750. lb/ft^2
31	RCL 7	
32	DIVIDE	
33	STOP	P_x, 12.15 lb/in^2

RPN PROGRAM 9, Part 1 of 4
FORCED CONVECTION, FLAT PLATE, HIGH VELOCITY

Loc.	Name	Value	Dimension
0	T_0	400.	deg R
1	C_1	0.207	---
2	M_n	1.5	---
3	g	32.2	ft/sec^2
4	c_p	0.241	BTU/lb-deg F
5	μ	3.97×10^{-7}	lb-sec/ft^2
6	k	6.94×10^{-6}	BTU/sec-ft-deg F
7	scratch	---	

```
00    ---
01    RCL 2
02    SQUARE
03    RCL 1
04    X
05    STO 7
06    1
07    +
08    RCL 0
09    X
10    STOP        Ts , 586. deg R
11    RCL 3
12    RCL 4
13    X
14    RCL 5
15    X
16    RCL 6
17    DIVIDE
18    STOP        Pr, 0.44
19    SQRT
20    STOP         r, 0.666
21    RCL 7
22    X
23    1
24    +
25    RCL 0
26    X
27    STOP        Taw , 524. deg R
```

RPN PROGRAM 9, Part 2 of 4

Loc.	Name	Value	Dimension
0	T_0	400.	deg R
1	T_{aw}	524.	deg R
2	T_{wall}	450.	deg R
3	C_1	0.22	---
4	C_2	2402.	kgR = 1.4 x 32.2 x (1544/53.3)
7	scratch	---	

```
00    ---
01    RCL 1
02    RCL 0
03    -
04    RCL 3
05    X
06    STO 7
07    RCL 2
08    RCL 0
09    -
10    2
11    DIVIDE
12    RCL 7
13    +
14    RCL 0
15    +
16    STOP        Tbl , 452. deg R
17    RCL 4
18    RCL 0
19    X
20    SQRT
21    STOP        Vc , 980. ft/sec
```

RPN PROGRAM 9, Part 3 of 4

FORCED CONVECTION, FLAT PLATE, HIGH VELOCITY

Loc.	Name	Value	Dimension
0	L	10.	ft
1	M	1.5	---
2	V^n	980.	ft/sec
3	ρ^c	7.37×10^{-4}	slugs/ft^3 = (lb/ft^3)/32.2
4	μ	3.97×10^{-7}	lb-sec/ft^2

```
00   ---                      06   RCL 3
01   RCL 0                    07   X
02   RCL 1                    08   RCL 4
03   X                        09   DIVIDE
04   RCL 2                    10   STOP          R_n , 2.729 x 10^7
05   X
```

RPN PROGRAM 9, Part 4 of 4

Loc.	Name	Value	Dimension
0	P_r	0.44	---
1	R_n	2.73×10^7	---
2	C_1	0.029 (turbulent) 0.332 (laminar)	---
3	C_2	0.80 (turbulent) 0.50 (laminar)	---
4	C_3	1.44×10^6	(L/k) = 10./(6.94 x 10^{-6})
5	T_{aw}	524.	deg R
6	T_{wall}	523.	deg R
7	scratch	---	

```
00   ---                      13   RCL 2
01   RCL 0                    14   X
02   1                        15   RCL 4
03   ENTER                    16   DIVIDE        h, 0.0136
04   3                        17   RCL 5           BTU/sec-ft^2
05   DIVIDE                   18   RCL 6                    -deg F
06   y^x                      19   -
07   STO 7                    20   X
08   RCL 1                    21   STOP          Q/A, 0.0136
09   RCL 3
10   y^x
11   RCL 7
12   X
```

RPN PROGRAM 10
NUCLEATE POOL BOILING

Loc.	Name	Value	Dimension
0	$T_{wall}-T_{sat}$	20.	deg F
1	h_{fg}	970.	BTU/lb
2	P_r	1.9	---
3	c_p	1.0	BTU/lb-deg F
4	c_{sf}	0.013	---
5	g_{liq}	2.05×10^{-4}	lb/sec-ft
6	C_1	124.	(see note)

Note: $C_1 = ((g/g_0)\ ((g\rho_{liq} - g\rho_{vap})/\sigma))^{1/2}$

```
00   ---                    11   7
01   RCL 3                  12   y^x
02   RCL 0                  13   DIVIDE
03   X                      14   3
04   RCL 1                  15   y^x
05   DIVIDE                 16   RCL 5
06   RCL 4                  17   X
07   DIVIDE                 18   RCL 1
08   RCL 2                  19   X
09   1                      20   RCL 6
10   .                      21   X
                            22   STOP       Q/A,
```

3.726 BTU/sec-ft^2

RPN PROGRAM 11

PEAK NUCLEATE HEAT FLUX

Loc.	Name	Value	Dimension
0	C_1	26.87	--- (see note)
1	a/g	1.0	---
2	P_r	1.88	---
3	σ	0.004	lb/ft
4	$g\ g_0$	1036.8	$(ft/sec^2)^2$
5	C_2	62.33	lb/ft^2 (see note)
6	scratch	---	

Note: $C_1 = 0.144\ h_{fg}\ (g\rho_{vap})^{1/2}$

$C_2 = (g\rho_{liq} - g\rho_{vap})^2/g_{liq}$

```
00   ---
01   RCL 5
02   RCL 4
03   X
04   RCL 3
05   X
06   SQRT
07   SQRT
08   STO 6
09   RCL 2
10   .
11   2
12   4
13   5
14   CHG SGN
15   y^x
16   STO X 6
17   RCL 1
18   .
19   2
20   5
21   y^x
22   STO X 6
23   RCL 0
24   STO X 6
25   RCL 6
26   STOP       Q/A,
```

92.3 BTU/sec-ft^2

RPN PROGRAM 12

MINIMUM FILM BOILING WALL TEMPERATURE

Loc.	Name	Value	Dimension
0	Sp.Gr.	2.71	Specific Gravity of wall
1	A	26.9	Atomic weight of wall
2	σ_{lv}	0.004	Liquid surface tension, lb/ft
3	T_{crit}	1164.	Liquid, deg R
4	C_1	0.444	---
5	C_2	0.99	---
6	scratch	---	

```
00   ---
01   RCL 0
02   RCL 1
03   DIVIDE
04   RCL 4
05   y^x
06   STO 6
07   RCL 5
08   STO X 6
09   RCL 2
10   3
11   1/x
12   CHG SGN
13   y^x
14   STO X 6
15   RCL 6
16   STOP        m, 2.25
17   CHG SGN
18   e^x
19   CHG SGN
20   1
21   +
22   RCL 3
23   INVERT x,y
24   y^x
25   1
26   .
27   5
28   X
29   STOP        T_Leid., 830. deg R
                          (370. deg F)
```

RPN PROGRAM 13, Part 1 of 2
FILM BOILING

Loc.	Name	Value	Dimension
0	P_r	0.94	---,vapor
1	h_{fg}	700.	BTU/lb
2	c_p	0.47	vapor, BTU/lb-deg F
3	dT	200.	deg F
4	μ_{vap}	3.9×10^{-7}	$lb\text{-}sec/ft^2$
5	D	0.10	ft
6	ρ_{liq}	1.86	$(lb/ft^3/(32.2\ ft/sec^2)$
7	ρ_{vap}	8.01×10^{-4}	" "

```
00   ---
01   RCL 1
02   RCL 2
03   DIVIDE
04   RCL 3
05   DIVIDE
06   STOP          θ , 7.45
07   RCL 5
08   RCL 4
09   DIVIDE
10   SQUARE
11   RCL 5
12   X
13   RCL 6
14   RCL 7
15   -
16   X
17   RCL 7
18   X
19   3
20   2
21   .
22   2
23   X
24   STOP          Gr , 3.15 x 10^8
25   RCL 0
26   X
27   STOP          Ra , 2.96 x 10^8
```

RPN PROGRAM 13, Part 2 of 2

Loc.	Name	Value	Dimension
0		7.45	---
1	R_a	2.96×10^8	---
2	D	0.10	feet
3	k_{vap}	9.0×10^{-4}	BTU/sec-ft-deg F
4	C_1	0.94	---
5	T_c	1165.	deg R
6	T_{sat}	672.	deg R
7	T_{wall}	860.	deg R

```
00   ---
01   RCL 0
02   RCL 6
03   RCL 5
04   DIVIDE
05   SQUARE
06   DIVIDE
07   RCL 1
08   X
09   SQRT
10   SQRT
11   RCL 4
12   X
13   STOP          Nu , 268.
14   RCL 3
15   X
16   RCL 2
17   DIVIDE
18   STOP          h, 2.41
                     BTU/sec-ft^2
                           -deg F
19   RCL 7
20   RCL 6
21   -
22   X
23   STOP          Q/A , 453 BTU/sec-ft^2
```

RPN PROGRAM 14
RADIATION HEAT GAIN

Loc.	Name	Value	Dimension
0	T_{source}	5000.	deg R
1	T_{body}	500.	deg R
2	S_t-B_z const.	4.758 x 10^{-13}	BTU/ft^2-sec-deg R^4
3	Area$_{object}$	10.	ft^2
4	Relative rad.	2.5 x 10^{-3}	---, $(r_1/r_2)^2$
5	e_s x a_b	0.9	---
6	scratch	---	

```
00   ---                      10   RCL 2
01   RCL 1                    11   X
02   4                        12   RCL 3
03   y^x                      13   X
04   STO 6                    14   RCL 4
05   RCL 0                    15   X
06   4                        16   RCL 5
07   y^x                      17   X
08   RCL 6                    18   STOP      Q, 6.69 BTU/sec
09   -
```

RPN PROGRAM 15
RADIATION HEAT LOSS

Loc.	Name	Value	Dimension
0	T_{body}	500.	deg R
1	$T_{background}$	460.	deg R
2	S_t-B_z const.	4.758 x 10^{-13}	BTU/sec-ft^2-deg R^4
3	Area$_{object}$	100.	ft^2
4	---		
5	e	0.8	---
6	scratch	---	

```
00   ---                      09   -
01   RCL 1                    10   RCL 2
02   4                        11   X
03   y^x                      12   RCL 3
04   STO 6                    13   X
05   RCL 0                    14   RCL 5
06   4                        15   X
07   y^x                      16   STOP      Q, 0.67 BTU/sec
08   RCL 6
```

RPN PROGRAM 16, Part 1 of 3

HEAT EXCHANGER EFFICIENCY

Loc.	Name	Value	Dimension
0	dW_c	2.5	lb/sec
1	dW_h	1.5	lb/sec
2	c_{p-c}	0.99	cold, BTU/lb-deg F
3	c_{p-h}	1.10	hot, BTU/lb-deg F
4	A	20.0	ft^2
5	U	0.25	BTU/sec-ft^2-deg F
6	scratch	---	
7	scratch	---	

```
00   ---
01   RCL 0
02   RCL 2
03   X
04   STOP          c_c , (2.48)
05   STO 6
06   RCL 1
07   RCL 3
08   X
09   STO 7
10   STOP          c_h , (1.65) = x
11   RCL 6                      = y
12   IF(x less than y)
13   GO TO 16
14   DIVIDE
15   GO TO 19
16   STO 7
17   INVERT, x,y
18   DIVIDE
19   STO 6
20   STOP          C_min/C_max
21   RCL 4           (0.67)
22   RCL 5
23   X
24   RCL 7         C_min
25   DIVIDE
26   STO 7
27   STOP          N_tu (3.03)
```

RPN PROGRAM 16, Part 2 of 3
COUNTERFLOW HEAT EXCHANGER EFFICIENCY

Loc.	Name	Value	Dimension
6	C_{min}/C_{max}	0.67	---
7	N_{tu}	3.03	---

```
28   1
29   RCL 6
30   -
31   RCL 7
32   CHG SGN
33   X
34   e^x
35   STO 7
36   RCL 6
37   X
38   1
39   INVERT x,y
40   -
41   STO 6
42   RCL 7
43   1
44   INVERT x,y
45   -
46   RCL 6
47   DIVIDE
48   STOP                eff. , 0.84
```

RPN PROGRAM 16, Part 3 of 3
PARALLEL FLOW HEAT EXCHANGER EFFICIENCY

Loc.	Name	Value	Dimension
6	C_{min}/C_{max}	0.67	---
7	N_{tu}	3.03	---

```
28   1
29   RCL 6
30   +
31   RCL 7
32   X
33   CHG SGN
34   e^x
35   1
36   INVERT x,y
37   -
38   STO 7
39   RCL 6
40   1
41   +
42   RCL 7
43   INVERT x.y
44   DIVIDE
45   STOP                eff., 0.60
```

RPN PROGRAM 17

WATER VAPOR PRESSURE

Loc.	Name	Value	Dimension
0	Temp. Const.	460	Deg. R.
1	C_1	7.262	---
2	C_2	-3.505	---
3	Crit. Temp.	1165.	Deg R.
4	Scratch	---	---

$$P = \log^{-1} (C_1 + C_2 (1/(T/T_c)))$$

```
00   Enter Value, 32. deg F              06   RCL 2
01   RCL 0                               07   X
02   +                                   08   RCL 1
03   RCL 3                               09   +
04   DIVIDE                              10   10^x
05   1/X                                 11   STOP    P = 0.0917 psi
```

RPN PROGRAM 18

HUMIDITY RATIO AT DRY BULB TEMPERATURE

Loc.	Name	Value	Dimension
0	W_2 ratio	0.0124	Lb water/lb dry air
1	T_1 dry bulb	80.	Deg F
2	T_2 wet bulb	64.	Deg. F
3	c_{pa}	0.24	BTU/lb-deg F, specific heat
4	c_{pv}	0.44	" " "
5	H_g	1101.	BTU/lb, heat of evaporiz.
6	H_1	20.3	BTU/lb, heat of sat. liquid

$$W_1 = W_2 - ((T_1 - T_2)(c_{pa} + c_{pv}))/(H_g - H_1)$$

```
00   ---                          09   X
01   RCL 0                        10   RCL 5
02   RCL 4                        11   RCL 6
03   X                            12   -
04   RCL 3                        13   DIVIDE
05   +                            14   CHG SGN
06   RCL 1                        15   RCL 0
07   RCL 2                        16   +
08   -                            17   STOP      W1 = 0.0088 lb/lb
```

RPN PROGRAM 19
HUMIDITY RATIO AT WET BULB TEMPERATURE

Loc.	Name	Value	Dimension
0	M_v/M_a	0.6215	---
1	P_v	0.2947	psi, vapor pressure
2	P_{atm}	14.69	Atmosp. Press, psia

$$W_2 = (M_v/M_a)\ P_v/(P_{atm} - P_v)$$

```
00   ---                      05   INVERT
01   RCL 2                    06   DIVIDE
02   RCL 1                    07   RCL 0
03   -                        08   X
04   RCL 1                    09   STOP      W2 = 0.0127 lb/lb
```

RPN PROGRAM 20
ENTHALPY OF AIR OR WATER VAPOR

Loc.	Name	Value		Dimension
		Air	Water	
0	C_1	1202.	925.	
1	C_2	0.3	9.8	
2	C_3	860.	1166.	
3	C_4	460	460.	Reference, deg R
4	n	1.0	0.7	

$$H = C_1 - C_2\ (C_3 - (460 + T))^n$$

```
00   Enter T, 64 deg F        07   y^x
01   RCL 3                    08   RCL 1
02   +                        09   X
03   CHG SGN                  10   CHG SGN
04   RCL 2                    11   RCL 0
05   +                        12   +
06   RCL 4                    13   STOP    H= 20.31 water
                                          H= 1101.2 air
                                             BTU/lb
```

RPN PROGRAM 21

VAPOR PRESSURE AT DRY BULB TEMPERATURE AND RELATIVE HUMIDITY

Loc.	Name	Value	Dimension
0	W_1	0.0088	Humidity ratio, lb/lb
1	P_{atm}	14.69	psia
2	M_a/M_v	1.6088	
3	$P_{v\text{-}sat}$	0.5015	pai

$$P_{v1} = W_1 \, P_{atm} \, (M_a/M_v)/(1 + (M_a/M_v) \, W_1)$$

$$\phi = P_{v1}/P_{v\text{-}sat}$$

```
00   ---
01   RCL 0
02   RCL 2
03   X
04   1
05   +
06   RCL 2
07   INVERT
08   DIVIDE
09   RCL 1
10   X
11   RCL0
12   X
13   STOP       Pv1 = 0.2051 psi
14   RCL 3
15   DIVIDE
16   STOP       Ø = 0.409 or 41 %
```

ALGEBRAIC PROGRAM NUMBER 1
WALL COMPONENT CONDUCTION, SINGLE ELEMENT

Loc.	Name	Value	Dimension
0	A_x/A	1.0	--- (Area ratio)
1	k	3.6×10^{-5}	BTU/sec-ft-deg F
2	L_x	0.1	feet

```
LRN                                05   DIVIDE
00    ---                          06   RCL 2
01    RCL 0                        07   EQUAL
02    X                            08   R/S        U , 3.6 x 10^-4
03    RCL 1                             LRN         x   BTU/sec-ft^2
04    EQUAL                                                   -deg F
```

ALGEBRAIC PROGRAM NUMBER 2
WALL COMPONENT CONDUCTION, MULTIPLE ELEMENT

Loc.	Name	Value	Dimension
0	U_1	3.6×10^{-4}	BTU/sec-ft^2-deg F
1	U_2	0.40	"
2	U_3	1.26×10^{-3}	"
3	U_4	10.0	"
7	scratch	---	

```
LRN                                08   SBR 1
00    ---                          09   RCL 3
01    0                            10   SBR 1
02    STO 7                        11   RCL 7
03    RCL 0                        12   1/x
04    SBR 1                        13   R/S         U, 2.79 x 10^-4
05    RCL 1                        14   2nd Lbl 1      BTU/sec-ft^2
06    SBR 1                        15   1/x                   -deg F
07    RCL 2                        16   SUM 7
                                   17   INV SBR
                                        LRN
```

ALGEBRAIC PROGRAM 3
FREE CONVECTION, VERTICAL FLAT PLATE, LAMINAR

Loc.	Name	Value	Dimension
0	$dT.C_1$	5.508	deg F.--- (double storage)
1	C_2	0.952	---
2	C_3	1.77×10^7	See note
3	P_r	7.9	---
4	L/k	10593.	ft/(BTU/sec-ft-deg F)
6	scratch	---	

Note: $C_3 = g \beta L^3 \rho^2/\mu^2$

```
LRN
00   ---
01   RCL 3
02   +
03   RCL 1
04   EQUAL
05   y^x
06   .
07   2
08   5
09   EQUAL
10   STO 6
11   RCL 0
12   2nd Int
13   X
14   RCL 2
15   EQUAL          Gr , 8.85 x 10^7
16   y^x
17   .
18   2
19   5
20   EQUAL
21   DIVIDE
22   RCL 6
23   EQUAL
24   STO 6
25   RCL 3
26   y^x
27   .
28   5
29   EQUAL
30   2nd  Prd 6
31   RCL 0
32   Inv 2nd Int
33   2nd Prd 6
34   RCL 6          Nu
35   DIVIDE
36   RCL 4
37   EQUAL          h, 0.00758
38   R/S               BTU/sec-ft^2
39   STO 6             -deg F
40   RCL 0
41   2nd Int
42   X
43   RCL 6
44   EQUAL
45   R/S            Q/A, 0.0379
     LRN                 BTU/sec-ft^2
```

$G_r\ P_r = 6.99 \times 10^8$

ALGEBRAIC PROGRAM 4

FREE CONVECTION, VERTICAL FLAT PLATE, TURBULENT

Loc.	Name	Value	Dimension
0	C_1	0.23	---
1	C_2	0.33	---
2	C_4	3.38×10^6	see note
3	dT	20.	deg F
4	P_r	0.72	---
5	k	3.72×10^{-6}	BTU/sec-ft-deg F
6	scratch	---	---
7	L	4.0	ft

Note: $C_4 = g\beta\rho^2/\mu^2$ $G_r = C_4 L^3 dT$

$G_r P_r$ must exceed 1×10^9

```
LRN
00    ---
01    (
02    RCL 0
03    X
04    RCL 5
05    )
06    STO 6
07    (
08    RCL 2
09    X
10    RCL 4
11    X
12    RCL 3
13    )
14    y^x
15    RCL 1
16    EQUAL
17    2nd Prd 6
18    RCL 6
19    R/S              h, 2.95 x 10^-4
                          BTU/sec-ft^2
                                 -deg F
20    (
21    RCL 7
22    y^x
23    3
24    EQUAL
25    X
26    RCL 2
27    X
28    RCL 3
29    X
30    RCL 4
31    )
32    R/S              Gr Pr, 3.1 x 10^9
      LRN
```

ALGEBRAIC PROGRAM 5
FORCED CONVECTION, FLAT PLATE, LAMINAR

Loc.	Name	Value	Dimension
0	P_r	7.9	---
1	L	1.5	ft
2	k	9.44×10^{-5}	BTU/sec-ft-deg F
3	V	4.0	ft/sec
4	ρ	1.94	$(\text{lb/ft}^3)/(32.2\ \text{ft/sec}^2)$
5	μ	2.34×10^{-5}	lb-sec/ft^2
6	$dT.C_1$	100.664	deg F.---
7	scratch	---	

```
LRN                                    18   2nd Prd 7
00   ---                               19   RCL 6
01   (                                 20   INV 2nd Int
02   RCL 1                             21   2nd Prd 7
03   X                                 22   (
04   RCL 3                             23   RCL 7              N_u
05   X                                 24   X
06   RCL 4                             25   RCL 2
07   DIVIDE                            26   DIVIDE
08   RCL 5                             27   RCL 1
09   )                                 28   )
10   R/S       R_n , 497,435.          29   R/S              h, 0.6587
11   SQRT                              30   X                   BTU/sec
12   STO 7                             31   RCL 6             -ft^2-deg F
13   RCL 0                             32   2nd Int
14   y^x                               33   EQUAL
15   3                                 34   R/S             Q/A, 5.869
16   1/x                                    LRN                 BTU/sec
17   EQUAL                                                       -ft^2
```

ALGEBRAIC PROGRAM 6

FORCED CONVECTION, ROUND TUBES, LAMINAR AND TURBULENT

Loc.	Name	Value	Dimension
0	Dia.	0.0254	feet
1	C_1	25.29	---
2	$C_4.C_2$	3000.023	---
3	μ	0.00008	$lb-sec/ft^2$
4	C_3	4.36	---
5	k	3.6×10^{-5}	BTU/sec-ft-deg F
6	P_r	1.9	---
7	scratch, t	----	

```
LRN                                          25   GO TO 2
00   Enter dW    0.01 Laminar                26   2nd Lbl 1
01   DIVIDE      0.30 Turb., lb/sec               Y^x
02   (                                       28   .
03   RCL 1                                   29   8
04   X                                       30   EQUAL
05   RCL 0                                   31   STO 7
06   X                                       32   RCL 6
07   RCL 3                                   33   Y^x
08   )                                       34   .
09   EQUAL                                   35   4
10   R/S        Rn, 195. laminar             36   EQUAL
11   STO 7          5838. turb.              37   2nd Prd 7
12   RCL 2                                   38   RCL 2
13   2nd Int                                 39   INV 2nd Int
14   x⇆t                                     40   2nd Prd 7
15   2nd x>t                                 41   (
16   SBR 1                                   42   RCL 7
17   (                                       43   X
18   RCL 4                                   44   RCL 5
19   X                                       45   DIVIDE
20   RCL 5                                   46   RCL 0
21   DIVIDE                                  47   )
22   RCL 0                                   48   R/S          h, turb., 0.0434
23   )                                            LRN       BTU/sec-ft^2-deg F
24   R/S       h, laminar, 0.0062
               BTU/sec-ft^2-deg F
```

ALGEBRAIC PROGRAM 7

AMBIENT TEMPERATURE AND COMPRESSIBLE TEMPERATURE

Loc.	Name	Value	Dimension
0	T_{sl}	519.69	deg R (temp. @ sea level)
1	Height	25.	1000 feet
2	C_1	-0.007888	---
3	C_2	460.	conversion factor
4	c_p/c_v	1.4	Ratio of specific heats
5	M_n	2.0	---
6	scratch	---	
7	scratch	---	

```
LRN
00    ---
01    RCL 2
02    X
03    RCL 1
04    EQUAL
05    INV ln x
06    X
07    RCL 0
08    EQUAL
09    STO 6
10    R/S           T0, 426. deg R
11    RCL 4
12    -
13    1
14    EQUAL
15    DIVIDE
16    2
17    EQUAL
18    STO 7
19    RCL 5
20    SQUARE
21    2nd Prd 7
22    1
23    SUM 7
24    RCL 7
25    X
26    RCL 6
27    EQUAL
28    R/S           Ts , 768. deg R
29    -
30    RCL 3
31    EQUAL
32    R/S           Ts , 308. deg R
      LRN
```

ALGEBRAIC PROGRAM 8

AMBIENT PRESSURE AND COMPRESSIBLE PRESSURE

Loc.	Name	Value	Dimension
0	C_1	2116.	lb/ft^2 (press. @ sea level)
1	C_2	-0.0414	---
2	Height	20.	1000. ft
3	c_p/c_v	1.4	---
4	M_n	1.0	---
5	scratch	---	
6	scratch	---	
7	scratch	---	

```
LRN
00   ---
01   RCL 1
02   X
03   RCL 2
04   EQUAL
05   INV ln x
06   X
07   RCL 0
08   EQUAL          P0, 924. lb/ft2
09   STO 5
10   RCL 3
11   -
12   1
13   EQUAL
14   DIVIDE
15   2
16   EQUAL
17   STO 6
18   RCL 4
19   SQUARE
20   X
21   RCL 6
22   EQUAL
23   +
24   1
25   EQUAL
26   STO 6
27   RCL 3
28   -
29   1
30   EQUAL
31   DIVIDE
32   RCL 3
33   EQUAL
34   1/x
35   STO 7
36   RCL 6
37   y^x
38   RCL 7
39   EQUAL
40   X
41   RCL 5
42   EQUAL
43   R/S            Ps , 1750. lb/ft2
44   DIVIDE
45   1
46   4
47   4
48   EQUAL
49   R/S            Ps , 12.15 lb/in2
     LRN
```

ALGEBRAIC PROGRAM 9, Part 1 of 4

FORCED CONVECTION, FLAT PLATE, HIGH VELOCITY

Loc.	Name	Value	Dimension
0	T_0	400.	deg R
1	C_1	0.207	$(\gamma -1)/2$
2	M	1.5	---
3	g_n	32.2	ft/sec^2
4	c_p	0.241	BTU/lb-deg F
5	μ	3.97×10^{-7}	lb-sec/ft^2
6	k	6.94×10^{-6}	BTU/sec-ft-deg F
7	scratch	---	

```
LRN
00   ---
01   RCL 0
02   X
03   (
04   RCL 2
05   SQUARE
06   X
07   RCL 1
08   +
09   1
10   )
11   EQUAL
12   R/S          Ts, 586. deg R
13   (
14   RCL 3
15   X
16   RCL 4
17   X
18   RCL 5
19   DIVIDE
20   RCL 6
21   )
22   R/S          Pr, 0.442
23   SQRT
24   R/S          r, 0.664
25   X
26   RCL 2
27   SQUARE
28   EQUAL
29   X
30   RCL 1
31   EQUAL
32   +
33   1
34   EQUAL
35   X
36   RCL 0
37   EQUAL
38   R/S          Taw , 523. deg R
     LRN
```

ALGEBRAIC PROGRAM 9, Part 2 of 4

FORCED CONVECTION, FLAT PLATE, HIGH VELOCITY

Loc.	Name	Value	Dimension
0	T_0	400.	deg R
1	T_{aw}	523.	deg R
2	T_{wall}	450.	deg R
3	C_1	0.22	---
4	C_2	2402.	γ g R = 1.4 x 32.2 x 53.3
5	scratch	---	

```
LRN
00   ---
01   (
02   RCL 3
03   X
04   (
05   RCL 1
06   -
07   RCL 0
08   )
09   )
10   STO 5
11   (
12   .
13   5
14   X
15   (
16   RCL 2
17   -
18   RCL 0
19   )
20   )
21   SUM 5
22   RCL 0
23   SUM 5
24   RCL 5
25   R/S         Tbl, 452. degR
26   (
27   RCL 4
28   X
29   RCL 0
30   )
31   SQRT
32   R/S         Vc , 980. ft/sec
     LRN
```

ALGEBRAIC PROGRAM 9, Part 3 of 4

Loc.	Name	Value	Dimension
0	L	10.0	ft
1	M	1.5	---
2	V_c^n	980.	ft/sec
3	ρ	7.37×10^{-4}	$(lb/ft^3)/(32.2\ ft/sec^2)$
4	μ	3.97×10^{-7}	$lb\text{-}sec/ft^2$

```
LRN
00   ---
01   (
02   RCL 0
03   X
04   RCL 1
05   X
06   RCL 2
07   X
08   RCL 3
09   DIVIDE
10   RCL 4
11   )
12   R/S         Rn , 2.729 x 10^7
     LRN
```

ALGEBRAIC PROGRAM 9, Part 4 of 4
FORCED CONVECTION, FLAT PLATE, HIGH VELOCITY

Loc.	Name	Value	Dimension
0	P_r	0.44	---
1	R_n	3.00×10^5 laminar 2.73×10^7 turbul.	(example) "
2	L/k	1.44×10^6	10. ft/(6.94 x 10^{-6})
3	T_{aw}	524.	deg R.
4	T_{wall}	523.	deg R.
5	scratch	---	
6	C_1	0.332 laminar 0.029 turbulent	---

```
LRN
00   ---
01   RCL 0
02   y^x
03   3
04   1/x
05   EQUAL
06   STO 5
07   RCL 1
08   y^x
09   .
10   5 (laminar), 8 (turbulent)
11   EQUAL
12   2nd Prd 5
13   RCL 6
14   2nd Prd 5
15   RCL 5
16   DIVIDE
17   RCL 2
18   EQUAL
19   R/S    h, 9.6 x 10^-5 laminar
20   X         0.0136 turbulent
21   (         BTU/sec-ft^2-deg F
22   RCL 3
23   -
24   RCL 4
25   )
26   EQUAL
27   R/S   Q/A, 9.6 x 10^-5 laminar
     LRN          0.0136 turbulent
                  BTU/sec-ft^2
```

ALGEBRAIC PROGRAM 10
NUCLEATE POOL BOILING

Loc.	Name	Value	Dimension
0	$T_{wall}-T_{sat}$	20.	deg F
1	h_{fg}	970.	BTU/lb
2	P_r	1.9	---
3	c_p	1.0	BTU/lb-deg F
4	c_{sg}	0.013	---
5	$g\mu$	2.05 x 10^{-4}	lb/sec-ft
6	C_1	124.	see note
7	scratch	---	

Note: $C_1 = ((g/g_0)\ ((g\rho_{liq} - g\rho_{vap})/\sigma))^{1/2}$

```
LRN
00   ---
01   (
02   RCL 3
03   X
04   RCL 0
05   )
06   STO 7
07   (
08   RCL 2
09   y^x
10   1
11   .
12   7
13   EQUAL
14   X
15   RCL 1
16   X
17   RCL 4
18   )
19   1/x
20   2nd Prd 7
21   RCL 7
22   y^x
23   3
24   EQUAL
25   X
26   (
27   RCL 5
28   X
29   RCL 1
30   X
31   RCL 6
32   )
33   EQUAL
34   R/S          Q/A, 3.726 BTU/
     LRN                 sec-ft2
```

ALGEBRAIC PROGRAM 11
PEAK NUCLEATE HEAT FLUX

Loc.	Name	Value	Dimension
0	C_1	26.87	see note
1	a/g	1.0	---
2	P_r	1.88	---
3	σ	0.004	lb/ft
4	g g_0	1036.	(32.2 ft/sec^2)^2
5	C_2	62.33	see note
6	scratch	---	

Note: $C_1 = 0.144\ h_{fg}\ (g\rho_v)^2$

$C_2 = (g\rho_l - g\rho_v)^2/g\rho_l$

```
LRN
00    ---
01    (
02    RCL 5
03    X
04    RCL 4
05    X
06    RCL 3
07    )
08    y^x
09    4
10    1/x
11    EQUAL
12    STO 6
13    RCL 2
14    y^x
15    .
16    2
17    4
18    5
19    CHG SGN
20    EQUAL
21    2nd Prd 6
22    RCL 1
23    y^x
24    4
25    1/x
26    EQUAL
27    2nd Prd 6
28    RCL 0
29    2nd Prd 6
30    RCL 6
31    R/S              Q/A, 92.3
      LRN                 BTU/sec-ft^2
```

ALGEBRAIC PROGRAM 12

MINIMUM FILM BOILING WALL TEMPERATURE

Loc.	Name	Value	Dimension
0	Sp. Gr.	2.71	Specific gravity of wall
1	A	26.9	Atomic number of wall
2	σ_{lv}	0.004	Liquid surface tension, lb/ft
3	T_{crit}	1164.	Liquid, deg R
4	C_1	0.444	---
5	C_2	0.99	---
6	scratch	---	

```
LRN
00    ---
01    RCL 0
02    DIVIDE
03    RCL 1
04    EQUAL
05    Y^x
06    RCL 4
07    EQUAL
08    STO 6
09    RCL 2
10    Y^x
11    3
12    1/x
13    CHG SGN
14    EQUAL
15    2nd Prd 6
16    RCL 5
17    2nd Prd 6
18    RCL 6
19    R/S          m, 2.25
20    CHG SGN
21    INV ln x
22    STO 6
23    1
24    -
25    RCL 6
26    EQUAL
27    STO 6
28    RCL 3
29    Y^x
30    RCL 6
31    EQUAL
32    X
33    1
34    .
35    5
36    EQUAL
37    R/S          T_Leid., 830. deg R
                            370. deg F
```

ALGEBRAIC PROGRAM 13, Part 1 of 2
FILM BOILING

Loc.	Name	Value	Dimension
0	Pr_v	0.94	---
1	h_{fg}	700.	BTU/lb
2	c_{pv}	0.47	BTU/lb-deg F
3	dT	200.	deg F
4	μ_v	3.9×10^{-7}	$lb\text{-}sec/ft^2$
5	D	0.10	ft
6	ρ_l	1.86	$(lb/ft^3)/(32.2\ ft/sec^2)$
7	ρ_v	8.01×10^{-4}	" "

```
LRN
00    ---
01    RCL 1
02    DIVIDE
03    (
04    RCL 2
05    X
06    RCL 3
07    )
08    EQUAL
09    R/S          Θ, 7.45
10    (
11    (
12    RCL 5
13    y^x
14    3
15    )
16    DIVIDE
17    (
18    RCL 4
19    y^x
20    2
21    )
22    X
23    (
24    RCL 6
25    -
26    RCL 7
27    )
28    X
29    3
30    2
31    .
32    2
33    X
34    RCL 7
35    )
36    R/S          Gr, 3.15 x 10^8
37    X
38    RCL 0
39    EQUAL
40    R/S          Ra, 2.96 x 10^8
```

ALGEBRAIC PROGRAM 13 Part 2 of 2

FILM BOILING

Loc.	Name	Value	Dimension
0	θ	7.45	---
1	R_a	2.96×10^8	---
2	D	0.10	ft
3	k_v	9.0×10^{-4}	BTU/sec-ft-deg F
4	C_1	0.94	---
5	T_c	1165.	deg R
6	T_{sat}	672.	deg R
7	T_{wall}	860.	deg R

```
LRN
00    ---
01    (
02    (
03    RCL 6
04    DIVIDE
05    RCL 5
06    EQUAL
07    y^x
08    2
09    CHG SGN
10    EQUAL
11    X
12    RCL 0
13    X
14    RCL 1
15    )
16    y^x
17    4
18    1/x
19    EQUAL
20    )
21    X
22    RCL 4
23    EQUAL
24    R/S              Nu, 268.
25    X
26    (
27    RCL 3
28    DIVIDE
29    RCL 2
30    )
31    EQUAL            h, 2.41
32    X                   BTU/sec-ft^2
33    (                            -deg F
34    RCL 7
35    -
36    RCL 6
37    )
38    EQUAL
39    R/S              Q/A, 453.
      LRN                     BTU/sec-ft^2
```

ALGEBRAIC PROGRAM 14
RADIATION HEAT GAIN

Loc.	Name	Value	Dimension
0	T_{source}	5000.	deg R
1	T_{body}	500.	deg R
2	S_t-B_z	4.758×10^{-13}	BTU/sec-ft^2-deg R^4
3	$Area_{body}$	10.	ft^2
4	$(r_b/r_s)^2$	2.5×10^{-3}	square of relative radius
5	$e_s \times a_b$	0.9	---

```
LRN                              11    (
00    ---                        12    RCL 2
01    (                          13    X
02    RCL 0                      14    RCL 3
03    y^x                        15    X
04    4                          16    RCL 4
05    -                          17    X
06    RCL 1                      18    RCL 5
07    y^x                        19    )
08    4                          20    EQUAL
09    )                          21    R/S          Q, 6.69 BTU/sec
10    X                                LRN
```

ALGEBRAIC PROGRAM 15
RADIATION HEAT LOSS

Loc.	Name	Value	Dimension
0	T_{body}	500.	deg R
1	$T_{background}$	460.	deg R
2	S_t-B_z	4.758×10^{-13}	BTU/sec-ft^2-deg R^4
3	$Area_{body}$	100.	ft^2
4	e_b	0.8	---

```
LRN                              10    X
00    ---                        11    (
01    (                          12    RCL 2
02    RCL 0                      13    X
03    y^x                        14    RCL 3
04    4                          15    X
05    -                          16    RCL 4
06    RCL 1                      17    )
07    y^x                        18    EQUAL
08    4                          19    R/S          Q, 0.67 BTU/sec
09    )
```

ALGEBRAIC PROGRAM 16, Part 1 of 3
HEAT EXCHANGER EFFICIENCY

Loc.	Name	Value	Dimension
0	dW_{cold}	2.5 (1.5)	Interchange for check, lb/sec
1	dW_{hot}	1.5 (2.5)	" " "
2	c_{p-c}	0.99	BTU/lb-deg F
3	c_{p-h}	1.10	"
4	Area	20.0	ft^2
5	U	0.25	$BTU/sec-ft^2$
6	scratch	---	
7	scratch	---	

```
LRN
00   ---
01   RCL 0
02   X
03   RCL 2
04   EQUAL
05   STO 6
06   R/S           C_cold, 2.457
07   x⇌t                  (1.485)
08   RCL 1          BTU/sec-deg F
09   X
10   RCL 3
11   EQUAL
12   R/S
13   2nd x > t
14   SBR 1
15   STO 7
16   DIVIDE
17   RCL 6
18   EQUAL
19   R/S           C_min/C_max, 0.67
20   (
21   RCL 4
22   X
23   RCL 5
24   DIVIDE
25   RCL 7
26   )
27   R/S           N_tu, 3.03
28   GO TO 2
29   2nd Lbl 1
30   DIVIDE
31   RCL 6
32   EQUAL
33   1/x
34   R/S           ((C_min/C_max),
35   (                  0.54
36   RCL 4
37   X
38   RCL 5
39   DIVIDE
40   RCL 6
41   )
42   R/S           (N_tu, 3.37)
43   2nd Lbl 2
44   R/S
     LRN
```

ALGEBRAIC PROGRAM 16, Part 2 of 3
HEAT EXCHANGER EFFICIENCY, COUNTERFLOW

Loc.	Name	Value	Dimension
0	N_{tu}	3.37	---
1	C_{min}/C_{max}	0.67	---
2	C_{min}	1.65	BTU/sec-deg F
3	T_{h-i}	160.	deg F
4	T_{c-i}	80.	deg F
5	scratch	---	
6	xcratch	---	

```
LRN
00    ---
01    (
02    1
03    -
04    RCL 1
05    )
06    X
07    RCL 0
08    EQUAL
09    CHG SGN
10    INV ln x
11    STO 6
12    CHG SGN
13    +
14    1
15    EQUAL
16    STO 5
17    RCL 6
18    X
19    RCL 1
20    EQUAL
21    CHG SGN
22    +
23    1
24    EQUAL
25    1/x
26    X
27    RCL 5
28    EQUAL
29    R/S              eff., 0.86
30    X
31    (
32    RCL 2
33    X
34    (
35    RCL 3
36    -
37    RCL 4
38    )
39    )
40    EQUAL
41    R/S              Q, 113. BTU/sec
      LRN
```

ALGEBRAIC PROGRAM 16, Part 3 of 3

HEAT EXCHANGER EFFICIENCY, PARALLEL FLOW

Loc.	Name	Value	Dimension
0	N_{tu}	3.37	---
1	C_{max}/C_{min}	0.67	---
2	C_{min}	1.65	BTU/sec-deg F
3	T_{h-i}	160.	deg F
4	T_{c-i}	80.	deg F
5	scratch	---	
6	scratch	---	

```
LRN
00    ---
01    (
02    1
03    +
04    RCL 1
05    )
06    X
07    RCL 0
08    EQUAL
09    CHG SGN
10    INV ln x
11    CHG SGN
12    +
13    1
14    EQUAL
15    STO 5
16    1
17    +
18    RCL 1
19    EQUAL
20    DIVIDE
21    RCL 5
22    EQUAL
23    1/x
24    R/S                eff., 0.596
25    X
26    (
27    RCL 2
28    X
29    (
30    RCL 3
31    -
32    RCL 4
33    )
34    )
35    EQUAL
36    R/S                Q, 78.75 BTU/sec
      LRN
```

ALGEBRAIC PROGRAM 17

WATER VAPOR PRESSURE

Loc.	Name	Value	Dimension
0	Temp. Const.	460.	Deg R.
1	C_1	7.262	---
2	C_2	-3.505	---
3	Crit. Temp.	1165.	Deg R.
4	Scratch	---	---

```
LRN
00    ENTER VALUE, 32. deg F          09    RCL 2
01    +                               10    EQUAL
02    RCL 0                           11    STO 4
03    EQUAL                           12    RCL 1
04    DIVIDE                          13    SUM 4
05    RCL 3                           14    RCL 4
06    EQUAL                           15    INV 2nd log
07    1/X                             16    STOP        p = 0.0917 psi
08    X                               LRN
```

$$P = \log^{-1} (C_1 + C_2 (1/(T/T_c)))$$

ALGEBRAIC PROGRAM 18

HUMIDITY RATIO AT DRY BULB TEMPERATURE

Loc.	Name	Value	Dimension
0	W_2	0.0124	Lb water/lb dry air
1	T_1 dry bulb	80.	Deg F
2	T_2 wet bulb	64.	Deg F
3	c_{pa}	0.24	BTU/lb-deg F, specific heat
4	c_{pv}	0.44	" " "
5	H_g	1101.	BTU/lb, heat of evaporiz.
6	H_1	20.3	BTU/lb, heat of sat. liquid.
7	Scratch	---	

$$W_1 = W_2 - ((T_1-T_2)(c_{pa} + W_2\ c_{pv}))/(H_g - H_1)$$

```
LRN                    13    RCL 2
00    ---              14    )
01    (                15    EQUAL
02    RCL 0            16    DIVIDE
03    X                17    (
04    RCL 4            18    RCL 5
05    )                19    -
06    +                20    RCL 6
07    RCL 3            21    )
08    EQUAL            22    EQUAL
09    X                23    CHG SGN
10    (                24    +
11    RCL 1            25    RCL 0
12    -                26    EQUAL
                       27    STOP      W1 = 0.0088 lb/lb
```

ALGEGRAIC PROGRAM 19
HUMIDITY RATIO AT WET BULB TEMPERATURE

Loc.	Name	Value	Dimension
0	M_v/M_a	0.6215	
1	P_v	0.2947	psi, vapor pressure
2	P_{atm}	14.69	psia, Atmo. Press.

$$W_2 = (M_v/M_a)\ P_v/(P_{atm} - P_v)$$

```
LRN
00    ---                                 07    X
01    (                                   08    RCL 1
02    RCL 2                               09    EQUAL
03    -                                   10    X
04    RCL 1                               11    RCL 0
05    )                                   12    EQUAL
06    1/x                                 13    STOP    W2 = 0.0127 lb/lb
                                          LRN
```

ALGEBRAIC PROGRAM 20
ENTHALPY OF AIR OR WATER VAPOR

Loc.	Name	Value		Dimension
		Air	Water	
0	C_1	1202.	925.	
1	C_2	0.3	9.8	
2	C_3	860.	1166.	
3	C_4	460.	460.	Reference, deg R
4	n	1.0	0.7	

$$H = C_1 - C_2\ (C_3 - (460 + T)^n$$

```
LRN
00    Enter T, 64 deg F                   09    RCL 4
01      +                                 10    EQUAL
02    RCL 3                               11    X
03    EQUAL                               12    RCL 1
04    CHG SGN                             13    EQUAL
05    +                                   14    CHG SGN
06    RCL 2                               15    +
07    EQUAL                               16    RCL 0
08    y^x                                 17    EQUAL
                                          18    STOP     H = 20.31 water
                                          LRN            H = 1101.2 air
                                                              BTU/lb
```

ALGEBRAIC PROGRAM 21
VAPOR PRESSURE AT DRY BULB TEMPERATURE AND RELATIVE HUMIDITY

Loc.	Name	Value	Dimension
0	W_1	0.0088	Humidity ratio, lb/lb
1	P_{atm}	14.69	psia
2	M_a/M_v	1.6088	
3	P_{v1-sat}	0.5015	psi

$$P_{v1} = W_1 P_{atm} (M_a/M_v)/(M_a/M_v) W_1)$$

$$\phi = P_{v1}/P_{v-sat}$$

```
LRN
00   ---
01   (
02   RCL 0
03   X
04   RCL 2
05   )
06   +
07   1
08   EQUAL
09   1/x
10   X
11   (
12   RCL 2
13   X
14   RCL 1
15   )
16   EQUAL
17   X
18   RCL 0
19   EQUAL
20   STOP        P v1 = 0.2051 psi
21   DIVIDE
22   RCL 3
23   EQUAL
24   STOP        Ø = 0.409 or 41 %
LRN
```

APPENDIX II.

ENERGY ANALYSIS DATA

FOREWORD

LIST OF FIGURES

LIST OF TABLES

FOREWORD

The energy analyst has a frequent need to estimate the properties of materials for which he is unable to discover values in the literature. Fortunately, there is a good deal of order in the universe.

One valuable tool in this search is the Principle of Corresponding States. This principle tells us that if the molecules are nearly the same and if their state is the same (liquid, gas or solid) then their properties should be proportional to the ratio of the temperature to the critical temperature.

For example, consider the elements nitrogen, with a molecular weight of 14.0 and oxygen with a molecular weight of 16.0. The critical temperature of nitrogen is 228 deg R and for oxygen is 279 deg R. Assume that we are given the thermal conductivity, k, of nitrogen at 182 deg R as 1.66×10^{-5} BTU/sec-ft-deg R. What might we expect for the value of oxygen?

1. The critical temperature ratio for nitrogen is

$$R_N = 182/228 = 0.80$$

2. The temperature of oxygen at 0.80 critical is

$$T_O = 0.80 \times 279 = 223 \text{ deg R}$$

3. Compare the actual value for k to that predicted by the principle of corresponding states (PCS).

By PCS, $k = 1.66 \times 10^{-5}$ BTU/sec-ft-deg R

Actual, $k = 1.72 \times 10^{-5}$ " "

The critical point itself is a valuable aid to organizing thermal and physical data. Surface tension and heat of vaporization vanish at the critical point. Specific weight of the gas and of the liquid are identical at the critical point. Vapor pressure, as shown previously, is found to vary as the logarithm of the inverse temperature ratio. Critical points and normal boiling points (boiling temperature at one atmosphere pressure) are usually found from tables in chemical handbooks, see <u>Engine Thermodynamics</u>, Ref. 14.

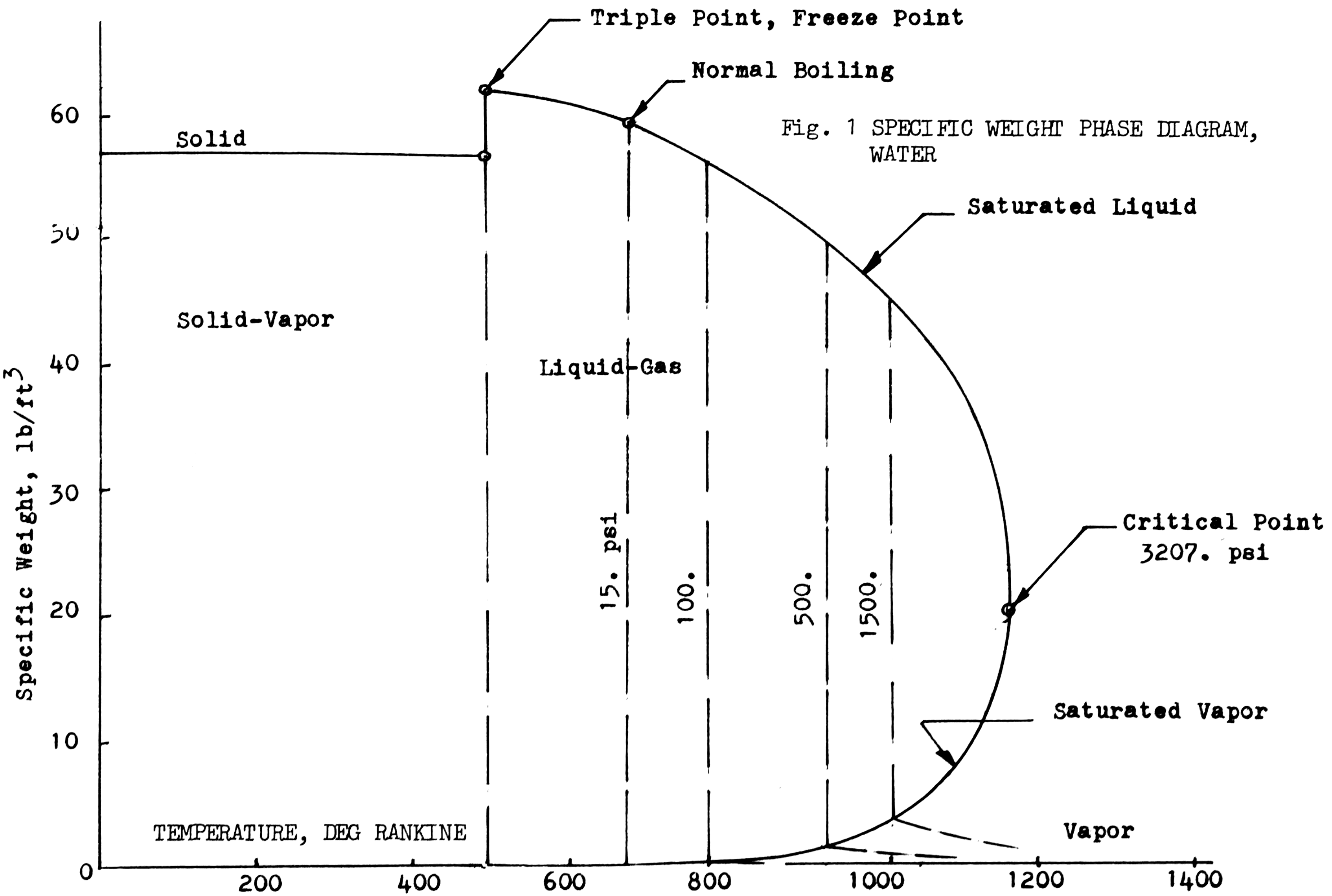


Fig. 1 SPECIFIC WEIGHT PHASE DIAGRAM, WATER

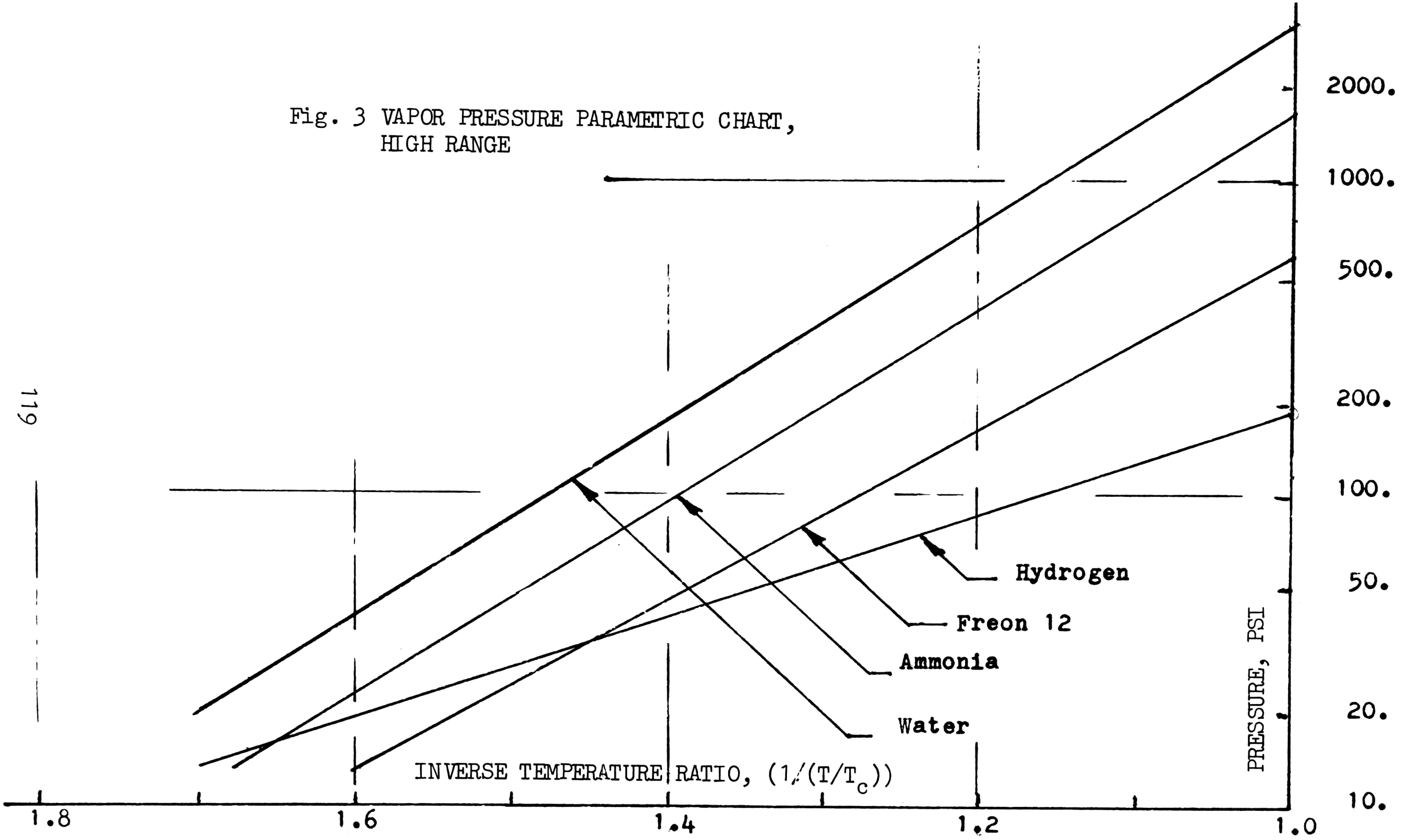


Fig. 3 VAPOR PRESSURE PARAMETRIC CHART, HIGH RANGE

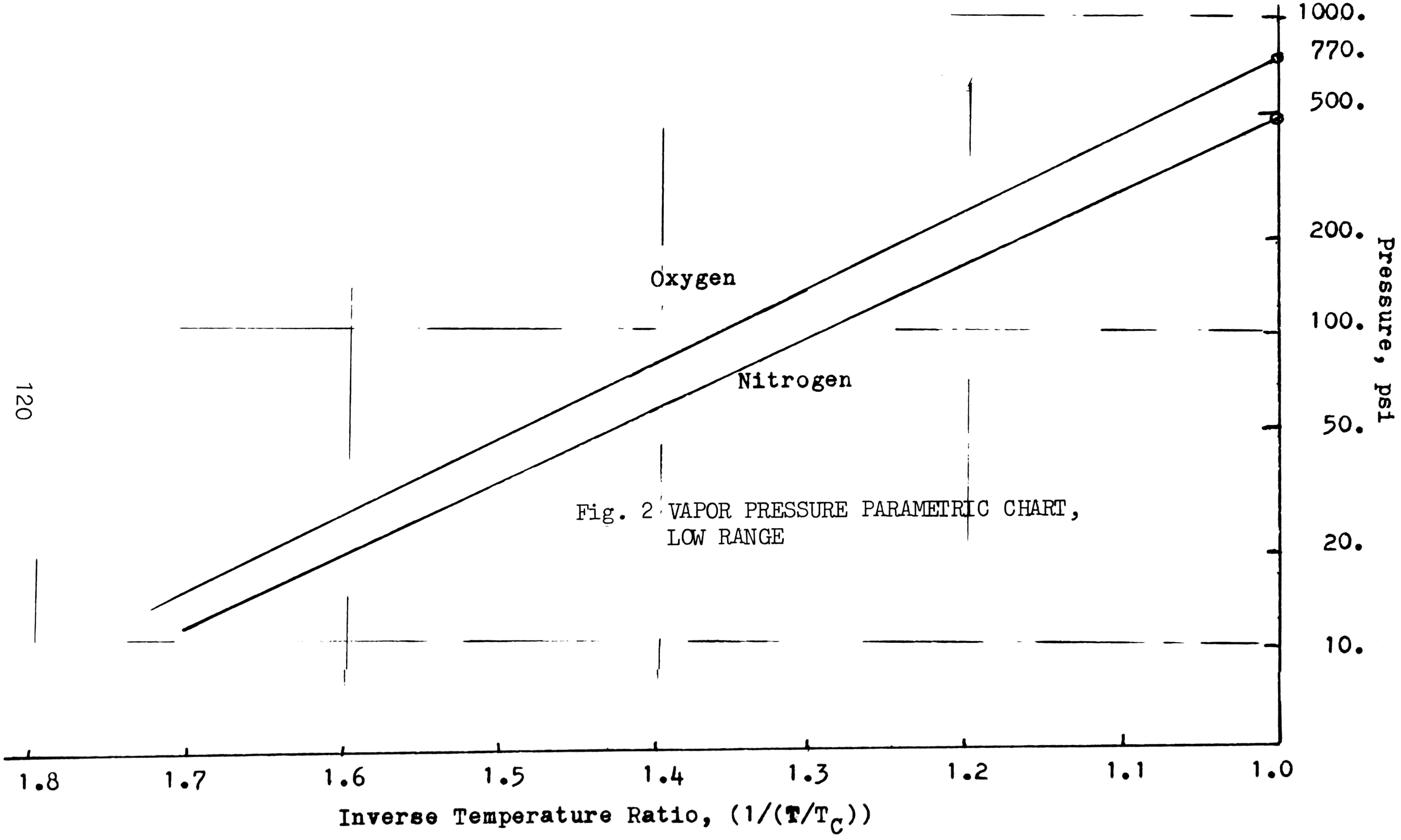


Fig. 2 VAPOR PRESSURE PARAMETRIC CHART, LOW RANGE

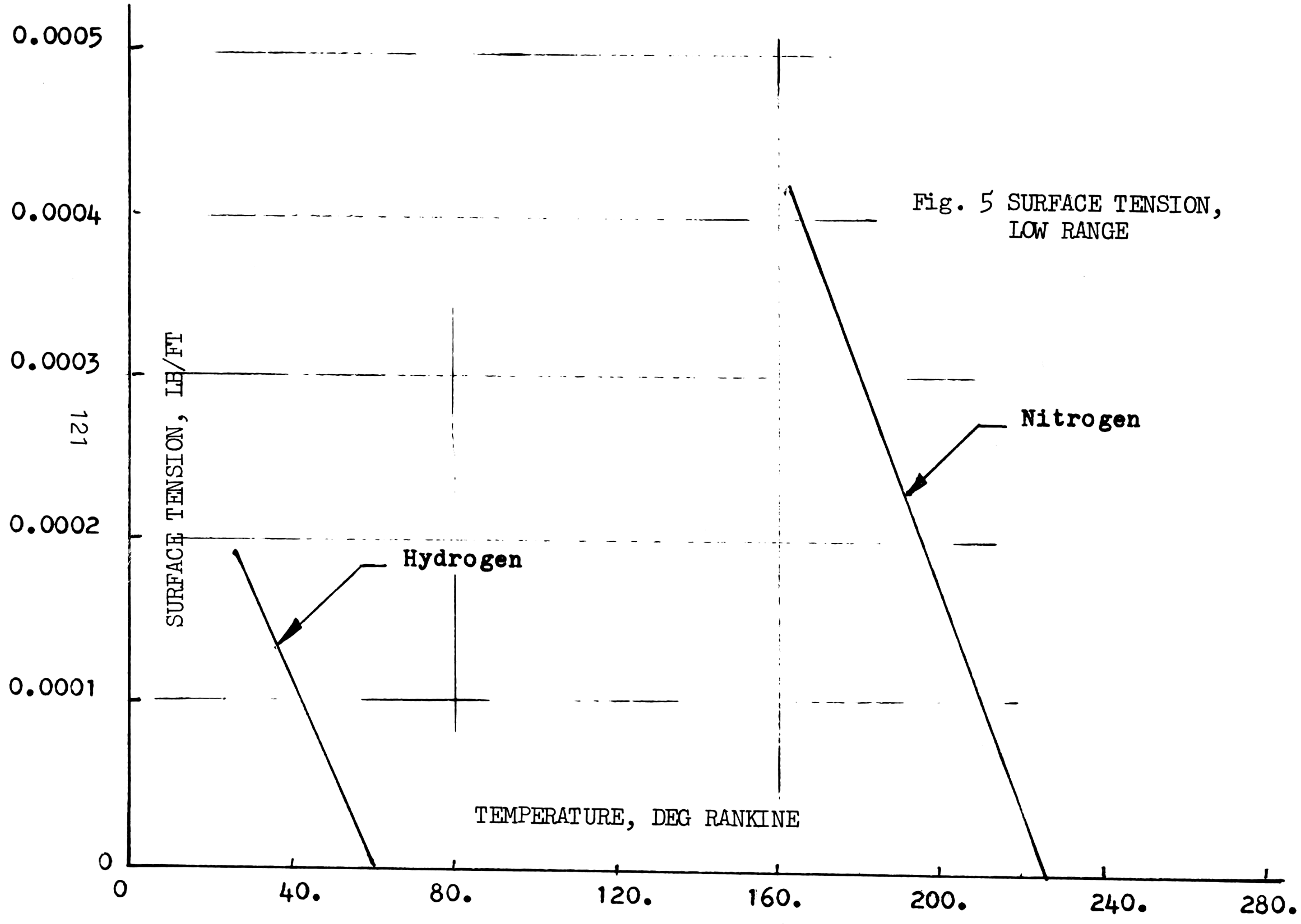


Fig. 5 SURFACE TENSION,
LOW RANGE

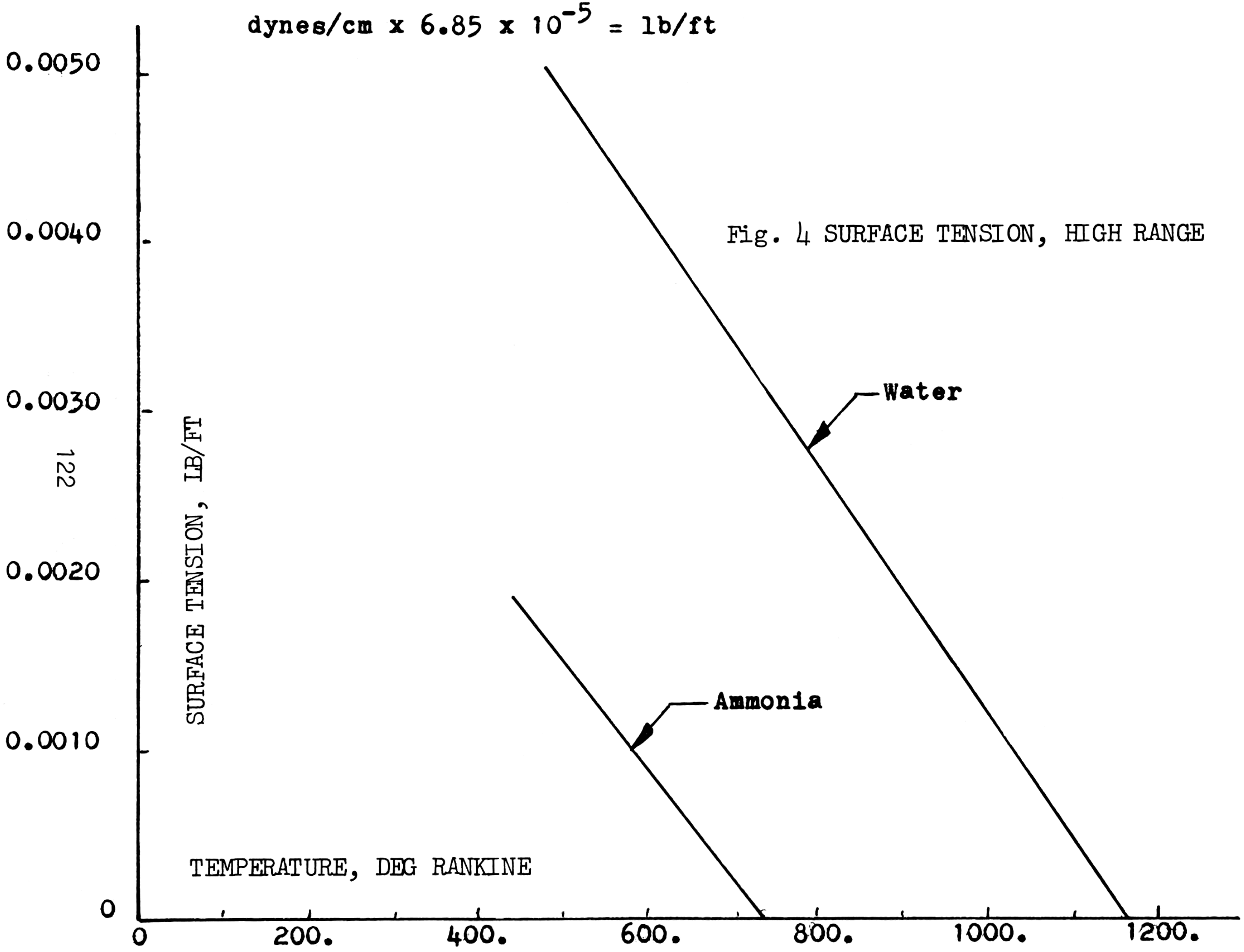


Fig. 4 SURFACE TENSION, HIGH RANGE

TABLE 1. CONDUCTIVITY AND SPECIFIC WEIGHT, GASES

MATERIAL (@ 14.7 PSI)	TEMP. deg F	SPECIFIC WEIGHT lb/ft^3	CONDUCTIVITY BTU/sec-ft-deg F
Air, Nitrogen	32.	0.080	3.77×10^{-6}
	212.		4.88×10^{-6}
Hydrogen	32.	0.00561	2.62×10^{-5}
	212.		3.33×10^{-5}
Ethane	32.	0.0847	2.92×10^{-6}
Chlorine	32.	0.201	1.23 "
Carbon Dioxide	32.	0.123	2.24 "
	212.		3.07 "
Ammonia	32.	0.048	3.26 "
	212.		4.47 "
Water Vapor	212.	0.0372	4.03 "
Freon 12	70.	0.0338	

TABLE 2. CONDUCTIVITY AND SPECIFIC WEIGHT, LIQUIDS

MATERIAL	TEMP. deg F	SPECIFIC WEIGHT lb/ft^3	CONDUCTIVITY BTU/sec-ft-deg F
Water	32.	62.	8.91×10^{-5}
	140.		1.05×10^{-4}
	270.		6.25×10^{-3}
Kerosene	68.	48.	2.43×10^{-5}
Ammonia	32.	40.0 (62. PSIA)	8.66×10^{-5}
Freon 12	32.	87. (45. PSIA)	1.16×10^{-5}

TABLE 3. CONDUCTIVITY AND SPECIFIC WEIGHT, SOLIDS

MATERIAL	SPECIFIC WEIGHT lb/ft^3	THERMAL CONDUCTIVITY k BTU/sec-ft -deg F	k BTU-in/hr $-ft^2$-deg F	1/k "R/inch"	"R-value" (4 inch)
Glass fiber	4. to 9.	6.11×10^{-6}	0.26	4.	16.
Polyurethane, foam	2.5	3.94×10^{-6}	0.17	6.	24.
Expanded Vermiculite		1.04×10^{-5}	0.45	2.	8.
Expanded Perlite		8.56×10^{-6}	0.37	3.	12.
Window Glass	162.	1.42×10^{-4}	6.13	0.2	
Corkboard	10.	6.94×10^{-6}	0.30	3.	12.
Ice	57.	3.56×10^{-4}	15.38	0.1	
Wood, Oak	51.	6.39×10^{-5}	2.76	0.4	
Wood, Balsa	7.3	7.41×10^{-6}	0.32	3.	12.
Fiberboard; Cellulose	14.	6.67×10^{-6}	0.29	4.	16.
Asbestos		1.29×10^{-5}	0.56	2.	8.
Plaster-Sand		1.29×10^{-4}	5.6	0.2	
Plaster-Sand, wood lath		5.79×10^{-5}	2.5	0.4	
Concrete Block, cored		1.30×10^{-4}	5.60	0.2	
Brick, Common	100.	1.16×10^{-4}	5.0	0.2	

TABLE 4. CONDUCTIVITY AND SPECIFIC WEIGHT, METALS

MATERIAL	TEMP. deg F	SPECIFIC WEIGHT lb/ft^3	CONDUCTIVITY BTU/sec-ft-deg F
Aluminum	32.	168.	3.41×10^{-2}
Brass	68.	528.	1.78×10^{-2}
Copper	32.	557.	6.20×10^{-2}
Gold	32.	1204.	4.79×10^{-2}
Iron	64.	491.	1.08×10^{-2}
Steel	32.	490.	7.18×10^{-3}
Nickel	32.	552.	9.44×10^{-3}
Platinum	32.	1338.	1.12×10^{-2}
Silver	32.	655.	6.50×10^{-2}
Sodium	32.	60.	2.19×10^{-2}
Zinc	32.	445.	1.79×10^{-2}

TABLE 5. VISCOSITY, SPECIFIC HEAT AND EXPANSION OF LIQUIDS

MATERIAL	TEMP. deg F	VISCOSITY lb-sec/ft^2	EXPANSION COEFFICIENT ft^3/ft^3-deg F	SPECIFIC HEAT BTU/lb-deg F
Water	32.	3.73×10^{-5}	-0.5×10^{-4}	1.01
	140.	1.11 "	2.5 "	1.00
	270.	5.18×10^{-6}	4.5 "	1.00
Kerosene	-50.	1.25×10^{-4}		
	60.	4.39×10^{-5}	5.0 "	0.50
	200.	1.44 "		
Ammonia	32.	5.00×10^{-6}	1.2×10^{-3}	1.11
Freon 12	32.	6.21 "	1.7 "	0.223

TABLE 6. VISCOSITY, SPECIFIC HEAT AND EXPANSION OF GASES

MATERIAL	TEMP. deg F	VISCOSITY lb-sec/ft^2	EXPANSION COEFFICIENT ft^3/ft^3-deg F	SPECIFIC HEAT BTU/lb-deg F
Water Vapor	212.	2.71×10^{-7}	2.18×10^{-3}	0.45
	500.	1.97 "		0.47
	1000.	3.48 "		0.51
	2000.	6.55 "		0.60
Hydrogen	59.	1.84×10^{-6}	2.0×10^{-3}	3.39
	392.			3.46
Ammonia	32.	1.92×10^{-4}		
	59.			0.52
	212.	2.69 "		
Air	0.	3.45×10^{-7}	2.0×10^{-3}	0.239
	100.	3.96 "		0.241
	200.	4.48 "		0.242
	500.	5.86 "		0.248
	1500.	9.14 "		0.277

TABLE 7. RATIO OF SPECIFIC HEATS, MOLECULAR WEIGHT AND GAS CONSTANT

MATERIAL	c_p/c_v	GAS CONSTANT, R	MOLECULAR WEIGH
Air	1.40	53.3	29.
Hydrogen	1.40	767.	2.0
Ammonia	1.31	89.5	17.0
Water Vapor	1.32	85.78	18.0

TABLE 8. HEAT OF EVAPORATION AND FUSION

MATERIAL	HEAT OF EVAPORATION @ one atmosphere BTU/lb		HEAT OF FUSION BTU/lb
Water	970.	(212. deg F)	143.
	1076.	(32. deg F)	
Ammonia	588.		194.
Hydrogen	192.		25.
Oxygen	92.		6.
Nitrogen	86.		11.

TABLE 9. RADIATION SURFACE TEMPERATURES

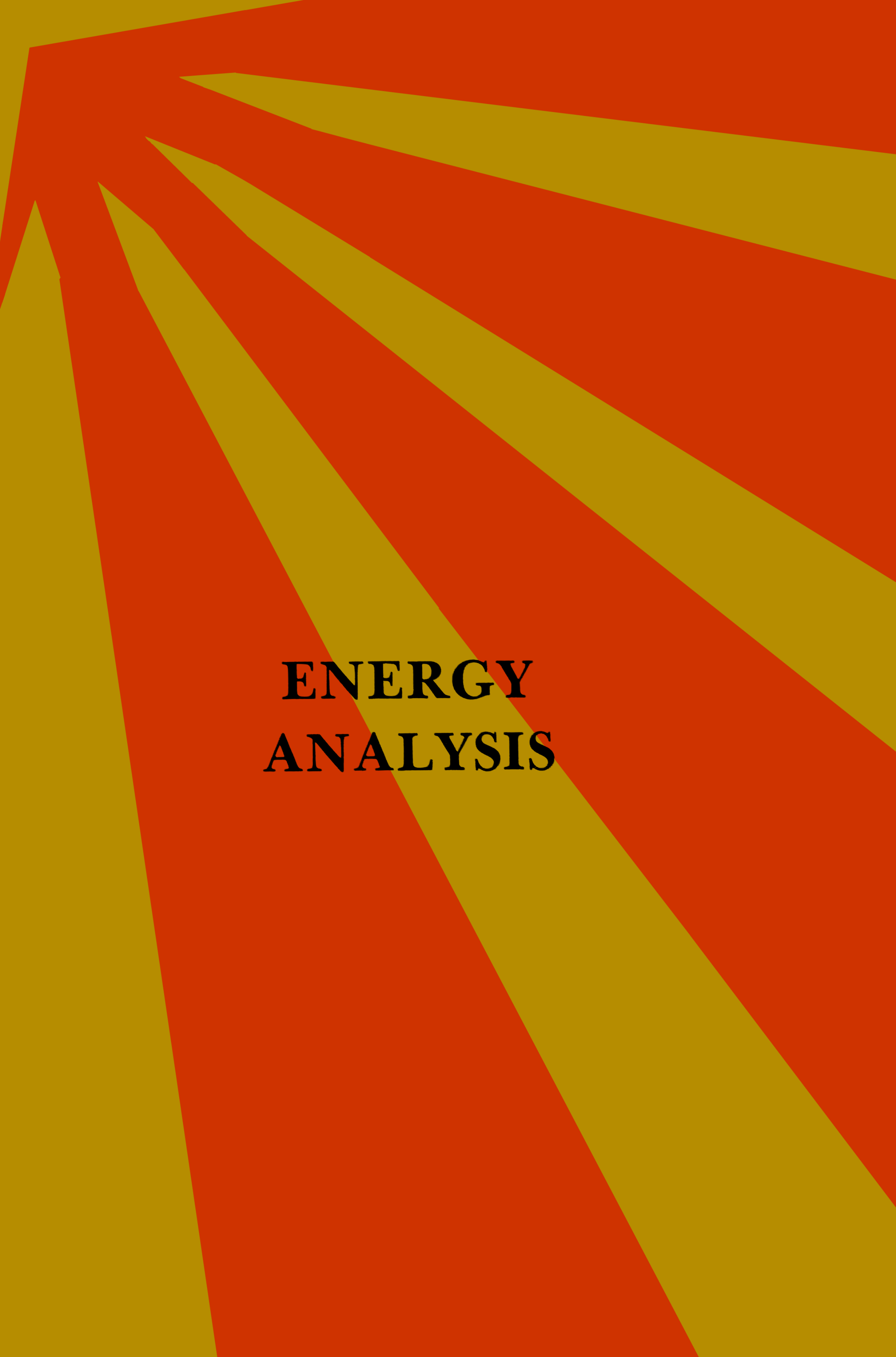
SURFACE	TEMPERATURE deg R
Incandescence	
Red, Visible in dark	1200-1300
Red, Visible in sunlight	1500
Yellow	2200-2300
Blue-White	3400
Sun	10800
Molten Metal	
Lead	1080
Sulfur	706
Copper, Gold	2440
Silicon	3050
Iron	3240
Platinum	3651

TABLE 10. SPECIFIC WEIGHT PHASE CHANGE DATA

MATERIAL	FREEZE		NORMAL BOIL		CRITICAL POINT		
	TEMP. deg R	SPECIFIC WEIGHT lb/ft^3	TEMP. deg R	SPECIFIC WEIGHT lb/ft^3	TEMP. deg R	SPECIFIC WEIGHT lb/ft^3	PRESS. lb/in^2
Water	492.	57.5 solid 62.5 liquid	672.	59.9 liquid 0.037 gas	1165.	19.2	3207.
Ammonia	352.	solid liquid	440.	40.6 liquid gas	730.		1636.
Freon 12 ($C\ Cl\ F_3$)	213.	solid liquid	438.	liquid 0.338 gas	692.		582.
Hydrogen	26.	5.4 solid 4.8 liquid	36.	4.36 liquid 0.08 gas	59.8	2.05	188.

TABLE 11. RADIATION PROPERTIES OF SURFACES

MATERIAL	EMISSIVITY		ABSORPTIVITY
Lampblack	0.96		0.97
Oxide Paints (all colors)	0.94		
Aluminum Paint, Bronze Paint	0.3 to 0.6		0.35
White Paper			0.25
Non-metallic objects	0.92 to 0.96		
	Polished Metal	Oxidized Metal	Polished Metal
Aluminum	0.05	0.10 to 0.20	
Brass	0.05	0.25 to 0.60	
Copper	0.04	0.55 to 0.75	0.50
Silver	0.025		0.07
Gold	0.03		
Iron, Steel	0.20	0.06 to 0.90	0.45
Platinum	0.036		
Nickel	0.06	0.04 to 0.60	0.40
Zinc	0.10		0.65



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