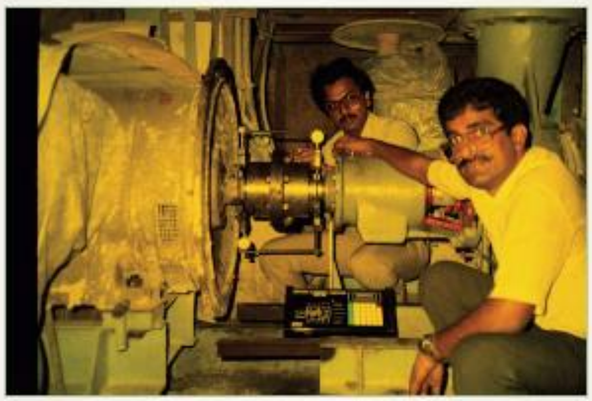
Manual and Guide

ENERGY EFFICIENCY

&

PLANT MAINTENANCE



Kaup Shenoy Associates

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Cover Page Photo:
Mr. V. V. Sundaram & M. Sreenivas Reddy (ex-NPC colleagues) checking stock pump coupling alignment in a pulp and paper mill.

Foreword

I have had the pleasure of knowing the author, Mr. M.K. Shenoy, since 1994. He has made it possible for me to be introduced to experts of National Productivity Councils (NPC) and to visit several key factories and colleges around India. This provides me valuable experience by observing how NPC engineers practice their expertise on real-life applications. I have been able to witness the success of satisfying a substantial demand from industry on their machinery and equipment regarding the optimum operation, low energy consumption, high reliability, and long service life.

Energy efficiency analysis of engineering equipment and plant operation planning has traditionally used ready-made nomographs and data tables to make approximate estimates or best guesses. However, in real-life applications, the operating state of the equipment changes in real-time and is determined by many interrelated parameters. Only relying on conventional tools such as graphs and tables that are not connected to each other cannot meet the latest industrial requirements. Innovations are now using smartphones and high-speed internet technology that can quickly integrate various engineering resources for a specific topic, draw efficiency analysis and operation strategy, and provide efficient and accurate recommendations. The kernel that makes a mobile phone "smart" is from the software embedded inside it. The App Suite was developed to act as a backbone of the phone for this purpose. The App Suite literally contains the brain of the author. In just holding the phone, the user possesses the complete “state-of-the-art” of a subject matter as it is known by the experts that created the package. Properly utilizing such an App Suite gives any user the power of a master engineer and the expertise of an entire industry. This Manual and Guide is a companion to the App Suite that gives easy-to-use guidelines and has compiled all relevant information in one convenient document. An important advantage of this manual is the standardization of maintenance procedures with illustrative examples that allow readers to easily follow industrial standards to achieve daily engineering design and maintenance goals more effectively and efficiently.

This manual presents the App Suite in a nutshell. It captures the professional experience of the author as a designer, researcher, and maintenance expert. The text provides a concise theoretical background, procedural maintenance practice, illustrative examples, and rule of thumb in performance trend monitoring. The Manual and the App Suite will serve both the industry and academia well.

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Preface

This is like a "**Manual"** for the application suite that is published for different smart phone platforms providing additional information on implementation and validation of various math models presented. Further, this addresses anomaly reported in some of the apps. For example, very recently one user reported thinking that is like anomaly in "Fluid DeltaP" app. Sent screen shots of pressure drop of hydraulic fluid at 10/20/30 °C. It is logical that at 10°C the pressure drop is more owing to increase in viscosity compare to 20 °C. However, pressure drop was also more at 30 °C compare to 20 °C. Thought it was an anomaly. Upon looking at the Reynold's number, flow is turbulent at 30 °C hence inertia dominant. So long as the flow is laminar, which is viscous-dominant, the pressure drop varies as temperature. There are number of similar situations where app users look for clarity and therefore used examples in this manual mainly from standard textbooks for validation and verification.

This is also like a "**Guide"** for using the application suite for performance trend monitoring to access different maintenance requirements in most common process equipment such as boilers, heat exchangers, pumps, compressors, and fans. This is highlighted in separate section on maintenance of respective equipment. Also put a separate section discussing the managerial aspects of productive maintenance to achieve cost effectiveness. In other words, "***Productive maintenance can become proactive by proper implementation of performance trend monitoring using apps***."

Last but not the least likes to acknowledge all ex-colleagues of National Productivity Council of India for providing the necessary help and guidance during development stage of PC version of the app suite. Especially late Shri R. Govind Rao who gave timely suggestions to improvise. Also received inputs from S/Shri R. Vasu, T. Raja Chidambaram, and D.P. Mujumdar at different capacities. Finally, I acknowledge the contribution of my daughter Shivany and son Siddharth as engineers in testing and validation of all apps.

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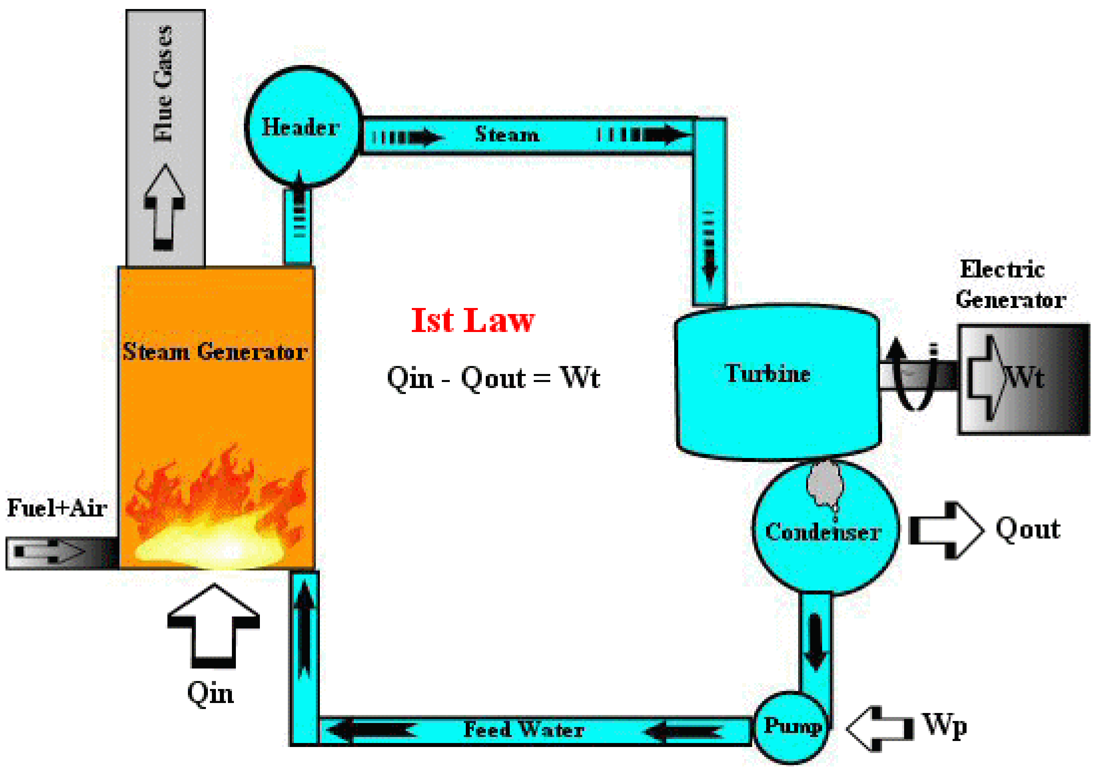
1. Introduction

1.1 Energy Efficiency

Energy Efficiency is synonymous with efficient management of plant and machineries in twenty-first century for productivity and overall cost effectiveness. Therefore, it is very essential to assess both equipment efficiency and its maintenance requirements time to time throughout the productive life cycle. This can be further illustrated with a simple example using our personal automobile. When a car is new, we often do not pay much attention to its fuel consumption in terms of miles per gallon (mpg) or kilometer per liter (kmpl). So long as we do all the periodic recommended schedule maintenance such as oil filter change, tire rotation, alignment and inflation checking etc. it continue to maintain the same level of fuel consumption and its energy efficiency level. However, as car put more mileage any excessive wear and tear of engine or transmission could result in to increase in its fuel consumption as well. Thus, both energy efficiency and maintenance are closely related or connected.

1.2 Law of Thermodynamics

Treatise on energy efficiency and maintenance engineering needs a very disciplined approach. The law of conservation of energy states "Energy neither can be created nor destroyed but can be transformed from one form to another or transformed from one system to another." Thus, the study of energy transformations as well as devices and materials used to accomplish this transformation is called thermodynamics. Hence all cyclical devices such as heat engines and heat pump obey the law of thermodynamics. The first law states, "Net amount of heat added to a cyclical device equals net amount of work produced." This is illustrated using a generic model of a heat engine as shown below.



Every heat engine has a source or reservoir where heat is added to a fluid medium at certain elevated temperature 'T_h' and a sink where heat is rejected at certain lower temperature 'T_l'. These two temperatures are the key parameters to know the upper bound on the amount of work a heat engine can produce and known as Carnot cycle efficiency.

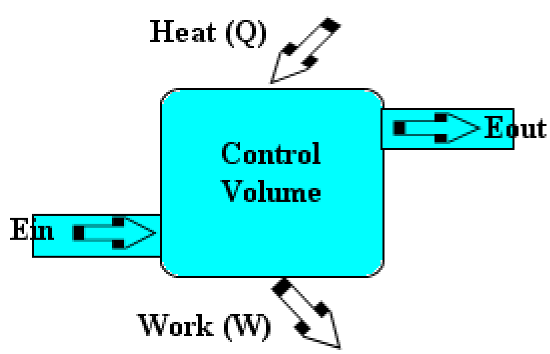
$$\eta_{carnot} = \frac{T_h - T_l}{T_h}$$

The second law dictates how far this conversion can take place within a device and brings in the concept of entropy would lie somewhere below this line. Moreover, the overall efficiency is further reduced since its

own auxiliaries such as pumps, fans etc., which utilize a small part of the power so generated to keep the fluid medium under circulation.
That means not all heat energy added to the device can produce work and hence we need to measure its conversion efficiency. Hence both heat and work can be quantified using the same units of measurement and often interchangeable.

Energy conversion devices such as boilers, turbines, turbo engines, pumps and compressors can be treated as steady flow devices that have one or more fluid streams entering as well as leaving a control volume. The most important presumption is their stream properties at the inlet or outlet always remain the same regardless of the time. This concept is presented in the following model along with the energy balance equation and applicable to all flow systems analyzed in the app suite.

First Law for Steady Flow System



$$Q - W = E_{out} - E_{in}$$

$$\dot{Q} - \dot{W} = \sum_{out} (h + e_p + e_k)\,\dot{m} - \sum_{in} (h + e_p + e_k)\,\dot{m}$$

1.3 Energy Conservation

Conversion process associated with all heat engines such as steam, gas etc., are primarily dictated by the laws of thermodynamics. Accordingly, no heat engine can ever produce work more than the Carnot cycle engine, which works between the same reservoirs (T_h) and sink (T_l) temperatures as discussed in energy conversion principles. Therefore, the work potential of any energy conversion process can be readily assessed using this concept and given as follows.

$$W_p = Q_c \, (1 - T_l \, / \, T_h)$$

Where Q_c is the amount of energy converted or transferred from the reservoir to the medium, any difference in the actual work and the work potential can be measured as conversion loss.

Similarly, transmission losses are the energy losses that occur between two energy conversion devices. Transmission loss in case of electrical energy system can be as high as 10% from the generating source to the consumer. Similarly, flow energy (pressure) loss coupled with heat in case of fluids can be much greater. Longer the transmission bigger the energy losses are.

Transmission losses in mechanical systems are often considered as a part of the conversion device efficiency and can be generalized as follows

IHP = BHP + FHP

Mechanical efficiency = BHP/IHP

Where

IHP: Indicated horsepower (input);

BHP: Brake horsepower (output)

FHP: Frictional Horsepower that measures the rate at which a mechanical energy is being converted into heat and dissipated back to the sink. This is the amount of heat generated due to friction between various mechanical elements such as shaft and bearings; speed reduction gear drives; belt drives etc.

1.4 Application Suite

Goal of any energy conservation program is to minimize both energy conversion loss as well as transmission loss. A detailed design analysis can unearth all the potential areas of energy conservation in a plant that are energy intensive such as utility power, petroleum, pulp and paper, and other chemical and process industries. Accordingly, an app suite consists of 12 individual apps have been developed that serves as onsite tool for energy engineers, auditors and technical personnel and published for different smartphone platforms such as Apple, Android, Windows and Amazon devices. This is further described under separate chapters of this book for its technical content, usage, and other application guidelines. However, people can also review these apps at app stores for their respective device platform. Screen shots are presented using specific case examples. Additional information with screenshots is also available at www.flowjoule.com

Apps can be grouped under three different categories such as supporting apps, device apps and energy cycle apps as given below.

Supporting Apps:

Thermo-Physical Properties

Fluid ΔP

Orifice flow

Engineering unit conversion

Device Apps:

Boiler Fuel Efficiency

Centrifugal Pump Specification

Fan Specification Performance

Compressor Capacity

Heat Exchanger Performance

Energy Cycle Apps

Rankine Cycle

HVAC Cooling Cycle

HVAC Heating Cycle

Users can save all their application data pertaining to energy cycles as well as major devices (boiler, pump, heat exchangers) in a file with unique title for reviewing and monitoring individual performance time to time.

1.5 Plant Maintenance System

The above app suite can perform the basic system design analysis to identify the potential areas for further analysis and modifications. This could also be a part of a regular performance-monitoring tool for different plant and machines in the system that may be degrading.

Consequently, it can aid the best practices under operation and maintenance. Some of these can be listed below.

- Periodic cleaning to improve and maintain heat transfer efficiency
- Excess combustion air control
- Fuel quality control to maximize conversion efficiency
- Fluid quality control (water treatment, blow down etc.)
- Optimization of system parameters (temperature, pressure and flow) for a given load
- Prioritization of shut down activities and conserve maintenance resources
- Comprehensive pipeline auditing

Transmission losses in mechanical systems mainly come from the frictional heat generated between various mechanical elements. The only way to combat this loss is by effectively implementing lubrication program as a part of preventive maintenance. A mal lubrication to any mechanical drives is like a double edge sword that can cut both ways meaning it increases the transmission losses and at the same time reduces the life of the component due to excessive wear and tear. More than 70% of the mechanical failures are attributed to lubrication failures. Therefore, a good maintenance management program is also a part of the energy conservation program.

A separate chapter is dedicated to deal with the advanced plant maintenance systems and practices covering managerial aspects. This is in addition to the specific technical aspects of individual devices described as a part of application suite.

2. Thermodynamic, Transport and Flow Properties

This app provides important thermo physical property data on fluids that are quite common in industrial applications. The data helps in computation of various parameters pertaining to flow system design involving heat and energy transfers. Transport properties such as density, viscosity, specific heat, and thermal conductivity primarily are function of temperature hence needs to be determined every time when system temperature changes. Here is the list of industrial fluids.

1. *Water*
2. *Steam*
3. *Air*
4. *Nitrogen*
5. *Oxygen*
6. *Hydrogen*
7. *Helium*
8. *CO_2*
9. *Methane*
10. *Ethane*
11. *Chlorine*
12. *Ammonia*
13. *Argon*
14. *Hydraulic Oil*
15. *HFC (R410A)*
16. *Mercury*

For steam, additional thermodynamic property data are provided in the form of a Steam Table covering all 3 states such as saturated water, saturated vapor and superheat conditions at a given pressure.
The property data is presented in both (SI and USCS) units depending upon the user's choice for given pressure and temperature conditions and divided in two parts. The first part is on transport properties that are common to all fluids and the second part is on property data that are exclusively for gases and steam as given below.

- Thermo-physical Transport properties.
    - Density (ρ) – function of both temperature and pressure
    - Viscosity (μ) – function of temperature
    - Specific Heat (constant pressure-cp) – function of temperature
    - Thermal conductivity (K) – function of temperature
- Additional property data for gases including steam.
    - Molecular wt. (MW)
    - Specific heat ratios (cp/cv)
    - Thermal diffusivity (α)
    - Prandtl number (Pr)
- Additional thermodynamic properties exclusively for steam.
    - Saturation temperature at given pressure
    - Specific volume (v)
    - Specific Internal energy (u)
    - Specific enthalpy (h)
    - Specific entropy (s)

2.1 Engineering Units

App suite uses SI units for all internal calculations while the 'user interface edit boxes will have some default units for all input variables. In most cases, the user will have an option to change and select units for both input and outputs. The program automatically handles the unit conversion from SI to Metric or SI to USCS and vice versa. The following table enlists the units for different property that can be preset for both Input and output display.

Engineering Unit Conversion app covers 18 major categories as follows. By default, app provides conversion factor. However, users can enter data to convert its value in another unit also generate the conversion table.

1. *Mass (kg, lbm, slug, ounce, grain, M. Tonne, Ton)*
2. *Length (meter, ft, cm, in, mm, km, mile)*
3. *Area (m^2, ft^2, cm^2, in^2)*
4. *Volume (liter, gal (US), gal (UK), m^3. Ft^3, fl_oz)*
5. *Temperature (deg C, deg F, Kelvin, Rankin)*
6. *Flow rate (lpm, gpm, cc/sec, ci/sec)*
7. *Force (Newton, dyne, kgf, lbf)*
8. *Pressure (kPa, Bar, psi, atm, kgf/cm^2, in_wg, mm_Hg)*
9. *Velocity (m/sec. ft/sc, kmh, mph)*
10. *Energy (Joule, cal, BTU, m.kgf, ft.lbf)*
11. *Power (kW, hp, kcal/s, BTU/hr, kgf.m/s, lbf.ft/s)*
12. *Density (kg/m^3, lb/ft^3, m/cc, lbf.sec^2/in^4)*
13. *Dynamic Viscosity (Pa.sec, poise, cP, reyn, kgf.sec/m^2, lbf.sec/ft^2)*
14. *Kinematic viscosity (stokes, cSt, newts)*
15. *Specific heat (joule/kg.K, BTU/lb.F, cal/kg.C)*
16. *Thermal Conductivity (W/m.K, BTU/hr.ft.F)*
17. *Enthalpy (kJ/kg, BTU/lb)*
18. *Entropy (kJ/kg.K, BTU/hr.R)*

2.2 Transport Properties of Gases

2.2.1. Density

Density is computed for the given pressure and temperature using the ideal gas law as follows:

$$\rho = \frac{P}{R_g \cdot T} \qquad (1)$$

Where:

P – Absolute pressure

R_g – Specific gas constant (Universal gas constant / molecular weight)

T – Absolute temperature

For example, density of air at atmospheric pressure and 20°C is given as:

P =1 atm =14.698psi = 101.325kPa

Universal Gas constant = 8.314 (joule/gm.mol.K)

MW = 28.966 gm/gm.mol

Rg = (Universal Gas constant/MW)

T = 20 + 273.15 = 293.15.K

Substituting the values in the above equation 1, we get the density of air as:

$\rho = 1.204\ kg/m^3 = 0.075\ lbm/ft^3$

2.2.2 Specific Heat

Specific heat C_p for any gas as a function of temperature is given by a fifth-degree polynomial equation:

$$C_p(T) = (A_0 T^0 + A_1 T^1 + A_2 T^2 + A_3 T^3 + A_4 T^4 + A_5 T^5) 10^3 \frac{Joule}{kg.K} \qquad (2)$$

Where T = Absolute temperature in degree Kelvin.

A_0, A_1, A_2, A_3, A_4, A_5 are polynomial coefficients.

For air, the above coefficients are given as:

$A_0 := 1.03409$

$A_1 := -0.2848870 \cdot 10^{-3}$

$A_2 := 0.7816818 \cdot 10^{-6}$

$A_3 := -0.4970786 \cdot 10^{-9}$

$A_4 := 0.1077024 \cdot 10^{-12}$

Substituting these in equation 2, the specific heat of air at 293.15 K (68°F) is given as:

$C_p(293.15)$ = 1.006 Joule/kg .K = 0.24 BTU/lb.R

2.2.3. Viscosity

Viscosity μ for any gas as a function of temperature is given by a fifth-degree polynomial equation:

$$\mu(T) = (C_0 T^0 + C_1 T^1 + C_2 T^2 + C_3 T^3 + C_4 T^4 + C_5 T^5) 10^{-6} \, Pa.s \qquad (3)$$

Where: T = Absolute temperature in degree Kelvin.

C_0, C_1, C_2, C_3, C_4, C_5 are polynomial coefficients.

For air, the above coefficients are given as:

$C_0 := -0.98601$

$C_1 := 0.09080125$

$C_2 := -1.17635575 \cdot 10^{-4}$

$C_3 := 1.2349703 \cdot 10^{-7}$

$C_4 := -5.7971299 \cdot 10^{-11}$

Substituting in equation (4) the viscosity μ of air at 293.15 K (68°F) is given as:

μ(293.15) = 1.821e-05 Pa.s = 3.804e-07 lbf.sec/ft^2

2.2.4. Thermal Conductivity

Thermal conductivity k for any gas as a function of temperature is given by a fifth-degree polynomial equation:

$$k(T) = (B_0T^0 + B_1T^1 + B_2T^2 + B_3T^3 + B_4T^4 + B_5T^5)\frac{W}{m.K} \qquad (4)$$

Where: T = Absolute temperature in degree Kelvin.

$B_0, B_1, B_2, B_3, B_4, B_5$ are polynomial coefficients.

For air, the above coefficients are given as:

$B_0 := -2.276501 \cdot 10^{-3}$

$B_1 := 1.2598485 \cdot 10^{-4}$

$B_2 := -1.4815235 \cdot 10^{-7}$

$B_3 := 1.73550646 \cdot 10^{-10}$

$B_4 := -1.066657 \cdot 10^{-13}$

Substituting in equation (3) the conductivity k of air at 293.15 K (68°F) is given as:

k(293.15) = 0.026 W/m.K = 0.015 BTU/hr.ft.R

Polynomial coefficients for transport properties of gases listed in the app suite are taken from the standard reference handbook on heat transfer. For steam and water as per IFC 1967 formulation. Other liquids only viscosity varies as temperatures and appendix 15.2.1. provides viscosity of common petroleum base fluids for reference.

2.3. Transport Properties of Liquids

2.3.1. Density

For incompressible liquids, the density is considered constant for all practical purposes. However, the specific volume changes with respect to temperature due to thermal expansion of the given liquid. Therefore, density of some standard liquid used in industrial applications is provided as a function of temperature.

2.3.2. Specific Heat

Liquid specific heat does not change much with the temperatures. Therefore, this property is treated as a constant. However, specific heat of some standard liquids used in industrial applications is provided as a function of temperature.

2.3.3. Thermal conductivity

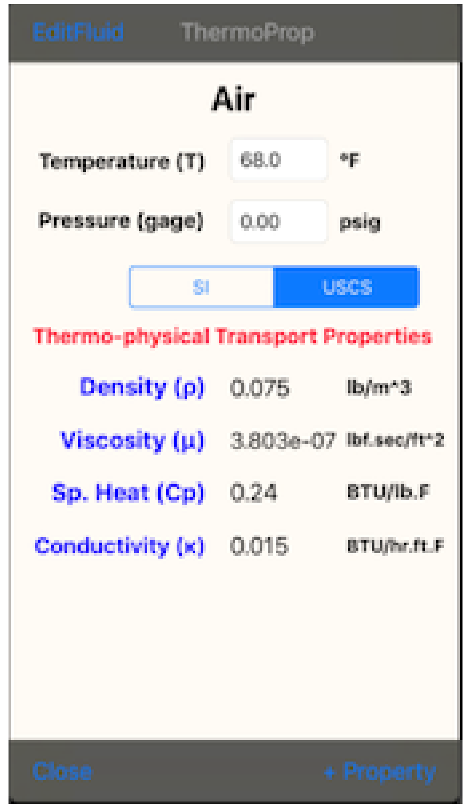
Thermal conductivity also does not change much with respect to temperature. Therefore, this property is treated as constant. However, conductivity of some standard liquids used in industrial applications is provided as a function of temperature.

2.3.4. Viscosity

Viscosity of liquids changes with the temperature and their relationship varies from liquid to liquid. In case of petroleum-based fluids, this relationship considered linear on log-log^2 scale for computation purposes. Viscosity of some standard liquids used in industrial applications is provided as a function of temperature.

iPhone App result – SI

iPhone App result – USCS

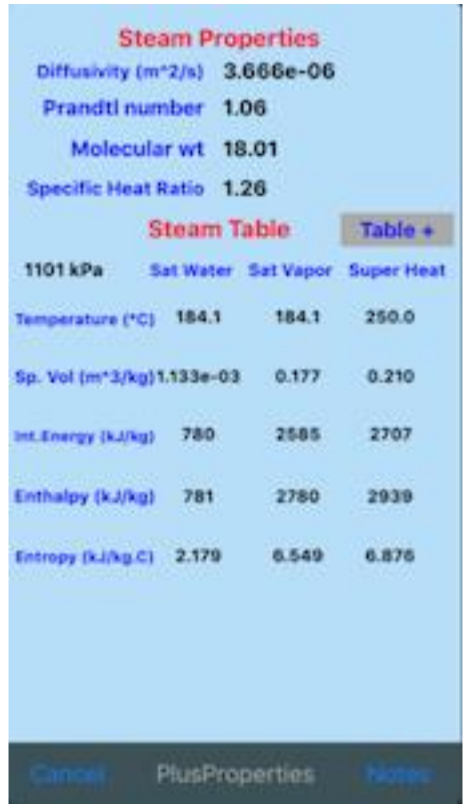


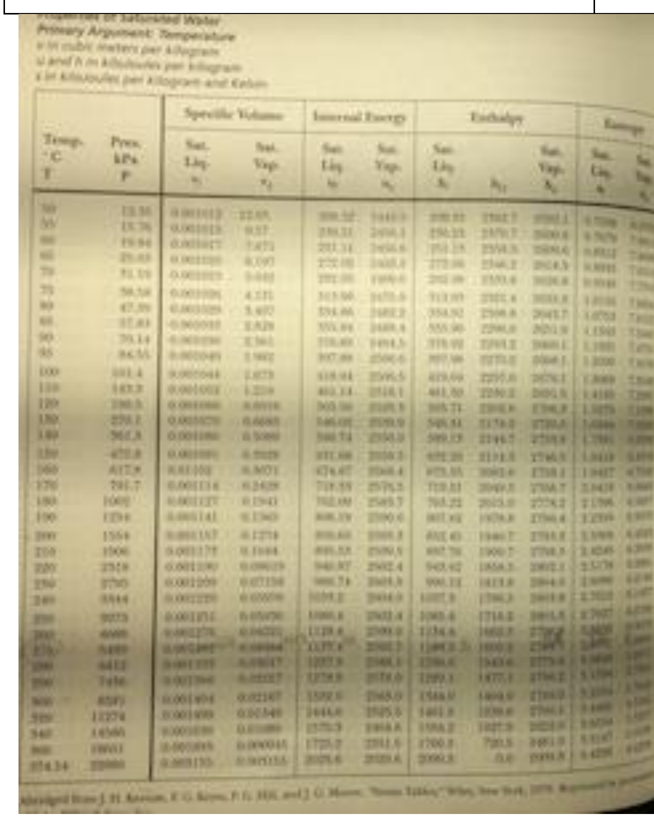
2.4 Steam Table

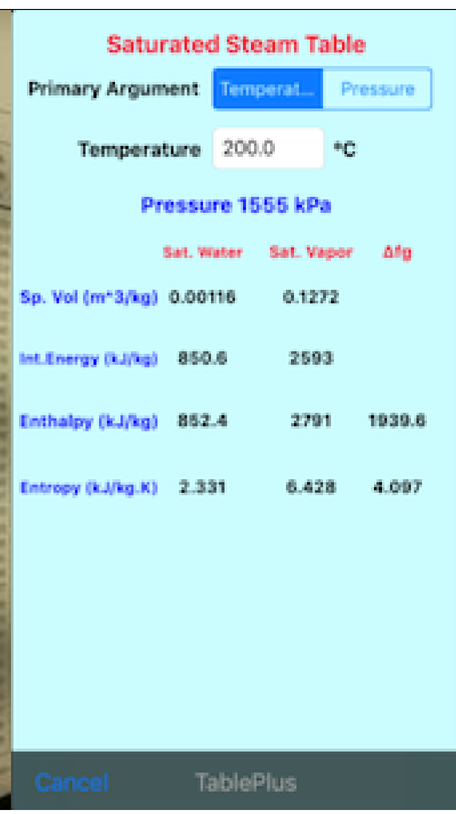
Steam table is referred for all thermodynamic properties as well as transport properties at given temperature and pressure. For saturation condition, the primary argument can be either temperature or pressure while for super heat condition, temperature, and pressure independent of each other. This app serves as a quick reference for steam table that provides both thermodynamic as well as transport properties of vapor at given conditions.

You can enter any condition so long it is with in the sub-region 1 and 2 as per 1967 IFC formulation. This is illustrated in the following example. Please note steam table uses absolute pressure (gage+atm).

Transport Properties of Steam	Steam Table







2.5 Psychrometrics

Psychrometric chart is the reference for all thermodynamic properties of humid air at a given dry bulb and wet bulb temperatures or dry bulb temperature and relative humidity as primary argument. This is illustrated for both SI and USCS system as follows

2.6. Gas flow rate specification

Gas flow rate is specified on either mass flow basis or volume flow at standard conditions (STP – standard temperature and pressure) or normal conditions (NTP – Normal temperature and pressure). For STP, the standard temperature and pressure is specified at 0°C and 1 Bar (100kPa) pressure (14.7 psi) while for NTP, the temperature is specified at 20°C. The units of measurement under both conditions are specified as scfm (standard cubic feet per minute) or slpm (standard liters per minute). Conversion of mass flow rate to standard flow rate is done using the following equation.

$$Q = \frac{m}{\rho}$$

Where:
m = mass flow rate (kg/min or lb/min) ρ = gas density at actual condition
Q = Actual volumetric flow (Acfm or Alpm)

$$Q_{std} = Q \frac{P}{P_{std}} \frac{T_{std}}{T}$$

flw2std = $(P/P_{std})(T_{std}/T)$ (Conversion factor)

Where:
P = Actual gas pressure; T = Actual gas temperature in absolute value (K or R).
Q = Actual volume flow (Acfm or Alpm) from equation.
Q_{std} = Volume flow rate at standard condition.
P_{std} = 1 bar
T_{std} = 273.15 K or 460 R.

Please note standard flow rate conversion from Scfm to Slpm and vice versa is also provided under unit conversion app.

2.7 Dimensionless Flow Parameters

2.7.1. Reynolds Number

Reynolds number is an important design parameter most associated with all fluid flow analysis. This is a ratio of inertia force to viscous force and hence determines whether the flow is inertia dominant or viscous dominant. In other words, if this number is less than 2000 the flow is viscous dominant or laminar. If the number is greater than 4000, the flow is considered inertia dominant or turbulent. This is given by the following equation where all units cancel out and become dimensionless number.

$$R_n = \frac{\rho V D}{\mu} = \frac{VD}{(\frac{\mu}{\rho})} = \frac{VD}{\nu}$$

Where:

ρ = Density of the fluid

μ = Absolute viscosity of the fluid

ν = Kinematic viscosity of the fluid

V= Fluid velocity

D= Flow passage diameter

All fluid properties are determined at the working temperature.

2.7.2. Prandtl's Number

Prandtl-number is yet another important dimensionless number used in all design analysis whenever the flow is associated with heat transfer. It is the ratio of viscosity to thermal diffusivity.

$$\alpha = \frac{k}{\rho . c_p}$$ Thermal diffusivity

$$\Pr = \frac{\nu}{\alpha}$$

Where:

k = Thermal conductivity

ρ = Density

C_p = Specific heat capacity

All fluid properties are determined at the average film temperature. This parameter determines the convective heat transfer coefficient in pipes and conduits.

2.7.3. Nusselt Number

Nusselt number is used in determining heat transfer coefficient under convective fluid flow conditions. It is a function of both Reynolds and Prandtl's numbers and thus forms a third dimensionless number. The Nusselt number is calculated mainly for a conduit flow involving heat transfer across the wall, a most common scenario in fluid flow analysis is given below.

$$Ns = 0.023.\left(Rn^{0.8}\right)\left(\Pr^{0.33}\right) \quad \text{And} \quad h = \frac{Ns.k}{d}$$

Where:

R_n = Reynolds Number

P_r = Prandtl Number

h = Internal heat transfer film coefficient

k = Thermal conductivity of the fluid

d = Tube diameter.

3. Fluid Pressure Drop

This is energy loss involved in transmission of fluids in pipes and conduits for industrial applications. Amount of energy loss depends on type of fluid, pipe schedules and sizes with different material of construction and flow rates. A judicious selection of pipe size is often a tradeoff between initial cost and subsequent energy cost involved in transmission of fluid from one point to another. Therefore, this app can be a helpful tool for practicing engineers and designers in analyzing different matrix.

Pipe Schedules:

10, 20, 30, 40, 60, 80, 100, 120, 140, 160.

Users can pick applicable sizes under nominal diameter for which app provides the corresponding inside and outside diameter.

Material of construction:

Steel, SS, GI, CI, Concrete, Copper, Aluminum, PVC, Glass, Wood, FRP, Rubber.

Based on the size, length and material of construction, the app gives the total weight of the pipe for trade-off analysis.

3.1. Darcy-Weisbach Equation

Pressure drop in pipes and tubes are primarily due to the friction between the fluid and surface of the pipe. Further, coefficient of friction (f) depends upon the surface roughness (ε/D) as well as the degree of turbulence (Reynolds number) present in the flow. The Moody diagram provides this relationship to compute the pipe friction under both laminar and turbulent condition. The Darcy's formula for pressure-drop in pipes and conduits of diameter ‘D’ length ‘L’ are given as follows (Eq.10).

$$\Delta P = f\,\frac{L}{D}\frac{\rho V^2}{2}$$

Where:

V = Liquid velocity

ρ = Liquid density

f = coefficient of friction - f (Rn, e/D)

Friction Factor:

In Darcy’s equation the coefficient of friction is dimensionless quantity, which is a function of pipe surface geometry and Reynolds number.
If the flow is laminar (Rn < 2000) the friction factor is computed as follows.

f = (64/Rn)

For Turbulent conditions determined using Moody diagram (Fig 15.2.2) as a function of relative roughness (e/D) the roughness of the pipe wall as compared to the pipe diameter. Generally, the pipe roughness for different material of construction is known and can be used for determining the friction factor. The app suite uses this standard value and has algorithm to compute this factor for any schedule pipes. For smooth pipes such as one used in hydraulic applications friction factor is also determined using the following equation.

f = (64/Rn^0.25)

Darcy's equation is valid for all liquids as well as gases. However, for gases, it is valid conditionally if the flow is isothermal and velocity is less than 40% of its sonic velocity. This referred in section 3.3 Isothermal flow.

3.2 Flow Through Valves and Fittings

Resistance Factor (K)

Fluid circuits in an industrial environment consist of number of different size pipes, valves, and fittings. They all resist flow resulting in drop in pressure energy. While the Darcy's equation for pressure loss due to friction is valid for pipes, the experimental data reveals that the pressure loss in valves and fitting due to friction is small. Instead, the pressure loss is directly proportional to its velocity head. The constant of proportionality is defined as resistance factor K. Therefore, it depends only on the given type and size of valve or fitting and independent of friction factor or Reynolds number. This is expressed in the following equation.

$$\Delta p = K \cdot \frac{\rho \cdot V^2}{2}$$

Where:

V = fluid velocity

ρ = fluid density

K = Resistance factor

The resistance coefficient K is theoretically constant for the given type and geometrically similar design of valves and fittings. Generally, the manufacturers provide the values for K factors or else it can be adopted from the experimentally determined values on identical design. App suite provides these numbers for most common type of valves and fittings that can be selected based on the actual condition.

In Darcy's equation for pipes and conduits, the resistance factor K represents the dimensionless quantity $f\frac{L}{D}$ as given below.

$$\Delta P = \left(f\frac{L}{D}\right)\frac{\rho V^2}{2} = K \cdot \frac{\rho V^2}{2}$$

This concept is especially useful in analyzing the total resistance in a fluid circuit consists of different pipes, valves, and fitting there by summing up the individual resistances or K factors. It also gives the equivalent length of any fluid circuit, which is given below.

$K_{total} = \Sigma\, K_{pipe} + \Sigma\, K_{valve} + \Sigma\, K_{fitting}$

$$L_{equ} = K_{total} \cdot \frac{D}{f}$$

The equivalent length is a single length of a pipe diameter D that would result into same pressure loss as its components in a circuit.

3.3 Isothermal and Isentropic Flow

3.3.1 Isothermal flow

Isothermal flow of compressible fluid in a conduit occurs if heat transferred out of the fluid and energy converted in to heat by friction off set each other so that the fluid temperature remains constant. Thus, compressible flow in a long conduit can be analyzed as isothermal flow and given as:

$$p_1^2 - p_2^2 = \frac{m^2 RT}{A^2}\left[f\frac{L}{D} + 2\ln\frac{p_1}{p_2}\right]$$

Where:

p_1 = pressure at point 1

P_2 = pressure at point 2

m = mass flow rate

R = specific gas constant

T = Absolute temperature of the Gas

g = acceleration due to gravity

A = pipe cross sectional area

f = coefficient of pipe friction

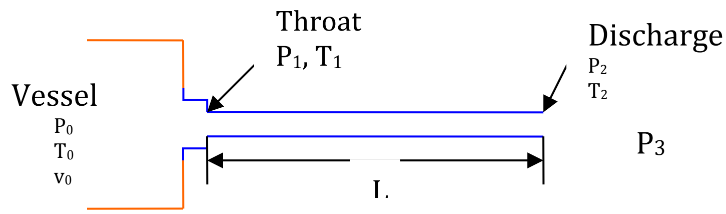
L = Pipe length

D = Pipe diameter

The above equation is used for gases where Darcy's equation cannot be applied.

3.3.2 Isentropic flow

Isentropic meaning constant entropy or there is no change in entropy. Hence, no addition of heat due to friction and the flow occurs fast in relatively short distances. Let us take an example of high-pressure steam or gas discharging through pipe from a large chamber or vessel as shown below.



Since Reynolds numbers for adiabatic flow are very high, the friction coefficient for the pipe generally constant. If pressure ratio P_0/P_3 is greater than the critical pressure ratio P_c given by equation, then the flow is choked flow, exit velocity reaches its sonic velocity and P_2 is independent of P_3. The mass flow rate under chocked flow condition as function of exit velocity, density and cross section area is given below.

$$P_C := \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}$$

$$Sonic\ velocity(v) = \sqrt{k \cdot R_{gas} \cdot T_2}$$

$$mass\ flow\ rate = \rho_2 \cdot A_2 \cdot v$$

$$T_2 = T_0 \cdot \left(\frac{2}{k+1}\right)$$

$$\frac{P_2}{P_0} = P_c \cdot \frac{G}{G_c}$$

Where P_c – Critical pressure ratio as per equation

G – Actual mass flow of the gas

G_c- Maximum mass flow under ideal isentropic nozzle flow

k – Specific heat ratio of the gas

R_{gas}- Specific gas constant (Universal Gas constant/MW)

The actual mass flow will vary depending upon the pipe resistance factor K discussed in section 3.2.

The conditions at the throat is given by the following equations

$$\frac{T_1}{T_0} = \left(\frac{P_1}{P_0}\right)^{\frac{k}{k-1}} \qquad \frac{P_0}{P_1} = \left[1 + \frac{G^2}{2}\left(\frac{k-1}{k}\right)\frac{R_{gas}T_1}{P_1^2}\right]^{\frac{k}{k-1}}$$

Therefore, pipe pressure drop under isentropic condition will be $\Delta P = P_2 - P_1$

3.4 Case Examples

3.4.1. Water at 100 kPa and 20° C flows through 2" schedule 40 pipe at a rate of 354 liter/minute. Calculate the pressure drop for100' steel pipe. Water properties at 100 kPa, 20° C is taken from Thermo-Physical app.

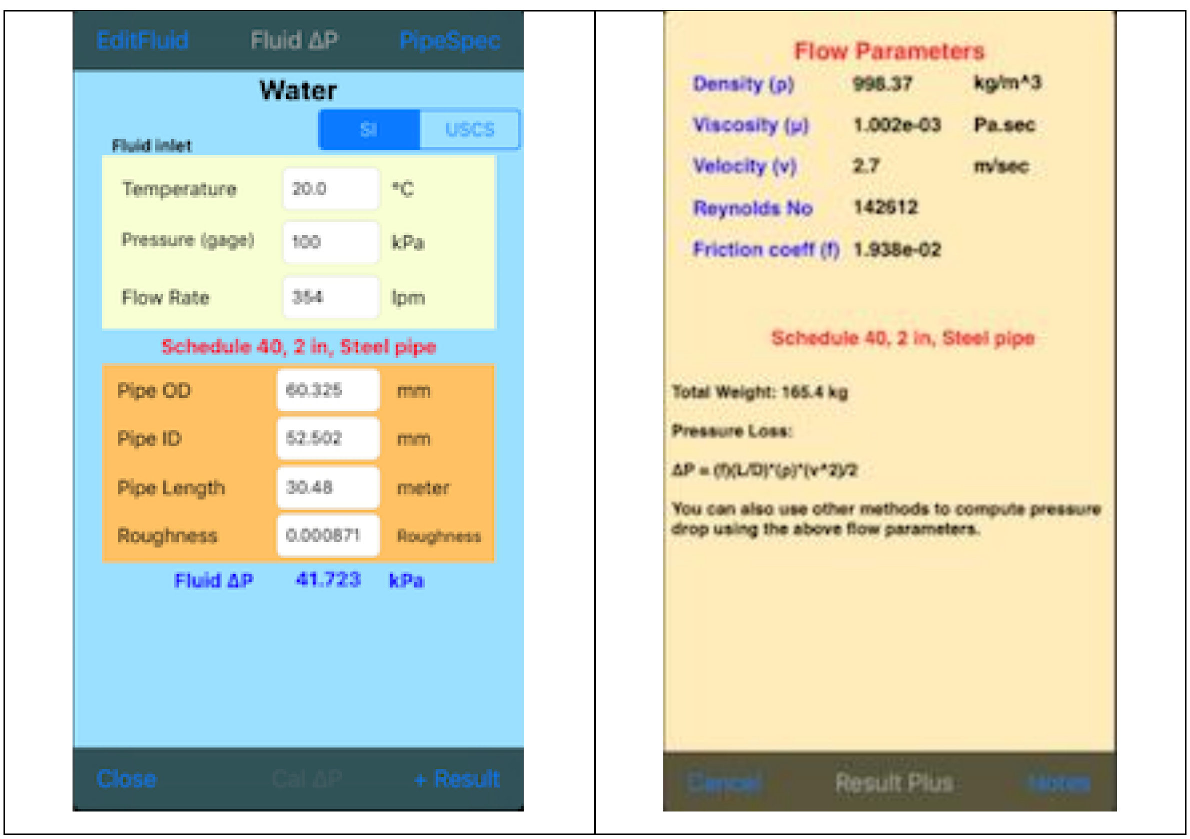
A. Density: 998.4 kg/m^3
B. Viscosity: 1.002e-02 Pa.sec
C. 2" 40sch pipe inner diameter D: 2.067" (52.502mm)
D. Pipe length: 100' (30.48 meter)
E. Roughness factor for steel (e/D): 0.000872
F. Q = 354 lpm

$V = Q/A = (354e\text{-}03 m^3)/(60.0s(0.25*\pi*52.5e\text{-}03\ m^2)) = 2.725\ m/s$

$\nu = \mu/\rho = 1.002/998.0 = 1.004$ centistokes.

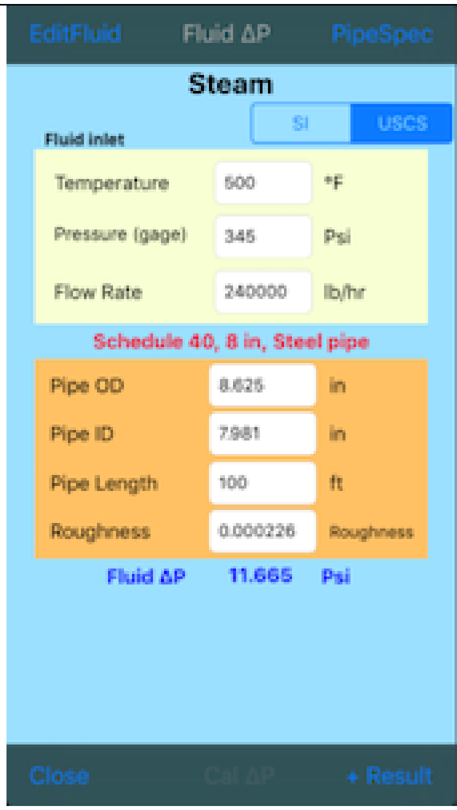
$Rn = V*D/\nu = 1.426e{+}05$ from Moody diagram f = 0.019 (Fig 15.2.1)

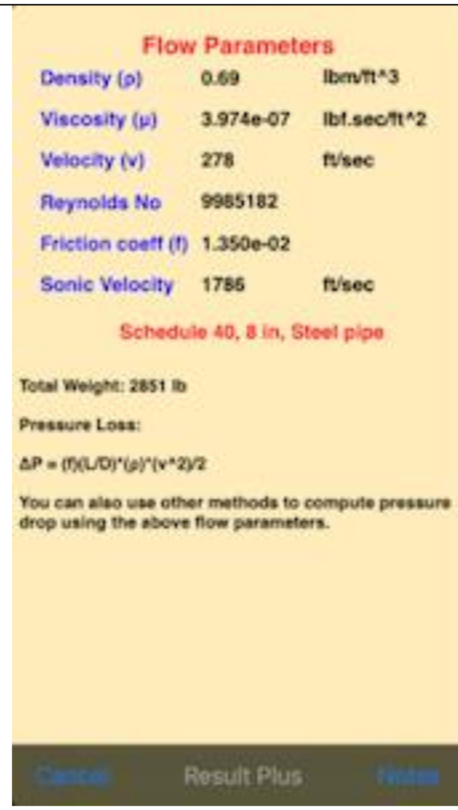
$\Delta p = f(L/D)(\rho V^2/2) = 0.019*(30.48/0.0525)*(998*2.725^2/2) = 41.7$ kPa



3.4.2 Steam at 345 psig and 500°F flowing through 8" schedule 40 pipe at rate of 240,000 lb/hr. Determine pressure drop per 100' of pipe.

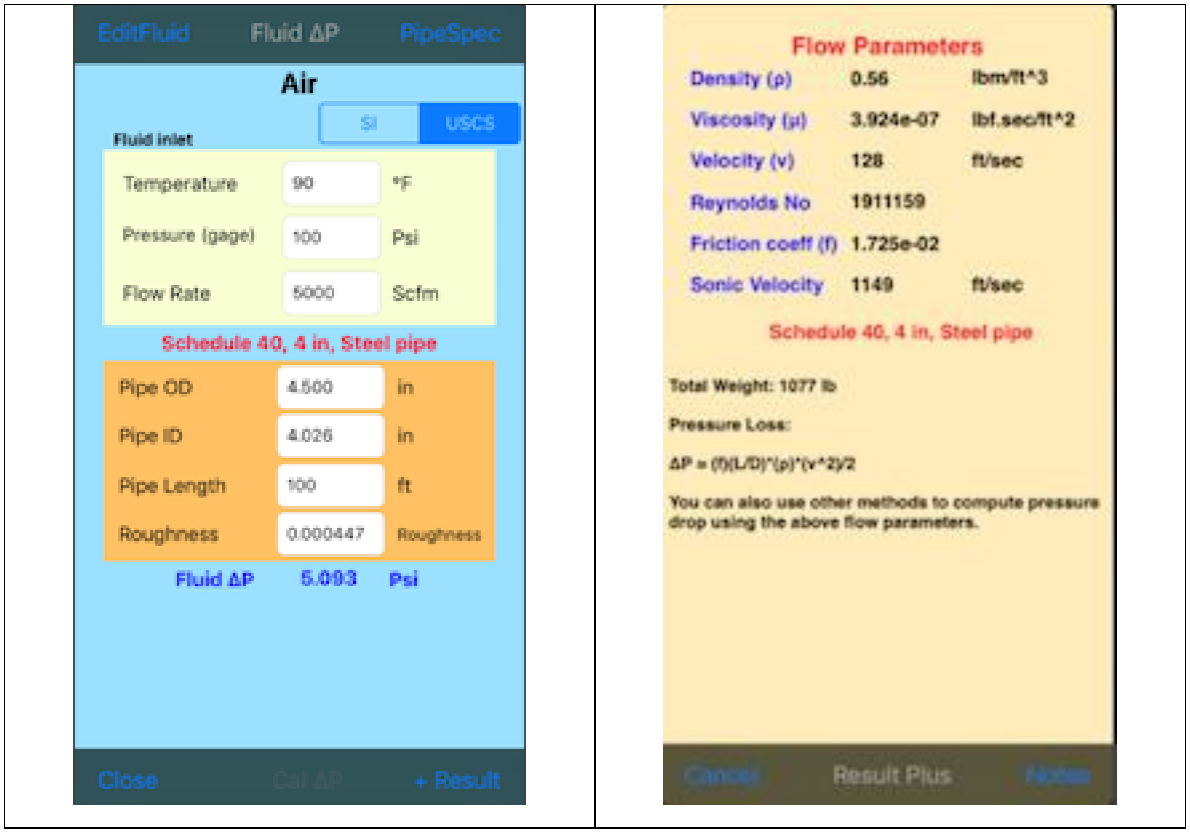
$$\Delta p = f(L/D)(\rho V^2/2)$$





3.4.3 Pressure drop is 5psi with 100-psig air at 90°F flowing through 100' 4" schedule 40 pipe. Determine the flow rate in Scfm (Standard cubic feet per minute).

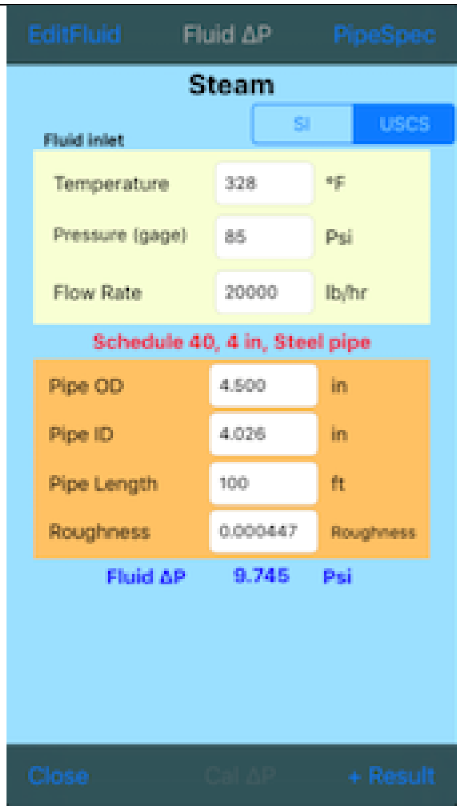
In this example flow rate is not given hence try interactively by choosing an arbitrary number until get the desired pressure drop. Accordingly, for 5000 Scfm the pressure drop is 5 psi as shown.

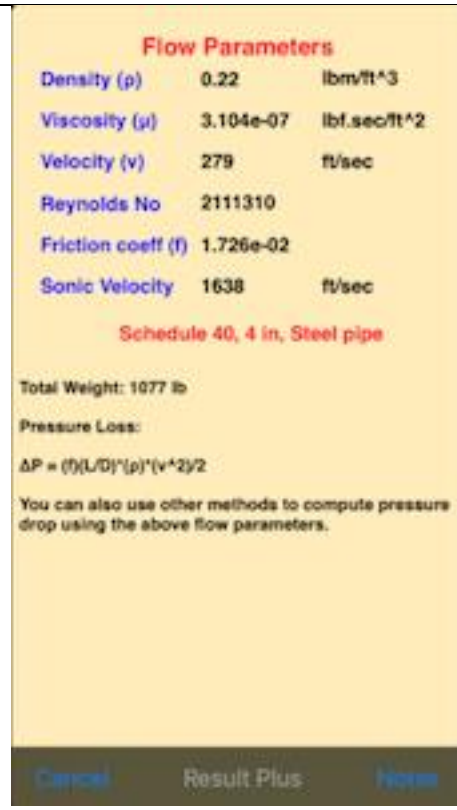


3.4.4. An 85 psig saturated steam line with 20,000 pounds per hour flow is permitted a maximum of 10 psi per 100' feet of pipe. Determine the smallest size of Schedule 40 pipe suitable.

Solution:

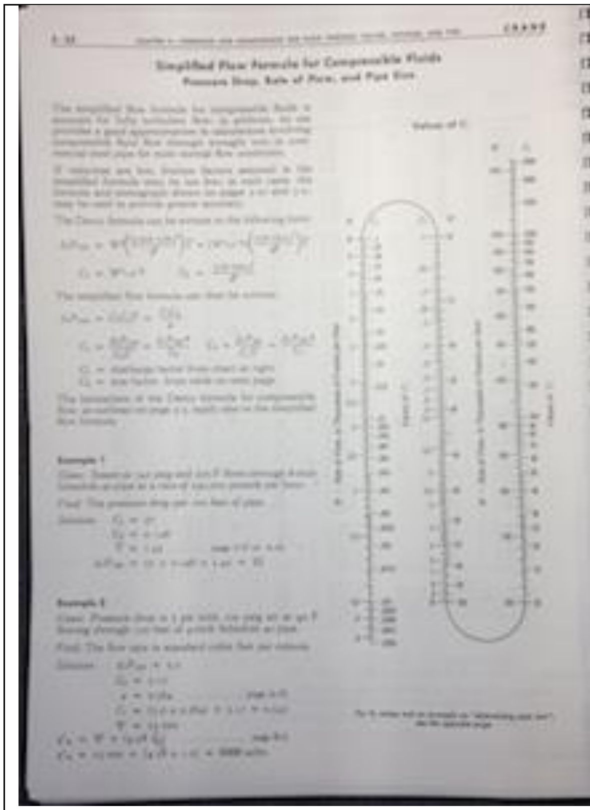
This is a typical design problem frequently encountered in process industries. Once again this is solved iteratively by choosing bigger size say 6" and then gradually reduce until pressure drop is less than 10 psi. Accordingly, for 4" Schedule 40 the pressure drops found to be 9.74 psi.

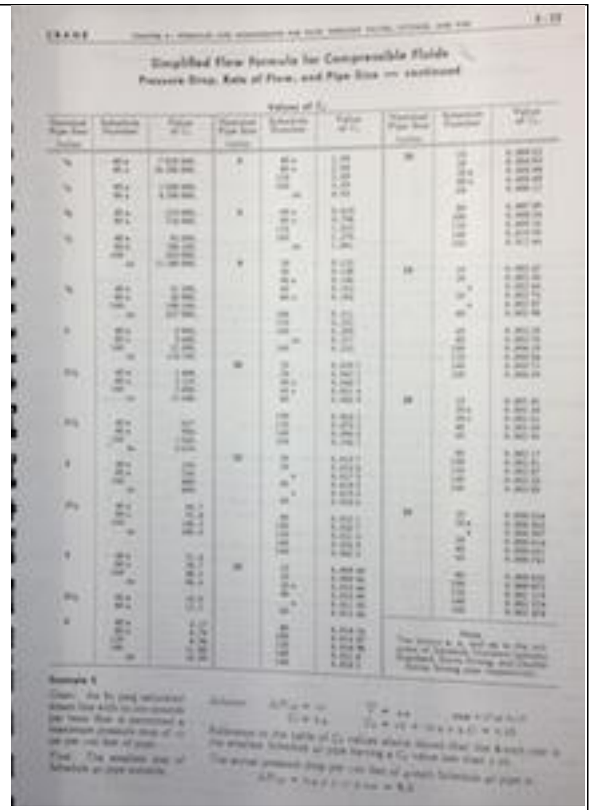




The above three examples were chosen from chapter 3 of “Formulas and Nomogram for Flow Through Valves, Fittings, And Pipe.” published by Crane Engineering company. Most practicing engineers in US used this very extensively prior to PC’s were invented in 80’s and now we can use smart phone apps like one presented in this Manual.

The above example page is shown in the following snapshot that also provides the solutions and results that are consistent with App suite.





In above example 1 of Crane Engineering the pressure drop for 100 feet of pipe is 12 psi. The App suite result shown in 3.4.2 is 11.67 psi that can be rounded off to 12 psi.

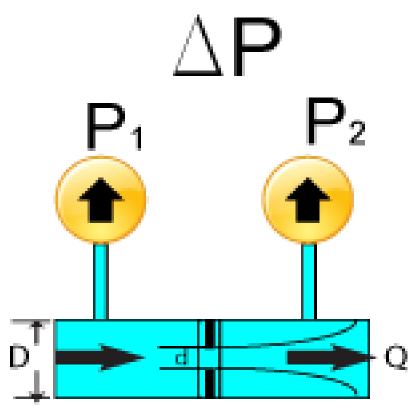
In example 2 of Crane Engineering flow rate computed for 5 psi drop as 5000 Scfm. The app suite result shown in 3.4.3 as 5 psi pressure drop for 5000 Scfm flow rate of air passing through 4” Schedule 40 steel pipe.

In example 3 of Crane Engineering the steam pressure drop for 4” Schedule 40 is calculated as 9.3 psi. The app suite result is shown in 3.4.4 as 9.745 psi steam pressure drop for the same pipe specification. (Please refer Appendix 15.2.4, 15.2.5 for details)

4. Measurement of Flow Through Orifices

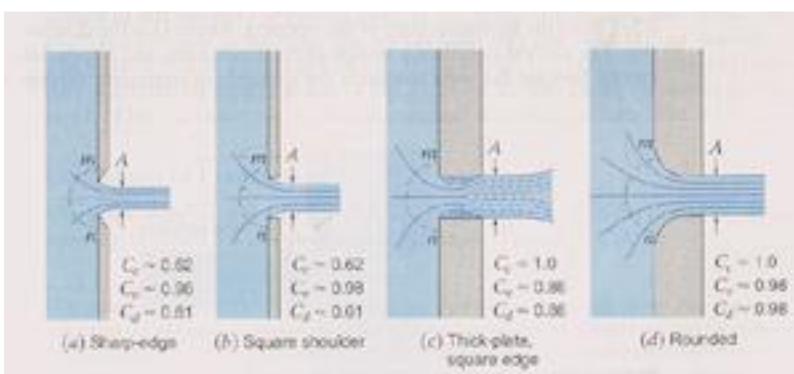
4.1 Orifice Design Analysis Model

Measurement of fluid flow in pipes and conduits using orifice plates are quite common in industrial applications. Here is a generic design analysis model using standard pressure taps as per ISO 5167 for measurement of pressure differential across orifice plates. Fluid properties are taken for the given upstream pressure Pu and temperature Tu for computation of flow rates Q.



Orifice Type

There are four basic orifice designs as shown below with recommended discharge coefficients based on the orifice geometry.



However, users can edit this using manufacturer's data for more accurate results.

4.2. Flow Equations

4.2.1 Liquid flow through orifice

The most general equation for flow through any orifice is of the form:

$$Q = C_d A \sqrt{\frac{2}{\rho} \Delta P}$$

Or

$$\Delta P = \left(\frac{Q}{C_d A} \right)^2 \frac{\rho}{2}$$

Where:

C_d = Orifice discharge coefficient (Cc * Cv)

ρ = Liquid Density

ΔP = Pressure differential or drop

Q = Discharge rate or flow rate

The discharge coefficient primarily depends upon the geometry of the orifice and can vary anywhere from 0.61 for a sharp edge orifice to 1.0 for a long smooth nozzle or venturi. The manufacturer normally provides this data, and it is conveniently used in the above equation. However, the users can also control the orifice size and the pressure differential that must be set to achieve the desired flow rate.

4.2.2 Gas flow through orifice

The mass flow rate through any orifice is determined based on the pressure ratio of upstream to downstream. The mass flow rate becomes constant irrespective of increased in upstream pressure when this ratio reaches a critical value. This flow condition is known as choked flow. The critical pressure ratio is a function of specific heat ratio and given by the following equation.

$$P_c := \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \tag{18}$$

Where:

k = specific heat ratio (c_p/c_v) and for air it is 1.4

Substituting this value, the critical pressure ratio for air is 1.893.

The mass flow rate is given by the following equation.

$$m_{orf} = C_d A_{orf} K_{orf} n_{orf} \frac{P_u}{\sqrt{T_u}}$$

Where:

Cd = Orifice discharge coefficient.

A_{orf} = Orifice area

$$K_{orf} = \sqrt{\frac{k}{R}\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}}$$

$$n_{orf} := \sqrt{\frac{\left(\frac{P_d}{P_u}\right)^{\frac{2}{k}} - \left(\frac{P_d}{P_u}\right)^{\frac{k+1}{k}}}{\left(\frac{k-1}{2}\right)\cdot\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}}}$$

P_u = Absolute pressure – upstream.

For choked condition this is always Pu = P_d*P_c

T_u = Absolute temperature – upstream

P_d = Absolute pressure – down stream

K = specific heat ratio

R = specific gas constant

Value of n_{orf} can be obtained from gas tables for air and for some other gases based on the pressure ratio. For choked condition, this value is always taken as 1.

4.3. Orifice Flow Examples

4.3.1 Water at 20 C flows through 2" Schedule 40 pipe is fitted with 5/8" sharp edge orifice plate with 100 kPa pressure differentials. Determine discharge flow rates.

$$Q = C_d A \sqrt{\frac{2}{\rho} \Delta P}$$

C_d = 0.61
D = 52.5 mm (2" Schedule 40) = 5.25 cm
d = 1.5875 cm (5/8" diameter sharp edge orifice)
A = 0.25 π d^2 = 1.98 cm^2

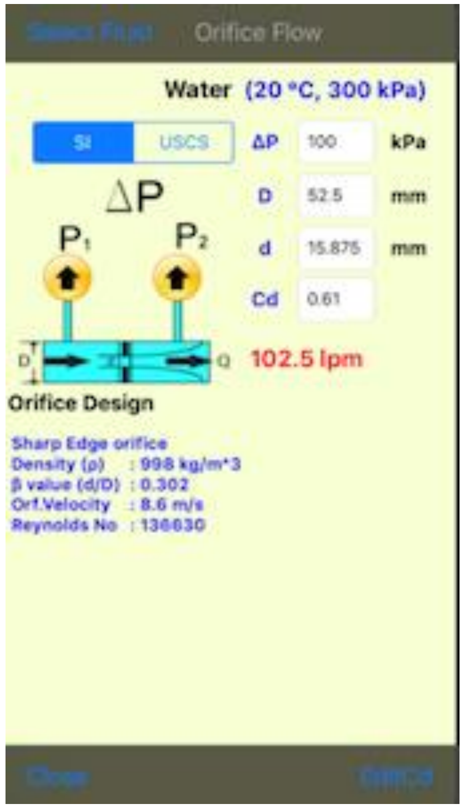
ρ = 0.998 gm/cc

ΔP = 100 kPa = 1.02 kgf/cm^2 = 1,02*981*kg/cm.sec^2

V = $\sqrt{2\Delta P/\rho}$ = $\sqrt{2}$*(1.02*981kg/cm.sec^2)÷0.998e-03kg/cm^3)

V = 1415 cm/sec

Q = 0.61*A*V = 1708 cc/sec = 102.5 liter/min

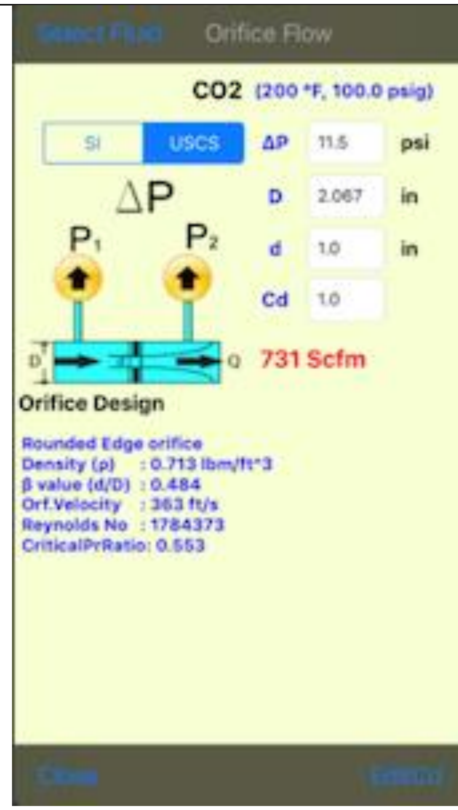


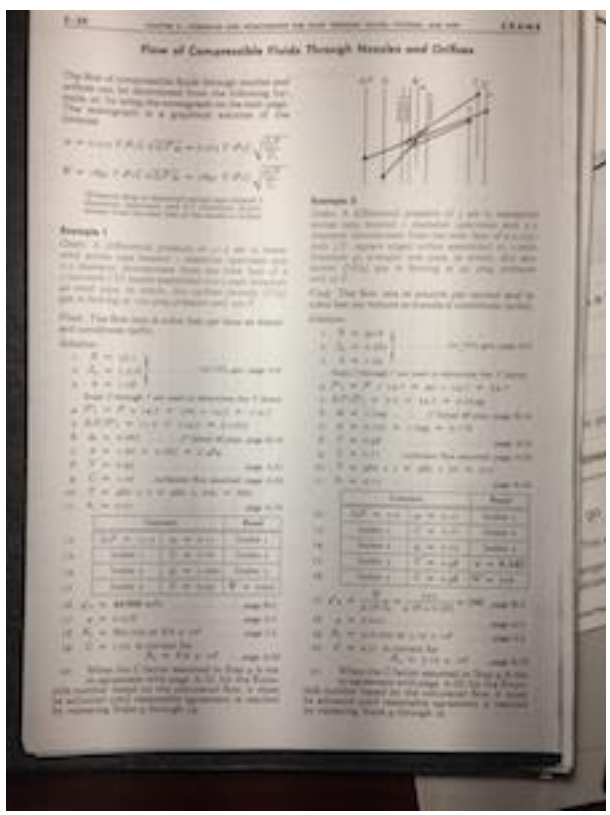
Please note that orifice velocity used for computing Reynolds number is the pipe velocity to determine type of flow (Laminar/Turbulent).

4.3. 2 A differential pressure of 11.5 psi is measured across taps located across 1-inch nozzle (same as rounded orifice) assembled in 2 inch Schedule 40 steel pipe, in which dry carbon dioxide (CO2) gas is flowing at 100 psig pressure and 200°F. Determine the flow rate in standard cubic feet per minute.

This example is taken from Crane Engineering for compressible fluids through Nozzles and Orifices.

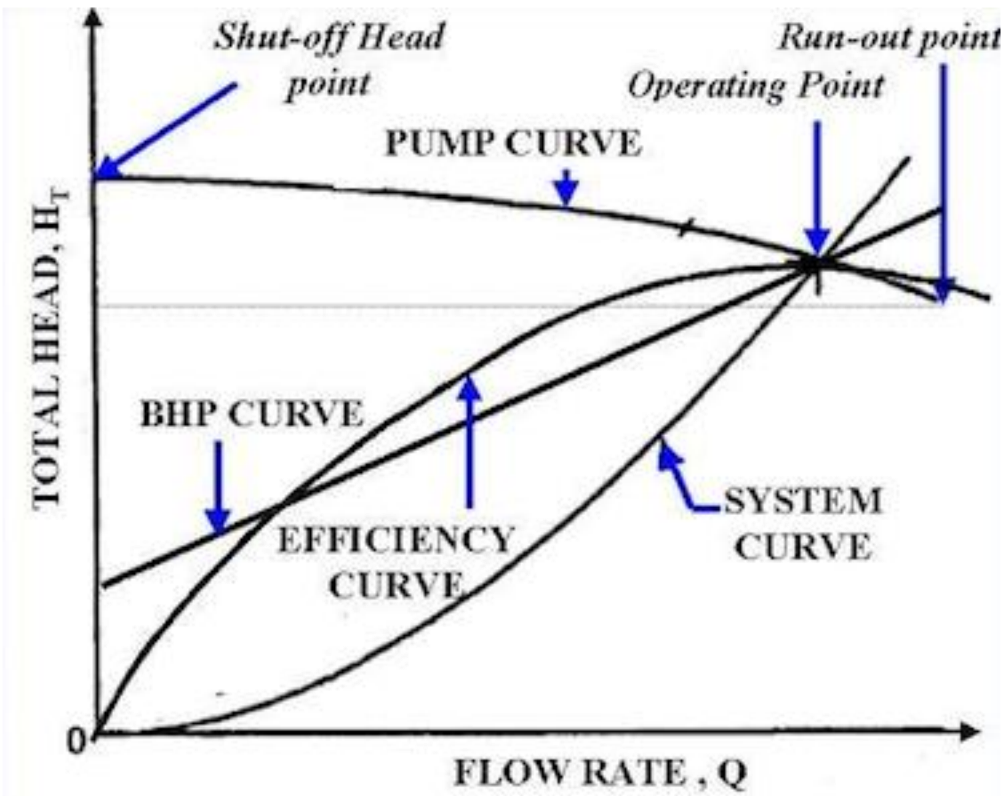
As per Crane (Fig 15.2.6) example flow rate computed as 44000 scfh (standard cubic feet per hour) i.e. 733.3 scfm results consistent with App suite as shown.




Flow of Compressible Fluids Through Nozzles and Orifices

5. Centrifugal Pump Specification

Centrifugal pumps are energy transferring devices mainly used for transporting liquids in domestic as well as industrial applications. Pumps are normally specified by flow rates (Q) delivered against given head (H). A typical performance characteristic of a centrifugal pump is as shown below. The drooping characteristics QH curve when cuts the system head curve gives us the operating point of a centrifugal pump.



Manufacturers provide these performance characteristics for different impeller diameter as well so that a most suitable pump is employed for any given application. App suite covers mainly single stage pumps that are commonly used in industrial applications. Multistage pumps are used where high head is required such as boiler feed water pump.

5.1. Design Specifications

There are three key factors in a centrifugal pump design specification such as 'System Head', 'NPSHA' and 'Sizing' requires attention in order to unveil the potential for energy savings. A judicious selection of pump is very essential for its energy efficiency and long-term satisfactory performance.

System Head

System Head is the total dynamic head or resistance offered by the system where pumping fluid required overcoming while transporting from reservoir to the destination. Technically it is the total pressure drop or head for the delivery line accounted for all pipes, valves and pipefittings that were discussed in section 3. A general equation using Darcy's formula is given below to determine pressure drop.

$\Delta P = f\frac{L}{D}\frac{\rho V^2}{2}$ For pipe pressure drop computation.

$\Delta p = K \cdot \frac{\rho \cdot V^2}{2}$ For valves and valve fitting pressure drops.

System head (dynamic head) = Static head + total pressure drop

NPSHA

Net positive section head available to charge the pump with desired flow rates without causing any cavitation damage. Each manufacturer provides this information for their pump and one needs to determine every time when the reservoir condition gets changes or altered. This covers suction lift, vapor pressure and pipe friction heads.

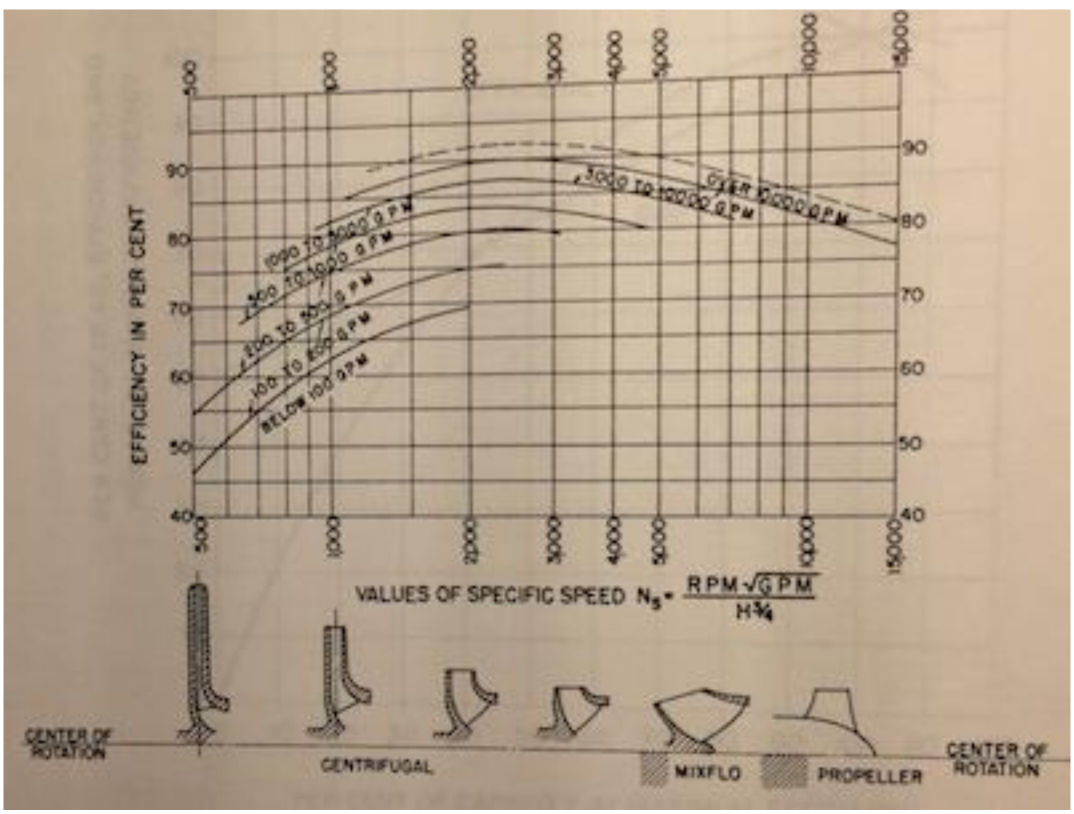
Sizing

The impeller diameter is often chosen based on maximum theoretical head and the impeller speed that depends upon the type of prime mover or an electric motor. The drooping characteristics of the pump determine the pump flow rate at this head. Yet another factor is the width of the impeller that is based on the speed. Manufacturers provide range of impellers that can fit in to the impeller casing. Thus, design and performance greatly depended upon relationship between the operating speeds, flow rate and required total head. This is combined into a single factor and determined as specific speed Ns and given as follows along with Ns vs. Efficiency for type of impeller.

Specific Speed Ns = RPM*√Q / H^0.75

Q is given in GPM for USCS and m^3/sec in SI

H is given in feet for USCS and meter in SI.

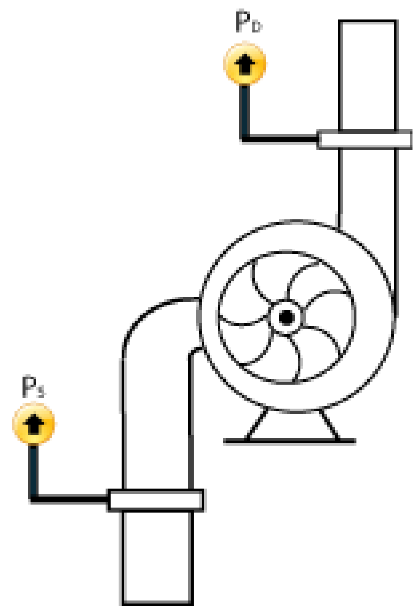


5.2. Total Head and NPSHA

Total head is also sometime referred, as system head is the energy transferred to the liquid by the pump, that is difference between the discharge head and suction head/lift. It is given as follows.

$$H_{total} = h_d - h_s$$

It can also be determined using the static head from supply level to discharge level, plus all the friction head due to pipes, valves and fittings. This is some time also can be measured by the gages attached to discharge and suction side as shown below.



The gage readings should be corrected for the pump centerline and the velocity head at the gage point. It is recommended to follow the test code of the hydraulic institute.

Pump manufacturers specify net positive suction head required to push the liquid in to pumping chambers. If the available head is less than required head specified cavitation occurs and pump performance gets affected very badly and sometime results into cavitation failure. Hence care should be taken to determine this especially for suction lift where chances are more for cavitation damage. This is given as follows.

NPSHA = ha – (hs + hf+ hvp + hms + hv);

Where:

ha – Atmospheric pressure (101.325kPa/ρg) 10.3 meter

hs – Static head from liquid kevel to pump centerline

hf - Head required to overcome pipe friction

hvp – Head due to vapor pressure

hms – Head due to minor losses (Valves and fittings)

hv – Velocity head (u^2/2g)

5.3. Case Examples

5.3.1 Furnace Oil #2 is pumped from ground storage to day tank located about 10 m from the ground. App suite analysis of the pump with specification data is as shown below.

Furnace Oil #2 Properties:

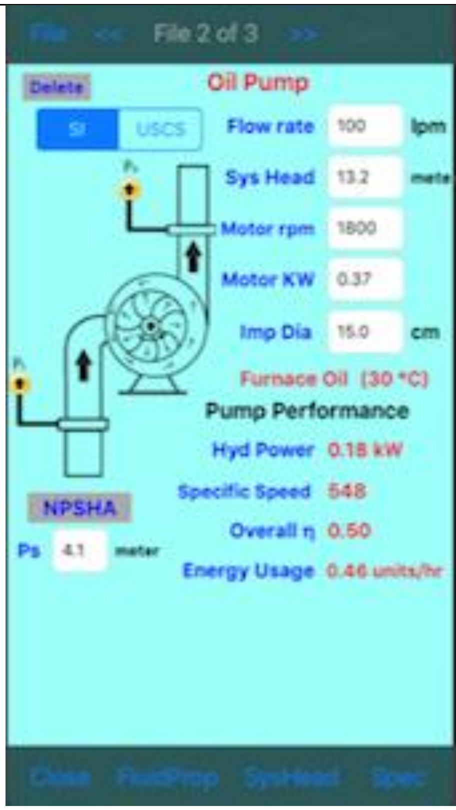
1. Specific Gravity: 0.85
2. Viscosity Cp @100°F: 2.7 (3.5-03 Pa.sec @ 30°C)

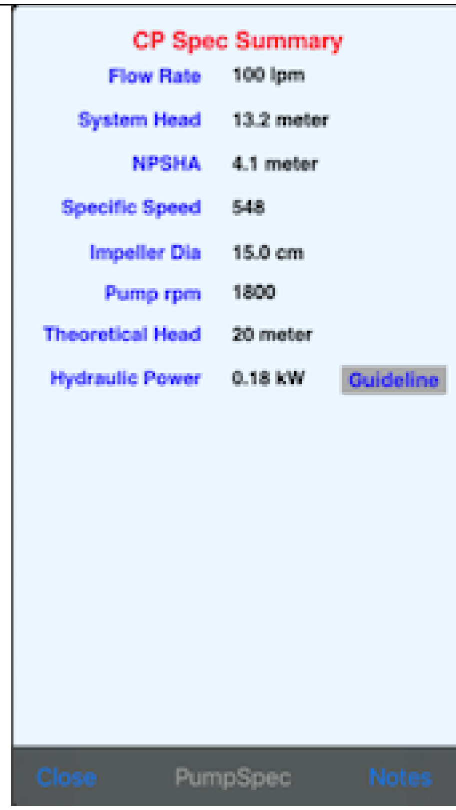
System Head:

H = Hs + Hd = 10.0 + 3.2 = 10.32 m

Motor: 0.5 HP/1800 rpm

Impeller size: 6 in (15 cm)


File 2 of 3
Delete
Oil Pump
SI
USCS
Flow rate 100 lpm
Sys Head 13.2
Motor rpm 1800
Motor KW 0.37
Imp Dia 15.0 cm
Furnace Oil (30 °C)
Pump Performance
Hyd Power 0.18 kW
Specific Speed 548
Overall η 0.50
Energy Usage 0.46 units/hr
NPSHA
Ps 4.1 meter


CP Spec Summary
Flow Rate 100 lpm
System Head 13.2 meter
NPSHA 4.1 meter
Specific Speed 548
Impeller Dia 15.0 cm
Pump rpm 1800
Theoretical Head 20 meter
Hydraulic Power 0.18 kW
Guideline
Close
PumpSpec
Notes


Furnace Oil
Temperature 30.00 °C
Vapor Pressure 2.34 kPa
Density 850.00 kg/m^3
Viscosity 3.500e-03 Pa.sec
Water
Custom
Notes:
For water, just edit the temperature data and other property data gets updated as per IFC 1967 for saturated liquid conditions.
If you choose custom, please edit all property data including vapor pressure at given temperature.
By default, most pump manufacturers give their pump performance (P-Q) curve that is generally valid for other liquids. However, NPSH requirement may vary that depends upon the vapor pressure of the given liquid.
PumpingLiquid
Cancel


(30 °C) Furnace Oil at 100 lpm
Static Head 10 meter
Component - Dia
Length / Qty
K-factor
ΔP (kPa)
Sch40 pipe 1.5 in 50 m 25.79
Gate valve 1.5 in 1 0.20 0.14
Bend-90 1.5 in 2 0.30 0.21
Elbow 1.5 in 4 0.61 0.42
System Head 13.2 meter
SystemHead

5.3.2 Water pump supplies water from reservoir at 90°F to the boiler house @50 gallons per minutes. App suite analysis of the pump with specification data is shown below.

Water Properties:

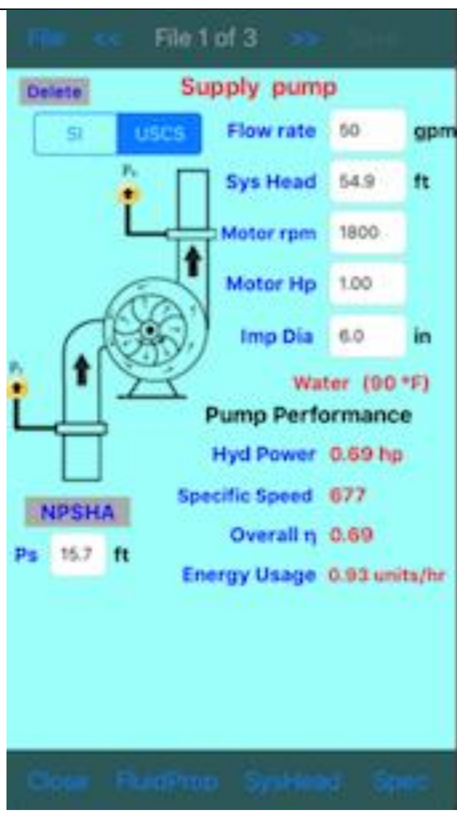
1. Density: 62.12 lbm/ft^3
2. Viscosity: 1.589 lbf.sec/ft^2
3. Vapor pressure: 0.70 psi

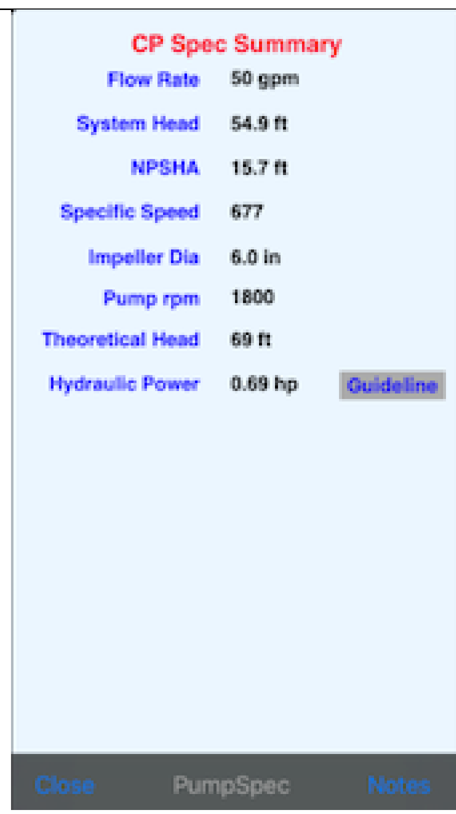
System Head:

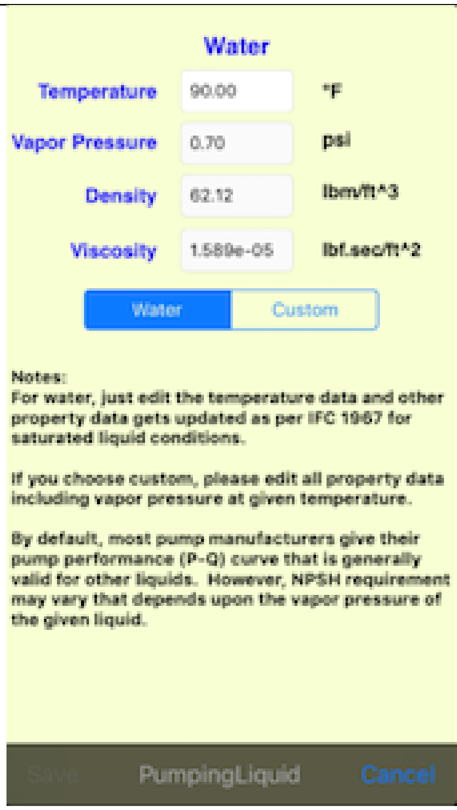
H = Hs + Hd = 33' + 31.9'(8.46 psi) = 54.9'

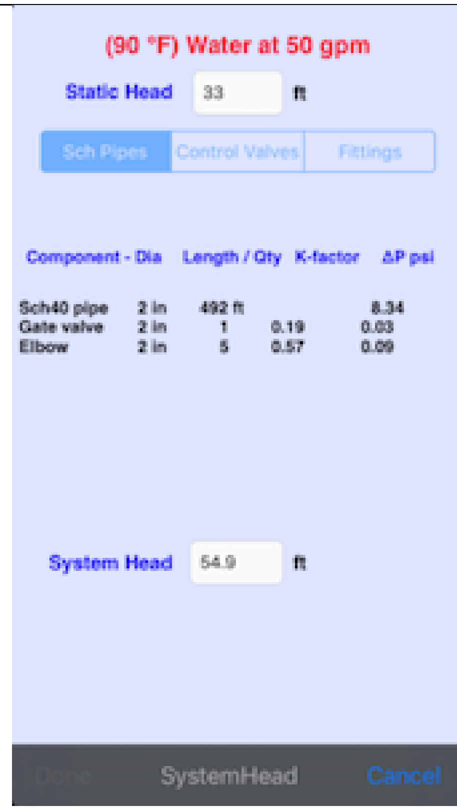
Motor: 1.0 hp/ 1800 rpm

Impeller: 6-inch diameter.









5.3. Centrifugal Pump Maintenance

5.3.1 Performance monitoring

Performance monitoring is part of routine maintenance to assess any degradation that needs immediate attention. Operational parameters such as pressure, flow and temperatures are often recorded or logged for process management in industrial applications also can be used for charting optimal or sub-optimal conditions including pump efficiency. Pump that is on continuous duty also needs to be assured for its operational reliability and structural integrity. This calls for to perform periodic checks and visual inspection of the pump while under running condition weekly or fortnightly depending upon the type of liquid or chemicals being handled by the pump and its severity and hazardous operating condition. A general checklist is given below.

1. Any remarkable changes in noise level or physical condition.
2. Visual checks for seal integrity and leakage.
3. Bearing housing temperature conditions or abnormal sound.
4. Measure vibration levels at the bearing housings of both motor and pump for overall structural integrity. Compare values with permissible standards such as ISO 2372 for general machinery.
5. Any excess vibration level needs further investigation using vibration analyzers that identify the source such as rotor unbalance, mechanical looseness or coupling misalignment etc.
6. In case of belt drives perform stroboscopic examination that detect any excessive slip or slackness.

5.3.2 Cleaning and Lubrication

Pump manufacturers often recommends periodicity of maintenance for a normal operating condition. Suppose the pump is operating in an adverse condition such as high temperature or dusty environment, periodicity need to be increased to combat the damage caused by the adverse operating conditions. Regular cleaning and lubrication are one of the most important preventive maintenance that can slowdown wear and tear of components that are under relative motion such as in gland box. Bearings are either grease or oil lubricated that needs to be applied based on the severity of operating conditions.

5.3.3 Wear ring, rotor, and volute casing inspection.

This is part of shutdown maintenance performed mostly during annual turn around and checked for any corrosion damage, excessive erosive wear on impeller blades, rotor balance and wear ring condition. Similarly suction and delivery lines also inspected for renewals.

6. Boiler Fuel Efficiency

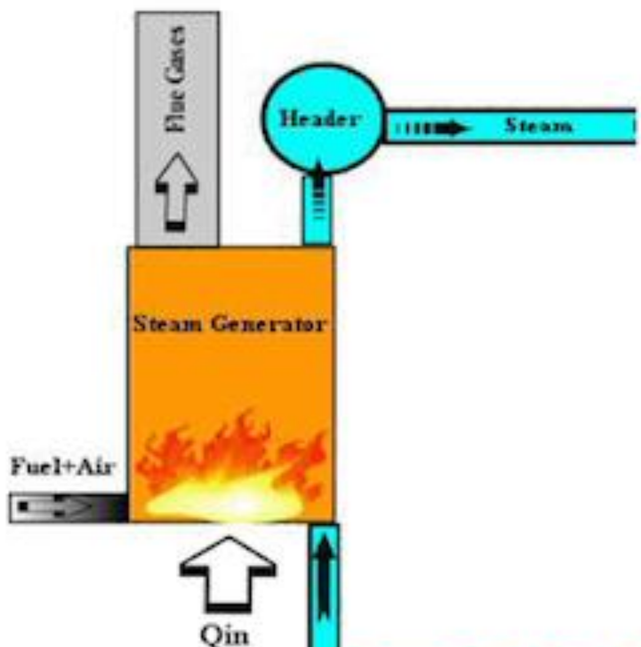
6.1. Generic Analysis Model

There are different international standards available for computation of boiler efficiency such as BS845, ASME PTC-4-1 and Indian Standards IS 8753. By enlarge these standards cover both direct and indirect methods in computing boiler efficiency. However, these standards do not cover blow down losses that is necessary to keep the concentration of dissolved solids to a minimum in a boiler. Hence the App suite includes this under miscellaneous category for the purpose of heat balance and graphical representation of a generic model is shown below.

Qin = Heat Input

Qout = Steam Heat

$$\eta = \frac{Qout}{Qin}$$



6.2. Combustion Model and Heat Balance Sheet

This section briefly describes the combustion model used in the App suite to quantify the Heat energy released and absorbed in a boiler and to keep an account in the form of a balance sheet.

Combustion is a chemical phenomenon where oxygen from the air combines with reactants such as Carbon, Hydrogen, and Sulfur in a fuel to form respective oxides by releasing the heat energy as shown below.

$C + O_2 \rightarrow CO_2$

A pound (0.454 kg) of Carbon combines with 2.66 lb (1.2 kg) of oxygen to liberate about 14,600 BTU (15,400 kJ) heat energy along with 3.66 lb (1.66 kg) of CO_2.

$H_2 + 1/2O_2 \rightarrow H_2O$

A pound (0.454 kg) of hydrogen combines with 8 pounds (3.628 kg) of oxygen to liberate about 62,000 BTU (65,410 kJ) of heat energy with production of 9 lb (4.08 kg) of H_2O.

$S + O_2 \rightarrow SO_2$

A pound (0.454 kg) of Sulfur combines with a pound (0.454 kg) of oxygen to liberate about 4000 BTU (4220 kJ) of heat energy with production of 2 pound (0.908 kg) of SO_2.

Since oxygen content in the air is about 23% by weight, it requires about 11.6 lb (5.261 kg) of air to completely burn a pound of carbon. This is the basis of chemistry and known as stoichiometry. While most of the heat is utilized in generation of pressurized steam in a boiler, some of it gets wasted or lost in the process as listed below.

1. Heat lost in the flue gas.

L1 = m*cp*ΔT

2. Heat loss due to radiation and convection.

L2 = about 2 to 3% of total heat released.

3. Heat carried away by the vapor formed by the hydrogen reaction.

L3 = mH2O*h (specific enthalpy)

4. Heat carried away by the vapor present in the atmospheric air depending upon relative humidity.

L4 = mAtmH20*cp* ΔT

5. Moisture present in the fuel depends upon the source or type of fuel used.

L5 = mFuelH2O*h (specific enthalpy)

6. Presence of carbon monoxide (CO) indicating incomplete combustion there by losing potential source of energy.

L6 = CO(in moles)*293e+3 kJ/kg.mole

7. Frequent blow down of boiler to maintain the concentration of dissolved solids in the feed water to prevent scale formation.

Blow down Heat = feed water*(TDSmin/(TDSmin+TDSmax))*h
Any deviation is accounted under Miscellaneous for further investigation.

8. In case of solid fuels such as Coal, heat is lost in fly ash (L7) and bottom ash (L8) and through partial burning of the fuel. These are estimated minimum values for a typical boiler.
L7/L8 = mAsh*cp*ΔT

9. Steam generation Heat. The boiler efficiency is computed using direct method based on this value.
L9 = (capacity*sp.enthalpy) + (sensible heat)
Where:
sp.enthalpy = Super heat enthalpy - sat.water enthalpy.
Sensible heat = (sat.Temp - feed water Temp)*sp.heat*capacity.

10. The term miscellaneous (L10) is to balance the total Heat credit and debit.
L10 = Total Heat - (L1+L2+L3+L4+L5+L6+L7+L8+L9).

This miscellaneous item, which is conservatively, estimated heat saving needs to be further investigated to realize this potential savings in a boiler. This can be attributed to different factors such as partial burning, blow-down heat, improper combustion, infiltration, bypass and leaks etc.

6.3. Flue gas analysis and Stoichiometry

One way to measure efficiency of the combustion process is to analyze the flue gas for the CO2 content using either portable or full-fledged continuous monitor. The amount of CO2 varies depends upon type of fuel and its air-fuel ratio. Any excess air could also result into loss of heat carried away and calculated using a general expression.

$$\%Heat\ loss = \frac{K(T-t)}{Y}$$

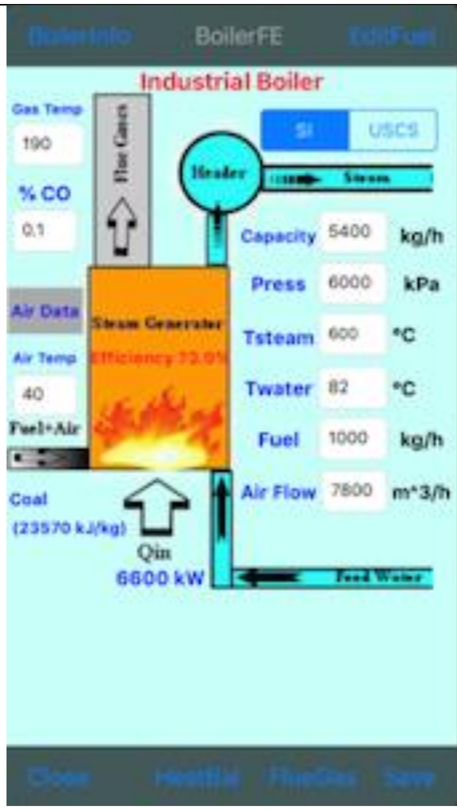
Where:

T = temperature of the flue gases in °C
t = temperature of the ambient air in °C
Y = percentage of CO2 in the flue gases
K = constant for the particular fuel

K value for coal varies from 0.65 to 0.70; fuel oil 0.54 and natural gas 0.40. Supplier of CO2 meter generally would provide specific K value for their respective brand. For an efficient combustion process the percent CO2 would be in the range of 10% - 15%.

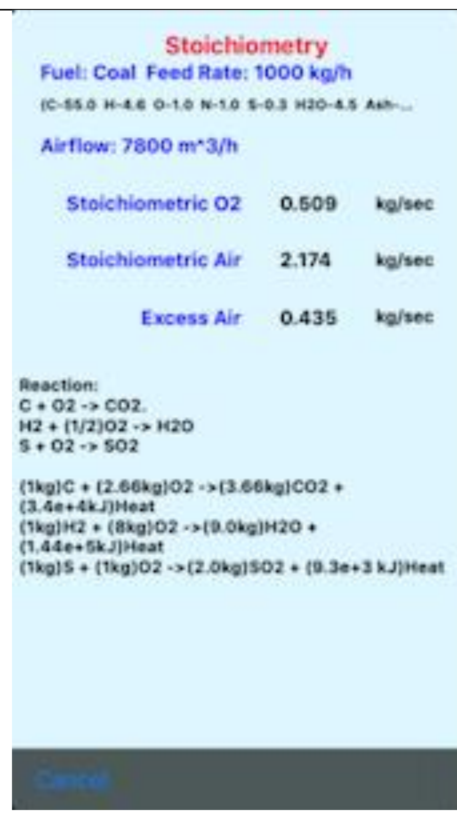
Stoichiometric analysis is done to compute the oxygen requirement for the complete combustion of reactant such as carbon, hydrogen and sulfur in the fuel and also enables us to calculate the actual composition of combustion products in the flue gas. Further this would help in determining amount of air required to supply using different Air fans.

A typical analysis done by the app suite is given below as an example for a generic boiler.


BoilerFE
Industrial Boiler
Gas Temp
190
% CO
0.1
SI
USCS
Flue Gases
Header
Steam
Capacity 5400 kg/h
Press 6000 kPa
Tsteam 600 °C
Twater 82 °C
Fuel 1000 kg/h
Air Flow 7800 m^3/h
Air Data
Steam Generator
Efficiency 73.9%
Air Temp
40
Fuel+Air
Coal
(23570 kJ/kg)
Qin
6600 kW
Feed Water


Combustion Heat Balance Sheet
Fuel: Coal Feed Rate: 1000 kg/h
Airflow: 7800 m^3/h Air Temp: 40 °C
in kW (%)
1. Fluegas heat 393.8 5.97
2. Vapor from H2 Redn 326.1 4.94
3. Fuel moisture heat 30.5 0.46
4. Atm moisture heat 7.3 0.11
5. CO reduction 28.7 0.43
6. Radiation & convection 196.4 2.98
7. Fly Ash 0.26 0.004
8. Bottom Ash 3.8 0.06
9. Steam generation heat 4877.3 73.90
10. Misc (to be reconciled) 735.8 11.15
Total Combustion heat 6600
HeatBalance


Flue Gas Analysis (Ideal)
Flue Gas flow rate 8485 m^3/hr
Gas Temperature 190 °C
Dew Point 47
Flue Gas Composition in % by Volume
Nitrogen (N2) 60.7
Carbon-dioxide (CO2) 13.0
Sulfur-dioxide (SO2) 0.03
Excess Air 15.3
Vapor (H2O) 10.8
Carbon-monoxide (CO) 0.10
% by weight
% by Volume
FlueGas


Stoichiometry
Fuel: Coal Feed Rate: 1000 kg/h
(C-55.0 H-4.6 O-1.0 N-1.0 S-0.3 H2O-4.5 Ash-...
Airflow: 7800 m^3/h
Stoichiometric O2 0.509 kg/sec
Stoichiometric Air 2.174 kg/sec
Excess Air 0.435 kg/sec
Reaction:
C + O2 -> CO2.
H2 + (1/2)O2 -> H2O
S + O2 -> SO2
(1kg)C + (2.66kg)O2 ->(3.66kg)CO2 +
(3.4e+4kJ)Heat
(1kg)H2 + (8kg)O2 ->(9.0kg)H2O +
(1.44e+5kJ)Heat
(1kg)S + (1kg)O2 ->(2.0kg)SO2 + (9.3e+3 kJ)Heat

6.4. Case Examples

6.4.1 Water and coal were measured in a coal–fired boiler at hourly intervals. Weighed quantity of coal was fed to the boiler during the trial period. Simultaneously water level difference was noted to calculate the steam generation during trial period. Blow down was avoided during the test. Following data recorded during the test period. Determine the boiler efficiency using direct method.

Total Heat in (Qin):

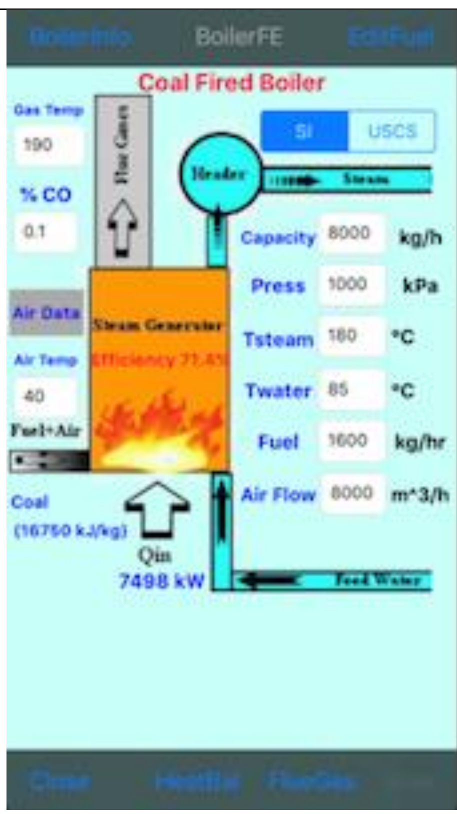
Qin = Mcoal*HCV + Mair*cp*ΔT = 7498 kW

Total Heat out (Qout):

Steam generation = 8000 kg @ 1000 kPa, 180 °C (h=2776 kJ/kg)

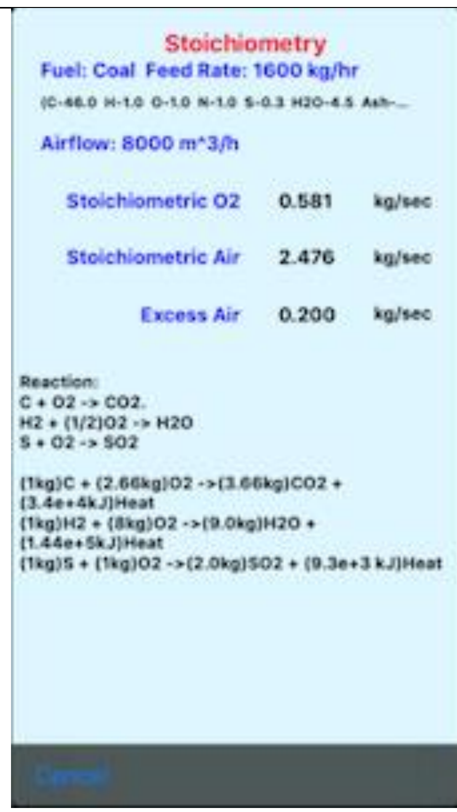
Feed water @85 °C (h = 356 kJ/kg)

Qout = 8000*(2776 – 356) = 5377 kW η = Qout/Qin = 71.7%



Please note in the above analysis result from App suite steam generation heat is calculated using the model given in L9 under section 6.1 is little different from conventional direct method. App suite calculates the total heat added for steam generation under each segment like sensible heat, latent heat and then superheat. However, the total should be same with a small difference due to round off error in the property table.

Flue gas analysis and stoichiometry of coal combustion is as shown below. Accordingly, the CO2 in flue gas is about 17.2% and excess air is limited to about 7%.



6.4.2 Utility boiler uses natural gas @ 500 m^3/hour to generate steam @ 4500 kg/hour with pressure 5000 kPa and temperature 450 °C. Feed water is supplied at 82 °C. Determine the boiler efficiency by direct method given natural gas density 0.76 kg/m^3 and HCV 49500 kJ/kg.

Total Heat in (Qin):

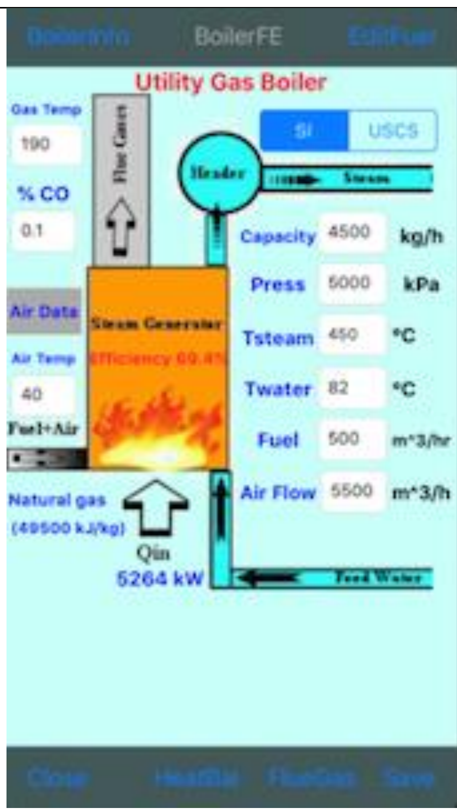
Qin = Q*ρ*HCV + Mair*cp*ΔT = 5264 kW

Total Heat out (Qout):

Steam generation = 4500 kg @ 5000 kPa, 450 °C (h=3318 kJ/kg)

Feed water @82 °C (h = 343 kJ/kg)

Qout = 8000*(2776 – 356) = 3718 kW   η = Qout/Qin = 70.6%



In the above example boiler efficiency calculated by App suite is less than the efficiency calculated by direct method. It is mainly due to the way total steam generation heat is computed using individual segments from feed water to superheat conditions. Combustion air supply rate is adjusted to optimize CO2 and excess air in the flue gas. The analysis results are given below.

6.5. Maintenance of Boiler and Auxiliaries

6.5.1 Performance Monitoring

Performance monitoring of boiler and auxiliaries can help assess the maintenance requirements time to time depending upon both combustion and energy conversion efficiencies. The modern instrumentation and control functions do this job on a continuous basis such as monitoring air-fuel ratio, pressure, temperature, and flow rates at certain critical points. However, a thorough evaluation of the overall performance needs to be assessed based on these parameters that enables to deploy maintenance resources for more reliable and cost-effective operation of the boiler house. Some of this can be listed below.

1. Excess air control using online O2 and CO2 analyzers.
2. Stack temperature and dew point control.
3. Air-fuel ratio and proper distribution of all reactants within combustion chamber with temperature control.
4. Feed water and boiler water quality management and control.
5. Cleanliness control especially in water tube boilers for optimum U value (Overall heat transfer coefficient).

6.5.2 Maintenance of Auxiliaries

There are number of critical boiler auxiliaries such as pumps, fans, blowers, feeders, and burners. All these can be grouped under rotating equipment that requires periodic cleaning, lubrication, inspections and replacement of worn-out components, seals, gaskets etc. Therefore, a comprehensive preventive maintenance schedule with checklists as per recommendations of the respective equipment supplier needs to be put in place with adequate maintenance resources for execution.

6.5.3. Boiler Shutdown Maintenance

It is common practice to carry out a thorough inspection and maintenance of boiler pressure parts and other static components once in a year during annual shutdown. This covers non-destructive examination of all pressure components such as boiler drum; tubes, pipes, valves, and all other external heat transfer areas that are subjected to corrosive and erosive wear (Table 15.1.1). Corrective actions are planned and executed accordingly and then certified by the relevant authorities after performing the pressure tests as per ASME or national standards as applicable.

In US, Occupational Safety and Health Administration's (OSHA) standard on the Control of Hazardous Energy (Lockout-Tag out), found in CFR 1910.147, spells out the step's employers must take to prevent accidents associated with hazardous energy. The standard addresses practices and procedures necessary to disable machinery and prevent the release of potentially hazardous energy while maintenance or service is performed. Here is some guideline as per US conventions and practice for lockout, Tag-out.

1. No two keys or locks should ever be the same.
2. A staff member's lock and tag must not be removed by anyone other than the individual who installed the lock and tag unless removal is accomplished under the direction of the employer.
3. Lock and tag devices shall indicate the identity of the employee applying the device(s).
4. Tag devices shall warn against hazardous conditions if the machine or equipment is energized and shall include directions such as: **Do Not Start. Do Not Open. Do Not Close. Do Not Energize. Do Not Operate.**
5. Tags must be securely attached to energy-isolating devices so that they cannot be inadvertently or accidentally detached during use.

7. Heat Exchanger Performance

7.1. Generic Design Analysis Model

Heat exchanger is any device that allows transfer of thermal energy from one fluid to another. Among various devices Shell-and-Tube is the most common device used in chemical and process industries. Many factors get into the design of heat exchangers such as thermal analysis, size, pressure drop, material of construction and cost. While structural design standards follow ASME code for Unfired Pressure Vessels, cost depended upon the size and other design parameters. A typical generic model of a two pass Shell-and-Tube exchanger is shown below. Hot fluid (Cyan) enters the tubes at temperature Th_in and exit at Th_out Cold fluid (Blue) enters the shell at temperature Tc_in and exit at Tc_out.

$Q_{hot} = m_h * Cp_h * \Delta T_h$

$Q_{cold} = m_c * Cp_c * \Delta T_c$

Under steady state condition

Heat lost = Heat gained

$Q = Q_{hot} = Q_{cold}$

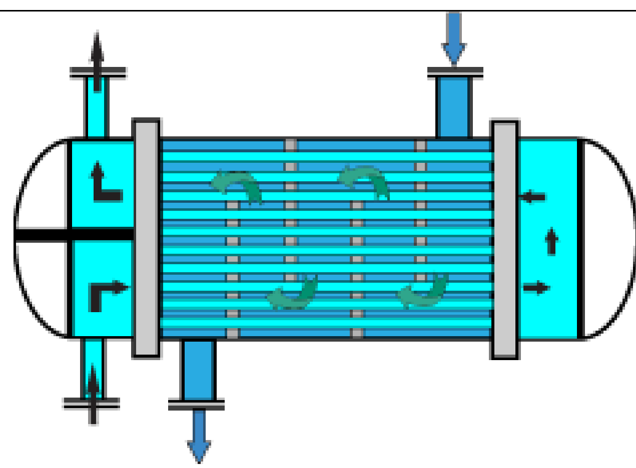
$Q = U * A * \Delta T$

Where:

U – Overall heat transfer coeff.

A – Total heat transfer area

ΔT – Average effective temperature difference.



7.2. Shell-and-Tube Design Analysis

7.2.1. Effectiveness

Heat exchanger effectiveness is defining as ratio of actual heat transfer to maximum heat transfer potential.

$$\varepsilon = Q_{actual}/Q_{max}$$

Where the maximum possible heat transfer is that which would result if one fluid underwent a temperature change equal to the maximum temperature difference available i.e., the temperature of the entering hot fluid minus the temperature of the entering cold fluid.

C = heat capacity = $m*C_p$;

$$Q_{actual} = C_h(T_{h\_in} - T_{h\_out}) = C_c\,(T_{c\_out} - T_{c\_in})$$

The maximum possible heat transfer occurs when smaller C undergoes maximum temperature difference available and given as:

$$Q_{max} = C_{min}(T_{h\_in} - T_{c\_in})$$

Combining above equations we get

$$Q_{actual} = \varepsilon C_{min}(T_{h\_in} - T_{c\_in})$$

7.2.2. Capacity ratio

It is the ratio of heat capacity ratio of cold fluid to hot fluid irrespective of whether the hot fluid is in shell or tube side. Heat capacity rate is designated as:

$$C = m*C_p.$$

Capacity Ratio $r = C_c/C_h$

7.2.3. Net transfer Units (NTU)

It is the maximum number of transfer units possible in a heat exchanger considered as its size factor and given by:

NTU = UA/C_{min} (C_{min} is minimum heat transfer capacity)

7.2.4. LMTD

The average effective temperature difference for Shell-and-Tube exchanger is called log-mean-temperature-difference (LMTD). Its value depends upon type of flow between cold and hot fluid. Flow can be counter flow, parallel flow or cross flow and given by the following equation.

$$LMTD = \frac{(\theta 1 - \theta 2)}{\ln\left(\frac{\theta 1}{\theta 2}\right)}$$

Where:

For counter flow:	For parallel flow:	For cross flow:
$\Theta 1 = T_{h\_in} - T_{c\_out}$	$\Theta 1 = T_{h\_in} - T_{c\_in}$	$\Theta 1 = T_{h\_in} - T_{h\_out}$
$\Theta 2 = T_{h\_out} - T_{c\_in}$	$\Theta 2 = T_{h\_out} - T_{c\_iout}$	$\Theta 2 = T_{c\_out} - T_{c\_in}$

7.2.5 Correction Factor

A correction factor is applied for cross flow as well as number of fluid-passes in a shell-and-tube heat exchanger is function of both capacity ratio and effectiveness and as given below.

$$F = \frac{(r+1)^{\frac{1}{2}} \ln(1-r\varepsilon)/(1-\varepsilon)}{(1-r)\ln\left(\frac{2-\varepsilon(r+1-\sqrt{r+1})}{2-\varepsilon(r+1+\sqrt{r+1})}\right)}$$

Correction factors for different heat exchanger conditions are also can be obtained from the standard handbook on heat transfer as a function of effectiveness and capacity factor. The app suite applies this factor; however, under certain conditions it could become complex (√-1) in which case default value will be used with a prompt to add correction from handbooks for more accurate results.

7.3. Case Examples

7.3.1 A regular shell-and-tube counter flow type heat exchanger is used for cooling hot oil @1200 lpm from 90 °C to 48 °C. Water is on tube side flows @975 lpm at 20 °C leaves the exchanger at 44 °C. There are 150 schedule 40, ¾" diameter pipes. Determine overall heat transfer coefficient of the exchanger.

Given:

Pipe Diameter -= 26.67 mm; L = 7.2 meter; No of pipes N = 150

Total heat transfer Area (A) = π*D*L*N = 90.5 m^2

Parameter	**Shell side (oil)**	**Tube side (Water)**
Flow rate (lpm)	1200 lpm	875 lpm
Inlet Temperature	90 °C	20 °C
Exit Temperature	48 °C	44 °C
Density (kg/m^3)	846	998
Specific heat (J/kg.K)	2177	4181
Mass flow (kg/sec)	16.92	14.55
Heat Capacity C	36.83	60.83
Heat (loss/gain)	1547 kW	1460 kW

Θ1 = 90 – 44 = 46

Θ2 = 48 – 20 = 28

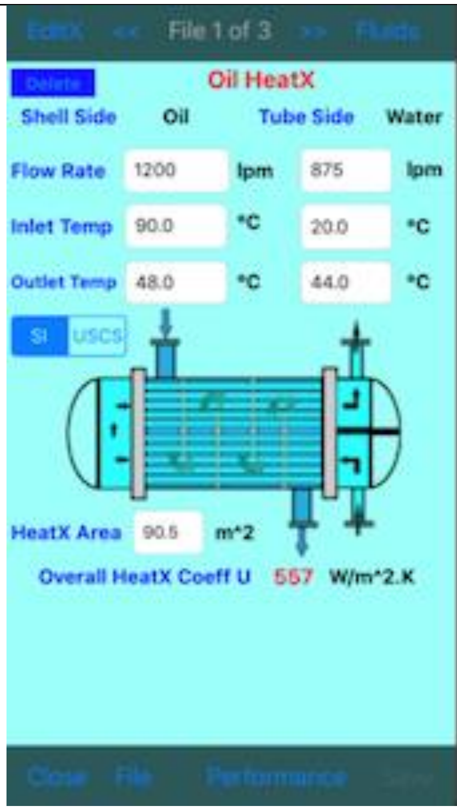
LMTD = (θ1 – θ2)/Ln (θ1/θ2) = 18/Ln (1.642) = 36.26 °C

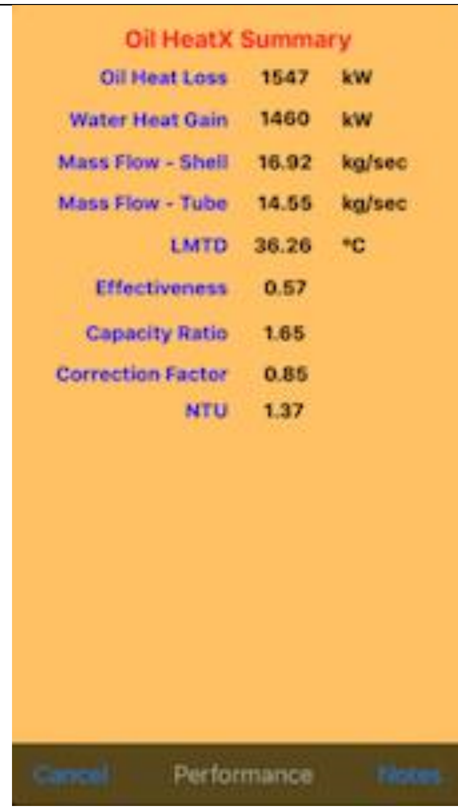
Capacity Ratio r = 60.83/36.83 = 1.65

Effectiveness ε = 0.57

Correction Factor F= 0.85

Overall Heat Transfer coefficient U = Q/(A.ΔTm*F) = 557 W/m^2.K





7.3.2 A surface condenser receives steam at pressure -0.25 Bar and 92 °C @ 1000 kg/hr. Cooling water flows through 140 schedule 40, ¾" pipe @900 lpm. Determine overall heat transfer coefficient of a single pass condenser.

Pipe Diameter -= 26.67 mm; L = 7.2 meter; No of pipes N = 140

Total heat transfer Area (A) = π*D*L*N = 85 m^2

Parameter	**Shell side (Steam)**	**Tube side (Water)**
Flow rate	1000 kg/hour	900 liter/min
Condition	-0.25 bar and 92 °C	20 °C / 30 °C
Latent heat (kJ/kg)	2278	-
Sp.Heat (J/kg.K)	2000	4182
Heat (Loss/Gain)	633 kW	626 kW

$\Theta 1 = 92 - 20 = 72$

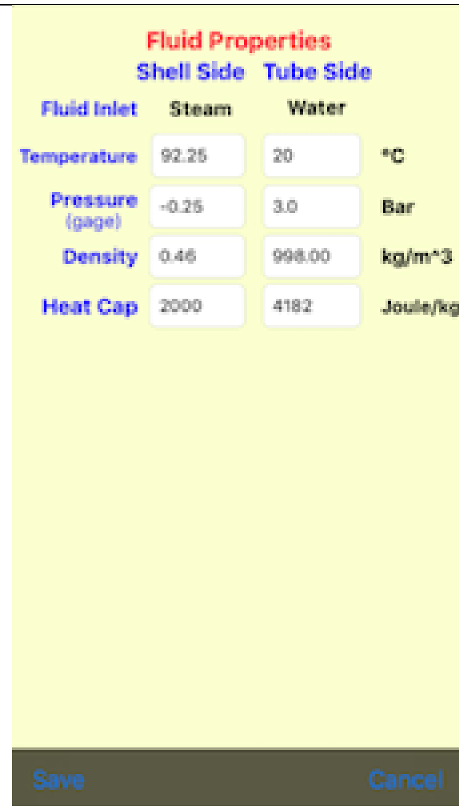
$\Theta 2 = 92 - 30 = 62$

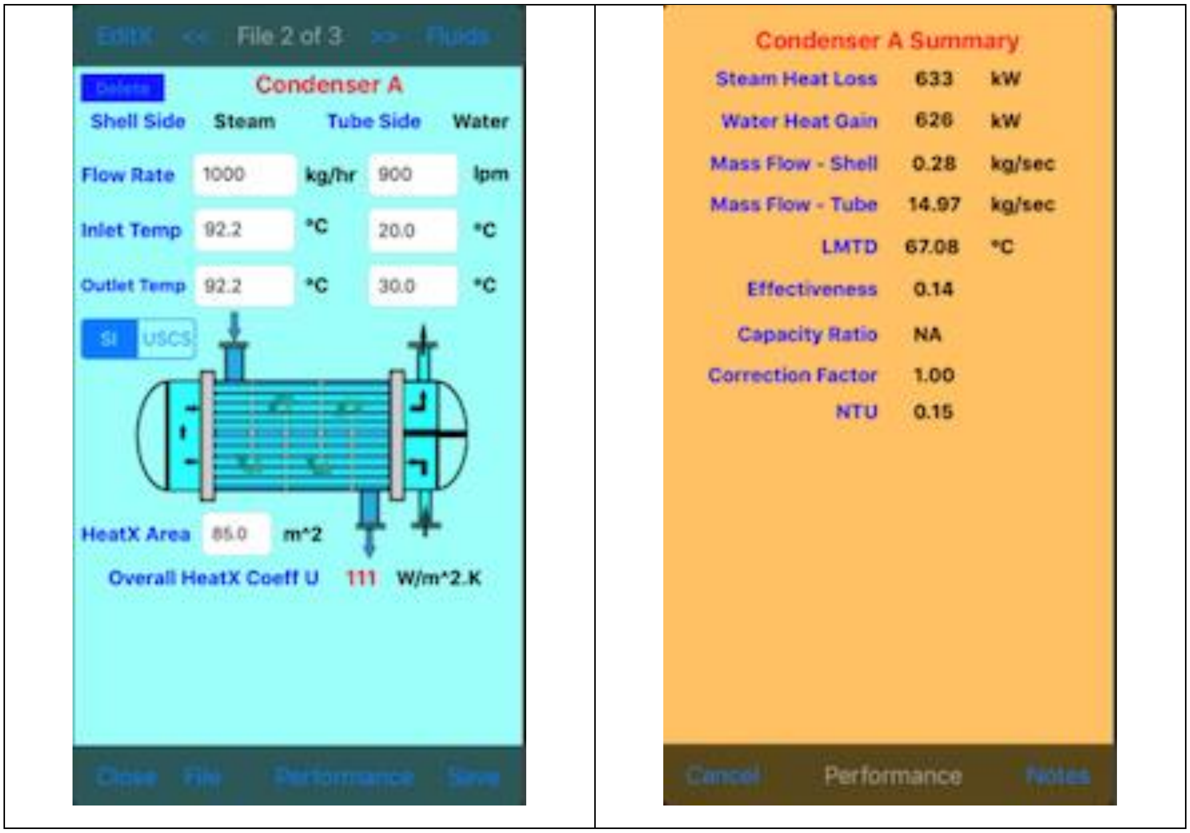
LMTD = $(\theta 1 - \theta 2)/\mathrm{Ln}\,(\theta 1/\theta 2) = 10/\mathrm{Ln}\,(1.161) = 66.88$ °C

F = 1 (For single pass)

Overall Heat Transfer coefficient U = Q/(A.ΔTm*F) = 111 W/m^2.K

App Suite Analysis:





7.4. Maintenance of Heat Exchangers

7.4.1 Performance Monitoring

Heat exchanger parameters used in computation of U values are constantly monitored or logged on daily basis. This helps mainly in trend monitoring of temperature differential; fluid pressure drops in both shell side and tube side-indicating fouling and build up increasing the resistance to flow as well as heat transfer. Based on the severity of application one can determine its periodicity of cleaning and other maintenance activities.

7.4.2 Maintenance criteria

Maintenance of heat exchangers is mainly dictated by the type and severity of application and fluid conditions. For example, steam condensers that uses cooling water source from sea or river or ponds its cleaning periodicity dictated by the water quality. Seawater can be very corrosive while river water can have more pollutants, dirt, algae, and scum that affects the life of heat exchanger components. Therefore, proper cooling water quality management is also part of heat exchanger maintenance that determines not only periodicity also type of inspection and testing required for shutdown maintenance.

7.4.3. Shutdown Maintenance

Heat exchangers are taken under shutdown maintenance depending upon the severity and criteria stated above. In US, OSHA guidelines are applicable for Lockout tag-out during shutdown maintenance. Here is a general checklist of activities under shutdown maintenance.

1. Removal of covers, tube bundle and rigging procedure.
2. Thorough inspection of internal components for fouling and corrosion and erosion wear.
3. Cleaning and renewal of tubes based on residual thickness.
4. Coating and final inspection and pressure testing.
5. Certification and box up procedures.

Finally, performance evaluation and determining new U values under steady state conditions.

8. Air Compressor Capacity Performance

8.1. Generic Design Analysis Model

Compressed air system is one of the energy intensive utilities in industrial applications that cover manufacturing, mining, chemical and process industries. In manufacturing mainly used in low-cost automation while in process industries it is used for instrumentation and control systems. Thus, compressors are made available for a wide range of pressure and capacities and it is also one of the least energy efficient as it always operates between two pressure points for loading and unloading besides a lot of leakage in the delivery systems. A generic model is presented here for analysis purpose.

Free Air Delivery (FAD):

Swept Volume $V1 = 0.25\pi D^2 L$

Clearance Volume V2 ≅ (0 – 8%)V1

Theoretical FAD = V1*rpm*stages*no of cyl

Pressure ratio r = (P2/P1)

Volumetric $\eta = \frac{Actual\ FAD}{Theoretical\ FAD}$

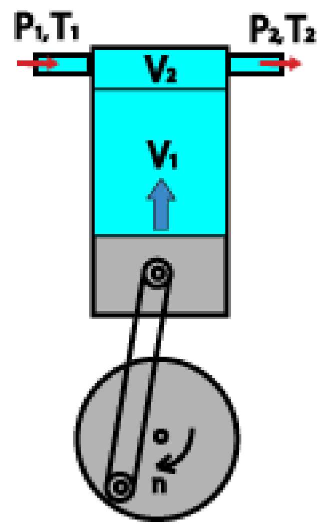
Isothermal $Power = \frac{P_1 V_1 \log\left(\frac{P_1}{P_2}\right) N}{60 * 1000}\ kW$

Isothermal $\eta = \frac{Isothermal\ Power}{Shaft\ power}$

Isentropic Power $= \left(\frac{P_1 * FAD}{60000}\right)\left(\frac{\gamma}{\gamma - 1}\right)\left(r^{\frac{\gamma-1}{\gamma}} - 1\right)$

Compressor $\eta = \frac{Isentropic\ Power}{Shaft\ Power}$

Specific Power $= \frac{Actual\ power}{FAD}$



8.2. FAD and Leakage Testing

8.2.1. FAD testing

Compressor capacity is specified by its specific power requirement as well as free air delivery (FAD) by the manufacturers. The capacity performance can degrade over a period depends upon extent of wear in the cylinder surface resulting into internal leakages. Therefore, it is always recommended to test capacity performance of compressors time to time. There are two methods commonly followed for testing FAD in industries.

a. Using standard orifice plate/nozzle method
b. Measurement of FAD by pump up method.

Nozzle Method:

In nozzle method pressure drop is measured across a standard nozzle of known diameter and then computed using equation given in section 3.1. However, care should be taken ensure proper damper chamber prior to the nozzle is fitted to reduce the effect of pulsating flow from a reciprocating compressor. However, for a rotary compressor this method works very well.

Pump up method:

This method is quicker way to estimate FAD for a reciprocating compressor since measurement is done as the compressor charge the tank of known volume using the following procedure.

1. Empty and drain water from the tank after tightly closing the delivery valve.
2. Note the tank pressure gage P1 and temperature T1 at '0'sec.
3. Start the compressor and start the stopwatch.

4. Note the time required to attain rated pressure P (t) and temperature T2 after t minute.
5. Calculate using the following equation.

FAD = $\left(\frac{V}{time}\right)\frac{(P_2 - P_1)}{P_{amb}}$ in liter/gallon per minute

Where:

V = Tank volume liter/gallon

P2 = P(t)T1/T2;

P_{amb} = Ambient pressure (atmospheric)

8.2.2 Leakage Testing

The above similar procedure can be followed to measure leakage in the compressed air system delivery line as given below.

1. Close all final delivery points.
2. Take initial and final tank pressure with time. Initial and final temperature assumed to be same.
3. Calculate using the following equation.

Leakage Flow = $\left(\frac{V}{time}\right)\frac{(P_1 - P_2)}{P_{amb}}$ in liter/gallon per minute

Apply appropriate unit standards if any for computing FAD either in lpm (liter per minute) or cfm (cubic feet per minute).

8.3. Case Examples

8.3.1 A compressed air tank maintained at 7 bar gage pressure at 25 °C is charged using a single stage 15 kW compressor running at 900 rpm with FAD capacity 2040 lpm and ambient air at 30 °C. Compressed air is passed through a cooler for dehumidification and also to maintain discharge temperature. Determine efficiencies and specific power of air compressor.

Given:

Pressure ratio r = (P2/P1) = (7+1)/1 = 8

Temperature T1 = (273.15 + 30) K

Specific heat ratio of air Y = 1.4

Temperature $T_2 = T_1\left(r^{\frac{Y-1}{Y}}\right) = 303.15 * 8^{\frac{0.4}{1.4}} = 549.14\ K$ = 276 °C

Specific power = kW/FAD = 15.0/(2.04*60) = 0.122 kW/m^3/hr

Isothermal $Power = \frac{P_1 FAD \log\left(\frac{P_1}{P_2}\right) N}{60*1000}\ kW = \frac{1*10^3 2040*\log(r)*900}{60*1000} = 7.07kW$

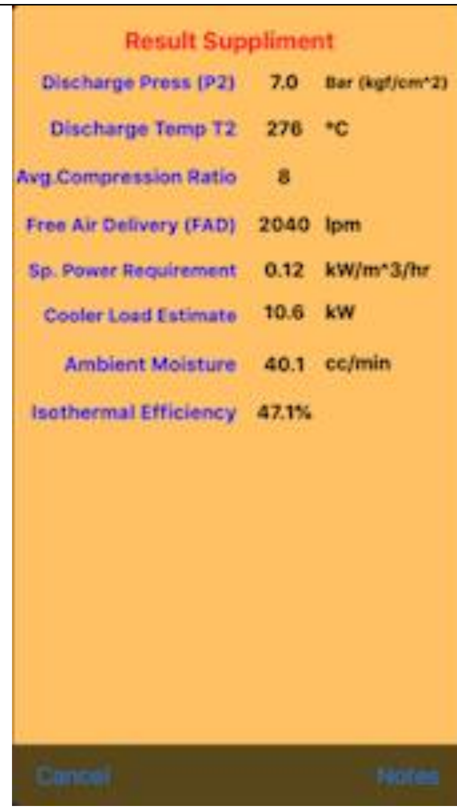
Isothermal $\eta = \frac{Isothermal\ Power}{Shaft\ power} = \frac{7.07}{15} = 0.4713$

Isentropic Power $=\left(\frac{P_1*FAD}{60000}\right)\left(\frac{Y}{Y-1}\right)\left(r^{\frac{Y-1}{Y}} - 1\right) = \frac{100*2040}{60000}\frac{1.4}{0.4}(8^{\frac{0.4}{1.4}} - 1)$

Isentropic Power=9.656 kW

Compressor η = $\frac{Isentropic\ Power}{Shaft\ Power} = \frac{9.7}{15} = 0.644$

App Suite Analysis



8.3.2. A 2500-liter capacity reservoir charged from 0 to 7 bar pressure in 9 minutes while temperature changes from 30 °C to 35 °C. Determine FAD of the compressor.

Given:

Tank volume V = 2500 liter

Pressure P1 = 0 (gage)

Ambient = 1 bar (absolute);

T1 = (273.15 + 30) K; T2 = (273.15 + 35) K

Pressure P2 = 7 bar *T1/T2

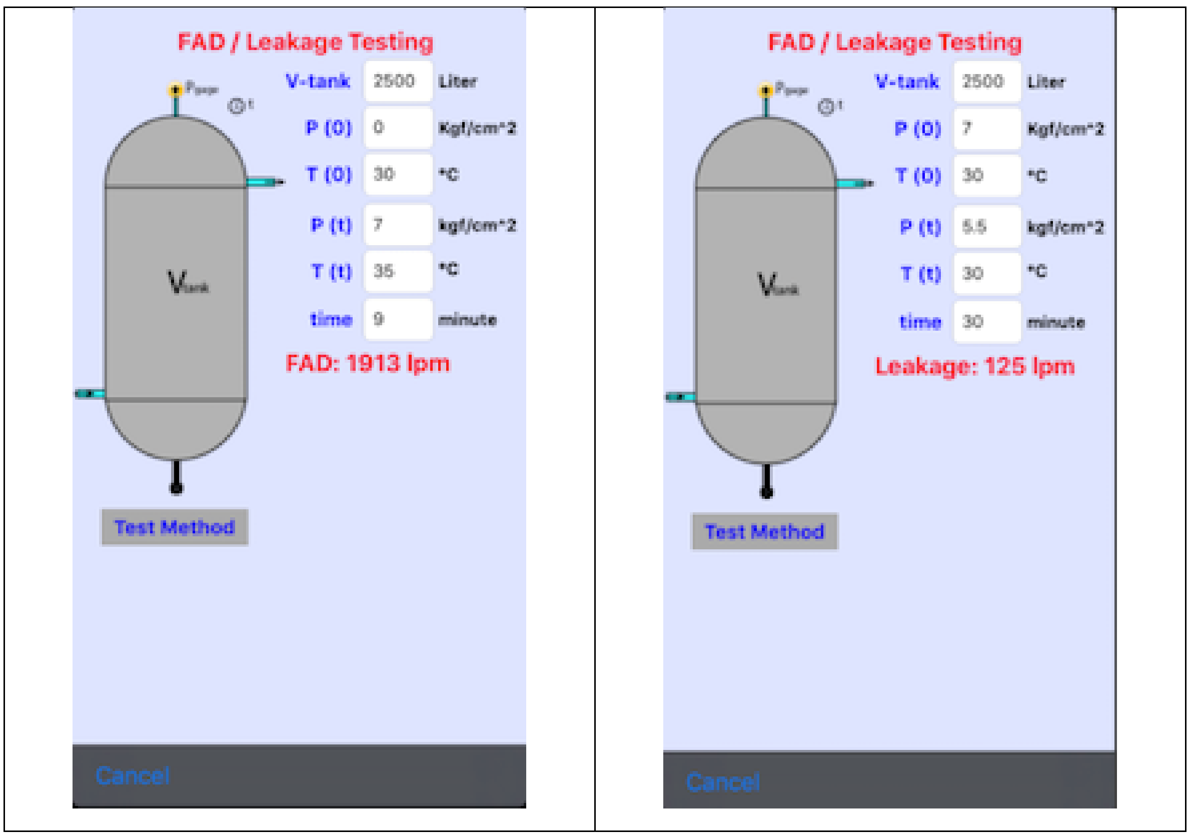
Time = 9 minute

FAD = $\left(\frac{V}{time}\right)\frac{(P_2-P_1)}{P_{amb}}$ in liter = 1913 lpm

8.3.3. In the above case under idle conditions the tank pressure drops from 7 bar to 5.5 bar in 30 minutes. Determine the leakage rate in the compressed air system.

Leakage Flow = $\left(\frac{V}{time}\right)\frac{(P_1-P_2)}{P_{amb}} = \frac{2500}{30}\left(\frac{150\ kPa}{100kPa}\right) = 125\ lpm$

App Suite Analysis



8.3. Maintenance of Air Compressors

8.3.1 Performance Monitoring

Specific power requirement by and large a good indicator of compressor performance. Therefore, FAD needs to be tested time to time or at least once in six months to assess its maintenance requirement. In addition, perform periodic checks and visual inspection of the compressors while under running condition weekly or fortnightly depending upon the type and severity and dusty operating conditions. A general checklist is given below.

1. Check compressor lubricating oil level and top-up if necessary.
2. Drive belts and mechanical coupling conditions for any slackness or looseness.
3. Tighten all foundation bolts using torque wrench as per recommended torque.
4. Observe for excessive noise and leakages in discharge lines.
5. Air intake manifold inspection and check filter conditions.
6. Loading-unloading valve and safety valve inspection.
7. Check intercooler, after cooler drain and clean and inspect internal components whenever compressor is idling.

Any increase in the specific power consumption is indicator of excessive internal leakages. Leakages caused by excessive wear of compressor body and moving parts hence calls for a complete overhauling.

8.3.2. Air Compressor Overhauling

Air compressor whether it is rotary or reciprocating type requires renewing and replacement of worn-out components such as wear ring, piston rings, belts, flexible coupling etc. after certain hours of continuous operation. Manufacturers often provide the guidelines for renewing and undertake major job in case of renewing cylinder liners.

9. Fan-Duct Spec Performance

9.1. Generic Design Analysis Model

Fan-Duct system is designed to cater various industrial applications such as boilers, HVAC, mining, and number of other process applications. Fans are categorized based on type of flow that can be either axial or radial by centrifugal action. In each case its performance characteristics are measured in terms of flow rate, static pressure and shaft power supplied by the motor and its efficiency and specific speed. There exists a definite relationship between fan rpm (N) and its performance characteristics and given in the form 'Fan Law' as given below. Specific speed is used mostly to compare two different fans for the same application, which is function of flow rate, pressure, and density.

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Fan Law:

1. Q (flow rate) ∞ N
2. H (Fan pressure) ∞ N^2
3. P (Fan power) ∞ N^3

Specific speed = $\frac{N*Q^{0.5}*\rho^{0.75}}{H^{0.25}}$

Reynolds No = $\frac{\rho*vel*D}{\nu}$

Fan Power = $\frac{Q*H*g}{60*1000}kW$

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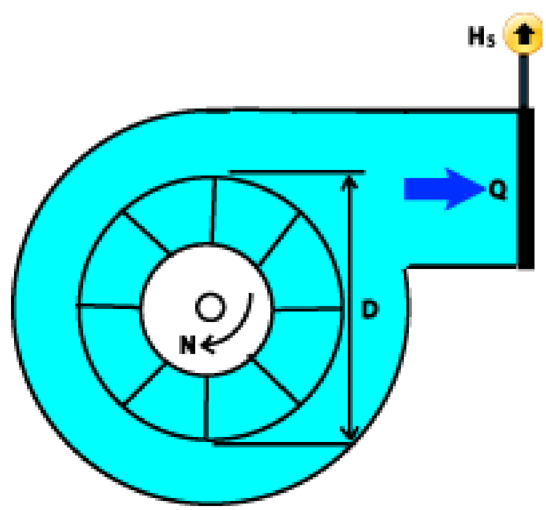

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Flow rate is a product of flow area and average velocity in ducts.

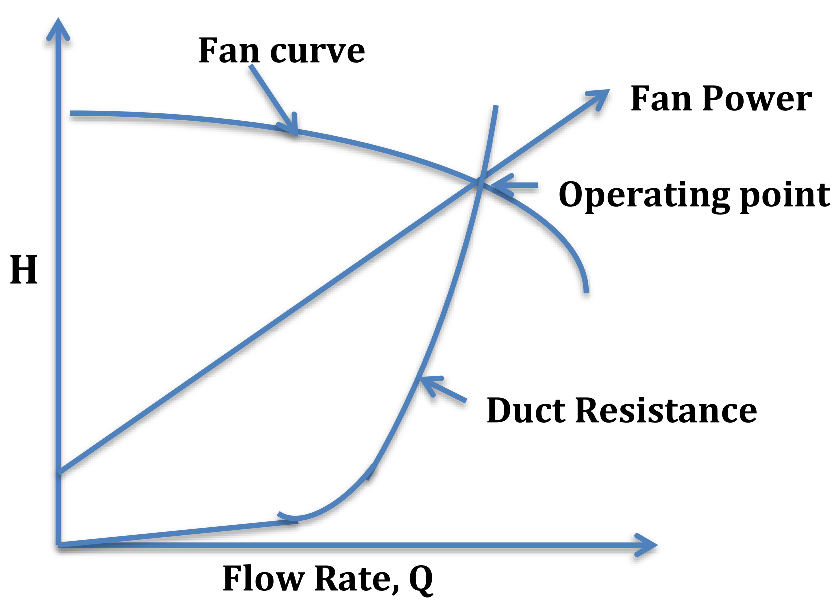
Velocity pressure H_v = H_d (dynamic) – H_s (static)

Average velocity v = √ (2g*H_v)

Flow rate Q = A (duct area) *v (velocity)

9.2. Fan-Duct system properties

Design and selection of duct system is important in determining the duct velocity and to deliver the required flow rate. Average velocity is measured in a straight duct using Pitot tube, which measures total dynamic pressure along the duct. The difference between dynamic pressure and the static pressure gives us the velocity pressure. Most manufacturers provide fan performance data in the form of fan curve (Q vs. H). However, the operating point on this curve determined by the total resistance offered by the fan duct system. For this purpose, total equivalent length is used as measure of total resistance that includes actual duct friction loss, which is function of duct material and Reynolds number, and other minor losses due to fittings and terminal boxes.



Ducts are classified under six different classes and cross section such as rectangular, square, and round.   Cross section area of a rectangular ducts depends on aspect ratio (long side/short side) and been suggested for a given duct class. The allowable leakage for a given duct class as per recommended standard is given as:

$$Q_{leak} = Class\# * H_s * 0.65$$

Where:

H_s – is given in inch water gage

Q_{leak} – cfm/100^ft^2 or lpc(liter/sec) for 20m^2 surface area

App suite provides the above standard values for any given duct class and aspect ratios in case of rectangular ducts for the given long side and short side. Aspect ratio is always taken as unity for both square and round duct cross sections.   Equivalent length needs to be calculated based on total pressure drop. One can use the procedure given in section 3.0 for any gaseous fluids to compute the duct loss due to fittings and terminal boxes. Alternatively, for Air, it can be computed using duct loss charts that are available in ASHRAE or any standard HVAC handbook.

It is often necessary to run validation tests to determine all the key parameters such as duct velocity, leakage and total duct resistance for satisfactory long-term performance and energy efficiency in a fan-duct system.

9.3. Case Examples

9.3.1. Axial flow HVAC fan supplies air @ 10600 cfm, 68 °F and measures static pressure at fan outlet as 1.25” water gage. A class 3, 26” GI duct is used with aspect ratio of 1.083 and equivalent length of 200’. Determine air horsepower (AHP) of the fan.

Air flow rate: 10600 ft^3/min (cfm) = 18000 m^3/hr

Duct area A: 26*(26.0/1.083) = 4.334 ft^2 = 0.4026 m^2

Duct velocity: Q/A = 10600/4.334 = 2446 ft/min = 12.42 m/sec

Static Pressure $H_s = 1.25"$ water gage (1” water gage = 249.1 Pa)

Velocity pressure $H_v = \frac{\rho * v^2}{2}\frac{1}{\gamma_{water}} = \frac{1.2*12.42^2}{2*1000*g} = 0.372\ in\ water\ gage$

Total dynamic head: $H_s + H_v = 1.621"\ water\ gage\ =\ 404\ Pa$

AHP = Q*H = 2.019 kW = 2.71 hp

App Suite Analysis

9.3.2. A boiler FD fan supplies air at 40 °C, 250 m^3/min. Fan has a centrifugal type impeller size 40 cm coupled to a 3-kW induction motor running at 1440 rpm. A class 3, 24" diameter round duct measures 40 mm static head with a 75-meter equivalent length. Determine air power and efficiency of the fan-duct system.

Given:

Air flow rate Q – 250 m^3/min;

Area A = $0.25 * \pi * 0.61 = 0.293\ m\text{^}2$

Duct velocity v = Q/A = 14.3 m/sec

Velocity pressure $H_v = \frac{\rho * v^2}{2} \frac{1}{Y_{water}} = \frac{1.12*14..3^2}{2*1000*g} = 11.7\ mm\ water\ gage$

Total dynamic head: $H_s + \ H_v = 40 + 11.7 = 51.7\ mm\ water\ gage$

H = 51.7 water gage = 507 Pa

Fan power = $\frac{Q(\frac{m^3}{\min})*H(kPa)}{60*1000} = \frac{250*507}{60*1000}$ = 2.1 kW

Efficiency η = $\frac{Fan\ power}{motor\ power} = \frac{2.1}{3.0} = 70\%$

App Suite Analysis:

Fan and duct system properties are shown in their respective input data screen.

The above result summary is displayed separately for fan performance as well as class 3, 24" GI round ducts. The specific speed, Reynold's number and duct loss per 100m-duct length are part of applicable standards for this system.

9.4. Maintenance of Fan-duct system

9.4.1. Performance Monitoring

Fans that are used for constant flow applications such as boiler, furnace etc. requires constant monitoring of both fan-duct system parameters discussed in the previous sections. It is expected to deliver flow under constant operating point on its characteristics curve. Therefore, it requires constant monitoring of static pressure in critical sections to know change in pressure drops for the given length of the duct. Any considerable change in the duct losses is an indication of excessive wear or build up that requires attention under maintenance. In addition, one can perform following maintenance checks weekly, monthly depending upon the severity of applications.

9.4.2. Maintenance Checks

1. Periodic inspection of all system components
2. Bearing lubrication and renewals
3. Belt tightening and replacement
4. Sound and vibration checks
5. Duct leakage and fouling, insulation breakdowns.

9.4.3. Shutdown Maintenance

It is general practice to clean both external and internal duct systems, fan casing, dampers, and control systems once in a year during annual shutdown or turn around maintenance. Some of the periodic checks and inspections performed help in planning these activities ahead of time. Reestablish all the key parameters and base line data of fan-duct system for subsequent performance monitoring.

10. Rankine Cycle

10.1. Generic Design Analysis Model

Rankine cycle is one of the most common vapor cycle heat engines that uses water as the working fluid that requires high latent heat of energy to convert from liquid phase to vapor phase. This phase change takes place under isothermal and isobaric conditions and absorbs 1024 kJ of heat energy per kg of water at atmospheric pressure. At high pressure and temperature, this energy is converted into mechanical energy upon expansion in a turbine and after releasing all the heat energy to become liquid that can recirculate. Thus, a basic rankine cycle comprises of boiler, turbine, condenser, feed pump and the graphical representation is given below with parameters that determine the efficiencies in a rankine cycle.

Q_{in} = Steam heat input

Q_{out} = Heat rejected

W_t = Power output

$Q_{total} = M_{fuel} * HHV$

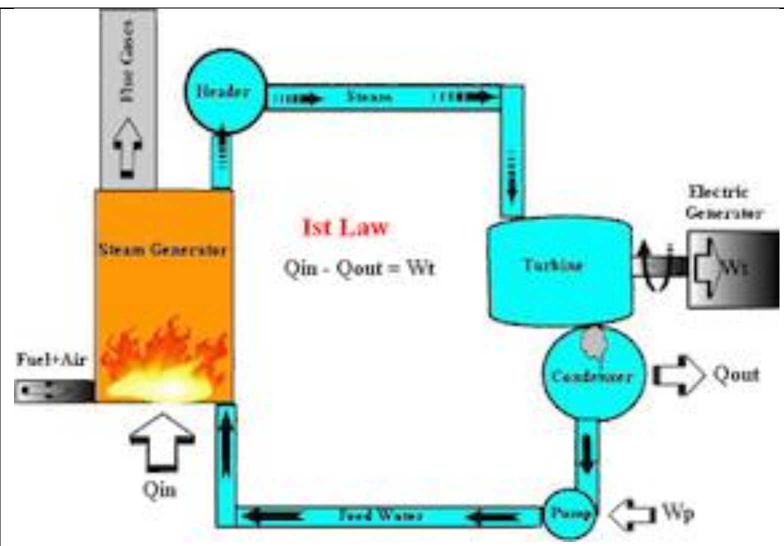
HHV = HighHeatValue

Thermal Efficiency:

$$\eta = \frac{W_t - W_p}{Q_{in} - Q_{out}}$$

Boiler Efficiency:

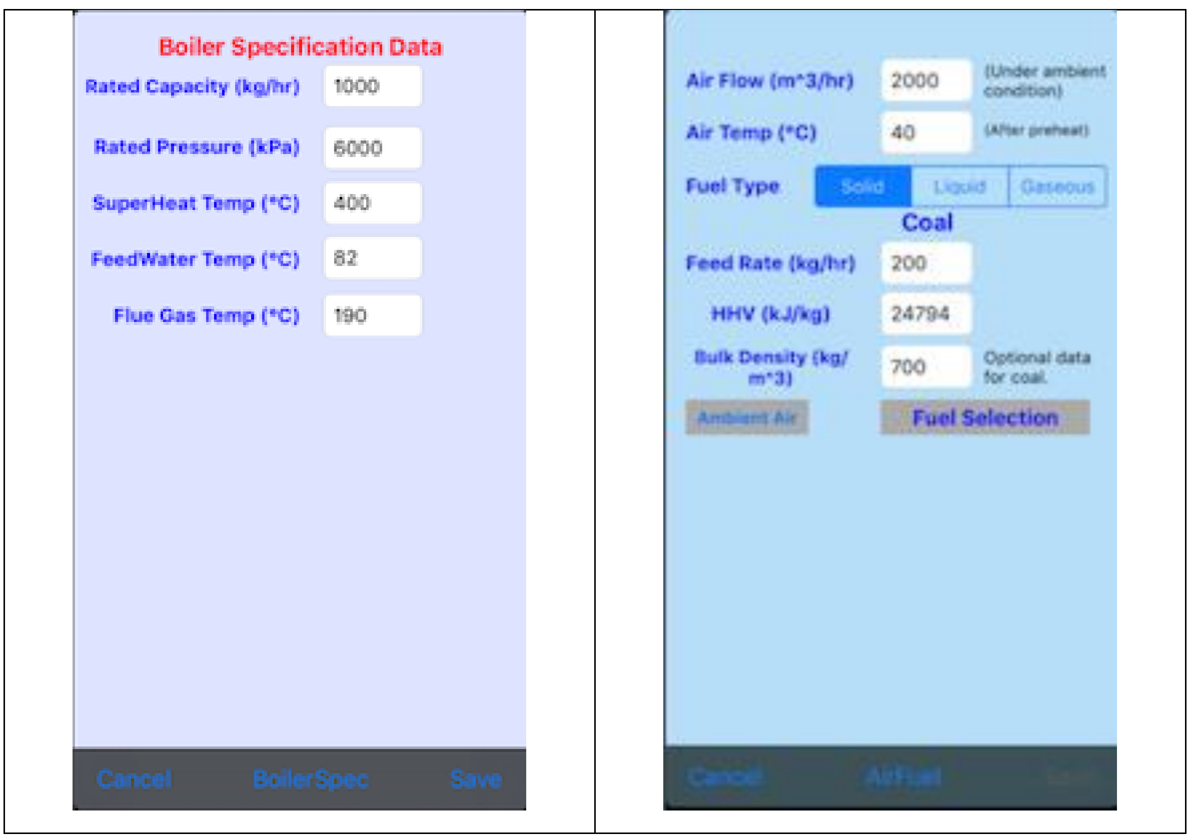
$$\eta_{boiler} = \frac{Q_{in}}{Q_{total}}$$



The individual components of rankine cycle with key design factors are briefly discussed in the following sections.

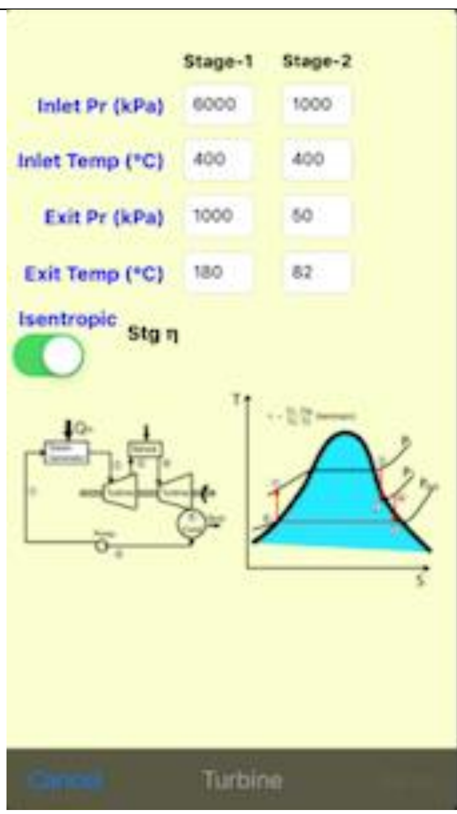
10.2. Boiler and Air-fuel system properties

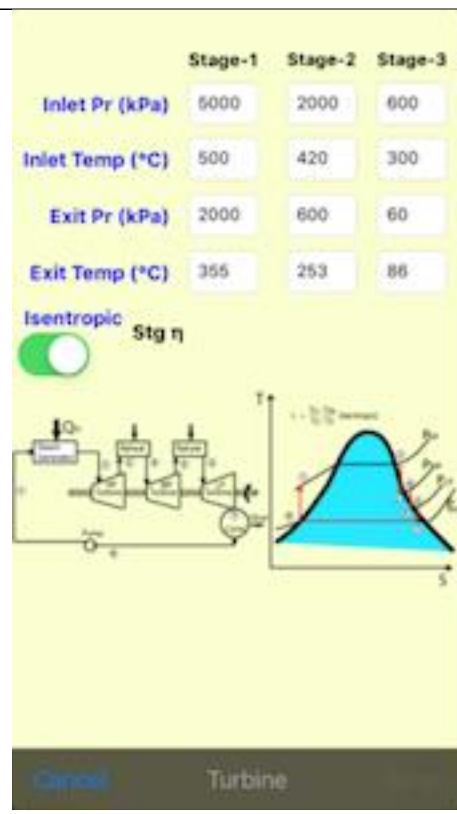
Boiler has several design features such as boiler type, rated capacity, fuel type, size and auxiliaries for air, fuel, and water quality management. This was discussed in section 6.0 in detail with examples to compute the boiler efficiency with heat balance sheet. Fuel properties are depended upon type of fuel such as solid (coal), liquid (oil) and gaseous (Natural gas) used in a specific boiler. Accordingly, right quantity of combustion air needs to be supplied to maximize combustion efficiency. Property data sheet for boiler and air-fuel is given below using snapshot from app suite example.



10.3. Turbine models & state properties

Turbine allows gradual expansion of high-pressure superheated steam over a series of vanes mounted on the turbine rotor that transfers potential energy of the steam to rotor kinetic energy. Accordingly, turbine sections can be classified as high pressure (HP), medium pressure (MP) and low pressure (LP). Sometime these sections are designed and separated such that steam exited from the preceding stage can be reheated before entering in to succeeding stage. Thus, each stage will have its respective inlet and exit pressure and temperature conditions and hence will have its corresponding state properties. Snapshots of turbine properties is taken from app suite examples for a two-stage and three-stage turbine and is shown below.





Expansion of steam inside the turbine is considered as adiabatic (isentropic). Hence there is no change in entropy and isentropic efficiency is considered as 100% i.e., for the change in pressure there is corresponding temperature change due to adiabatic expansion and app suite automatically computes its value and displays for the given exit pressure conditions. However, in practice it is very ideal situation and more often a small change in entropy does occur that is designated by its isentropic efficiency and shown as one of the properties of the stage.

10.4. Condenser and boiler feed water properties

Condensers operate at turbine exit pressure and normally under negative pressure that enable turbine to extract as much energy from the steam. Condensers are mostly shell and tube type heat exchangers and its design parameters have been discussed in section 7.0 with examples. In surface type condensers steam flows through shell side and cooling water flows through the tube side. Cooling water absorbs all residual latent heat energy from the steam to become saturated liquid at condenser operating temperature and pressure. Cooling water generally once through application if it is from river or sea in which case it needs constant monitoring to carry out cleaning and removal of any fouled tubes. Condensate then goes to a hot well from where water is pumped back to the boiler at its working pressure. Property data sheet for the condenser is shown below for a generic condenser, which is then edited for a specific app suite case example with Condenser U value (Overall heat transfer coefficient).

10.5. Thermal efficiency and plant heat rate

Thermal efficiency of rankine cycle is defined as ratio of net energy conversion to the total heat available in the fluid medium (steam) and given as follows.

$$\eta = \frac{W_t - W_p}{Q_{in} - Q_{out}}$$

Where:

$$W_t = Turbine\ output$$

$$W_p = power\ consumed\ by\ feed\ water\ pump$$

Q_{in} = Steam heat input

Q_{out} = Heat rejected

Plant heat rate signifies the net electrical energy generation potential of the rankine cycle for the given amount of combustion heat supplied and given as follows.

$$Plant\ heat\ rate = \frac{Q_{total}}{W_t - W_p}$$

Where:

$Q_{total} = Total\ heat\ energy\ supplied$

$W_t = Turbine\ output$

$W_p = power\ consumed\ by\ feed\ water\ pump$

10.6. Case Examples

10.6.1. Utility powerhouse runs a single stage Rankine cycle that operates a 1000 kg/hour boiler at 6000 kPa pressure and 600 °C super heat temperature. Turbine's exit to condenser is set at 50 kPa. Determine and verify the following using app-suite.

a. Turbine power output, boiler efficiency and the thermal efficiency of the Rankine cycle.
b. Plant heat rate assuming natural gas is fired at 120 m^3/hour that has a gross calorific value (HHV) 49,500 kJ/kg

Data Analysis:

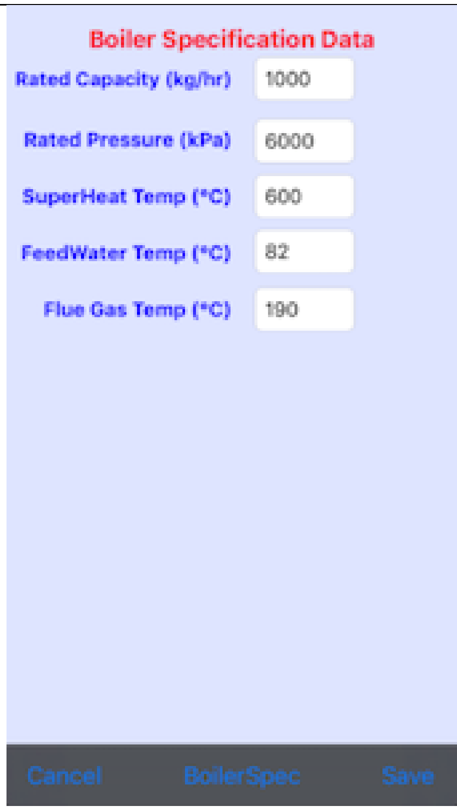
Boiler Data:

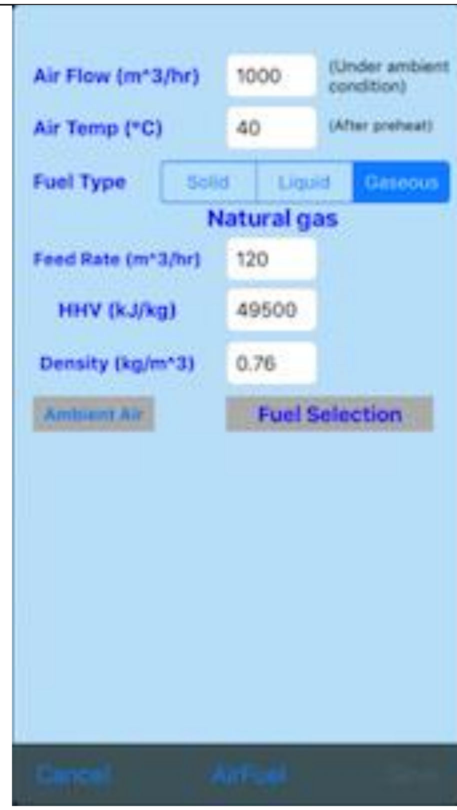
Pressure 6000-kPa, temperatures 400 °C generates steam @1000 kg/hr.

Turbine Data:

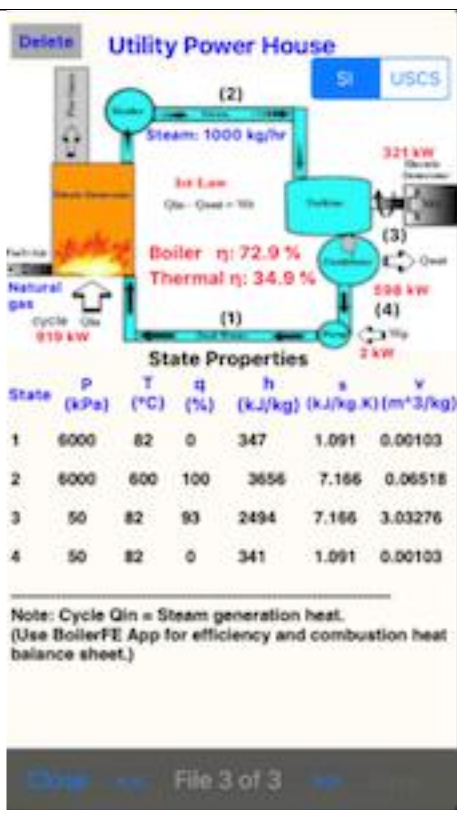
Inlet pressure is at 6000-kPa and temperature 600 °C. Assume adiabatic expansion of the steam that exits at pressure 50-kPa.

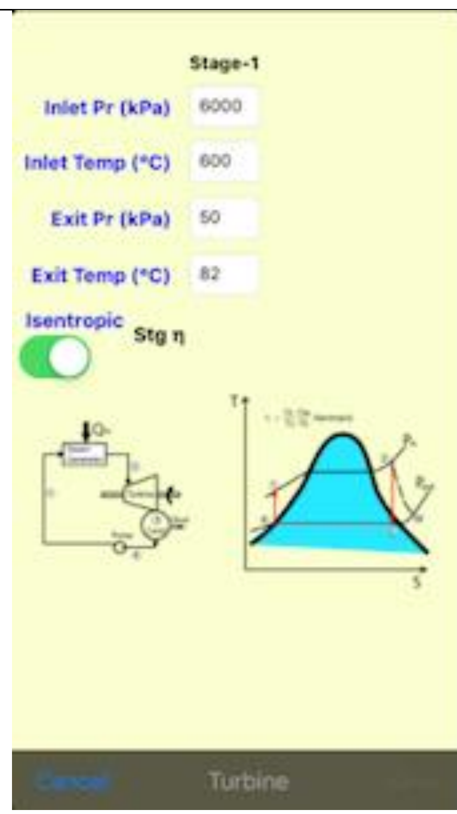
Property data sheet of Boiler and air-fuel is shown below.





The Rankine cycle with state properties and turbine data is given below.





The single stage Rankine cycle has four states and App suite uses the built-in steam table to display steam properties for the given pressure and temperature data for each state shown above.

Steam flow rate m = 1000 kg/hour = 0.277 kg/sec

By applying the first law to the turbine stage 1:

${}_2W_3$ = (h2 – h3)*m = (3656 – 2494) kJ/kg*0.277 kg/s = 321.8 kW

Applying first law to the boiler yields:

Q_{in} = (h2 – h1)*m = (3656 – 347)kJ/kg*0.277kg/sec = 917 kW

The power required by the pump:

W_{pump} = (h6 – h1)*m = [(347-341)*kJ/kg]*0.277*kg/sec = 1.6 kW

Thermal Efficiency $\eta = \frac{W_{net}}{Q_{in}} = \frac{321.8-1.6}{917} = 0.349 = 35\%$

Q_{total} = M_{fuel}*HHV = $\frac{120.0*0.76*kg}{3600*sec} * \frac{49500*kJ}{kg}$ = 1254 kW

Boiler Efficiency = $\frac{Q_{in}}{Q_{total}} = \frac{917}{1254} = 73\%$

$Plant\ heat\ rate = \frac{Q_{total}}{W_t - W_p} = \frac{1254}{321-2} = 3.84$

(Above example is taken from Thermodynamics by Edward.E.Anderson, page 285 on Rankine cycle)

10.6.2. A reheat Rankine cycle operates a 1000 kg/hour boiler at 6000 kPa, the reheat section at 1000 kPa, each turbine inlet is at 400 °C and the condenser at 50 kPa. Determine and verify the following cases using app-suite:

a. Turbine output and rankine cycle thermal efficiency.
b. Plant heat rate assuming coal is fired @ 200 kg/hr that has a gross calorific value (HHV) 24,794 kJ/kg.

Data Analysis:

Boiler data:

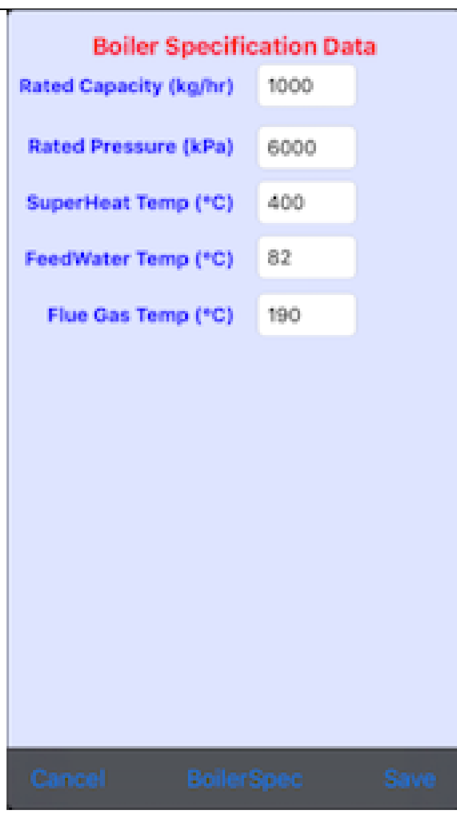
Pressure 6000-kPa temperatures 400 °C generates steam @1000 kg/hr.

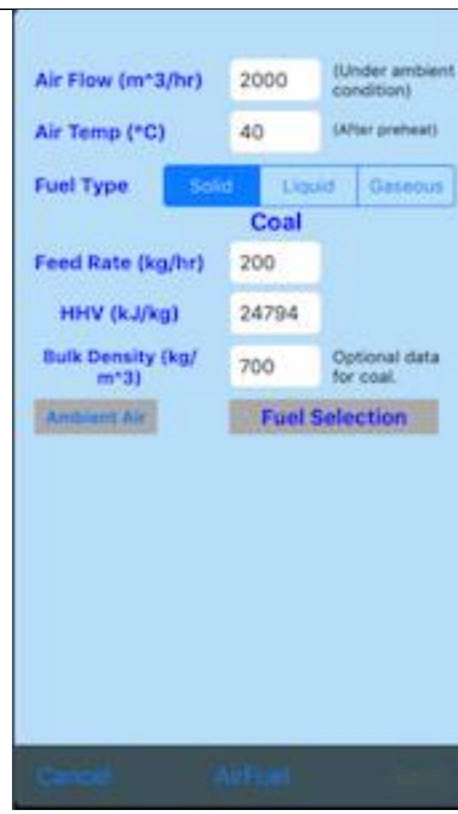
Turbine: Assume adiabatic (isentropic) expansion for each stage.

Stage 1 inlet pressure 6000 kPa and temperature 400 °C

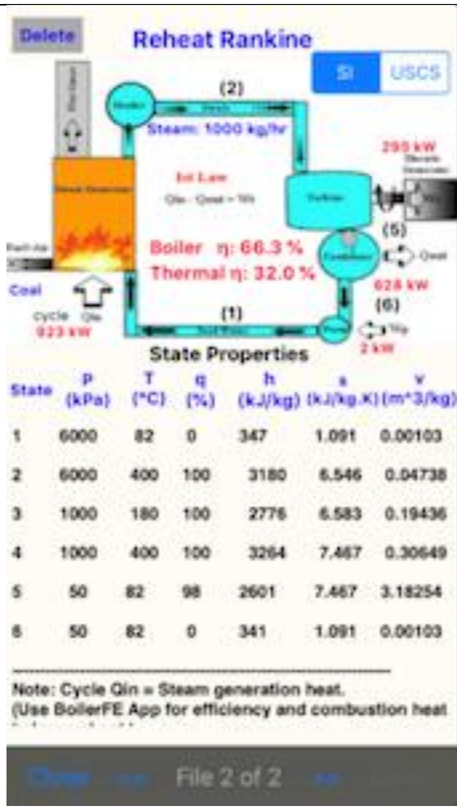
Stage 2 inlet pressure 1000 kPa and 400 °C

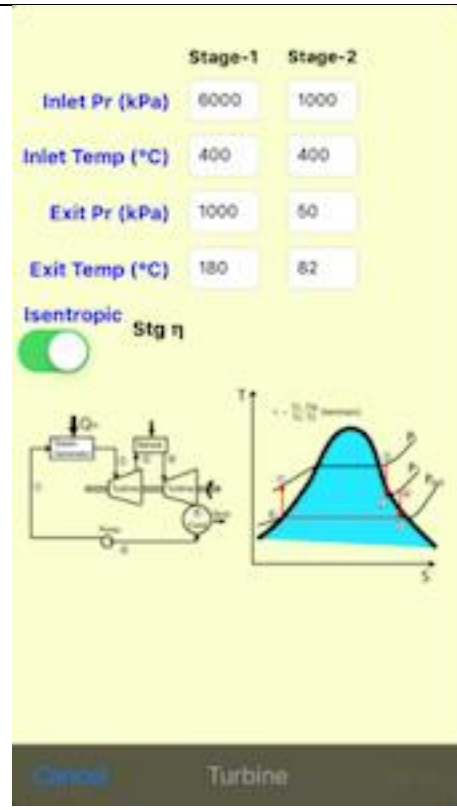
Condenser inlet pressure 50 kPa. Property data sheet for the boiler and air-fuel is given below. Using default data if data is not specified.





The rankine cycle with state properties and turbine data is given below.





The above reheat rankine cycle has total six states including two states for the turbine stages. Accordingly, state properties are shown in a tabular form so analysis can be done using the respective property data. App suite uses the built-in steam table shown in section 2 on thermo-physical properties to display steam properties for the given pressure and temperature data for each state.

Steam flow rate m = 1000 kg/hr = 0.277 kg/sec

By applying the first law to the turbine stage 1 and stage 2 we get:

$_2W_3$ = (h2 – h3)*m = (3180 – 2776) kJ/kg*0.277 kg/s = 111.90 kW

$_4W_5$ = (h4 – h5)*m = (3264 – 2601) kJ/kg * 0.277 kg/s = 183.6 kW

Total turbine output W_t = $_2W_3$ + $_4W_5$ = 295.5 kW

Applying first law to the boiler yields:

Q_{in} = (h2 – h1)*m + (h4 – h3)*m = (3180 – 347)*m + (3264 – 2776)*m

Q_{in} = 2833*0.277 kW + 1488*0.277 kW = 784.7 + 135.1 = 920 kW

The power required by the feed water pump:

W_{pump} = (h6 – h1)*m = (347 – 341)*0.277 kW = 1..66 /η = 2.0 kW

Thermal Efficiency $\eta = \frac{W_{net}}{Q_{in}} = \frac{293.5}{920} = 0.319 = 32\%$

Q_{total} = M_{fuel}*HHV = (200 kg/hr)*(24794 kJ/kg) = 1378 kW

Boiler Efficiency = $\frac{Q_{in}}{Q_{total}} = \frac{920}{1378} = 66.6\%$

$Plant\ heat\ rate = \frac{Q_{total}}{W_t - W_p} = \frac{1378}{295-2} = 4.7$

Note: The above example of Reheat Rankine cycle is taken from a textbook on “Thermodynamics” by Edward. E. Anderson, Page 287.

11. Combined Heat and Power (CHP) Cycle

11.1. Generic Design Analysis Model

The combined heat and power cycle often referred as cogeneration cycle that uses either steam turbine or gas turbine to generate power along with heat in the form of pressurized steam that is required for the downstream chemical and process equipment.

A typical Rankine cycle shown below has one steam extraction point from a condensing turbine for the process equipment at certain designated pressure and temperature conditions. Initially the high-pressure steam is expanded and used until extraction point and then part of the steam is used for power generation and part of the steam goes for the process application.

Q_{in}= Steam heat input

$Q_{process}$ = Extraction

Q_{out} = Heat rejected

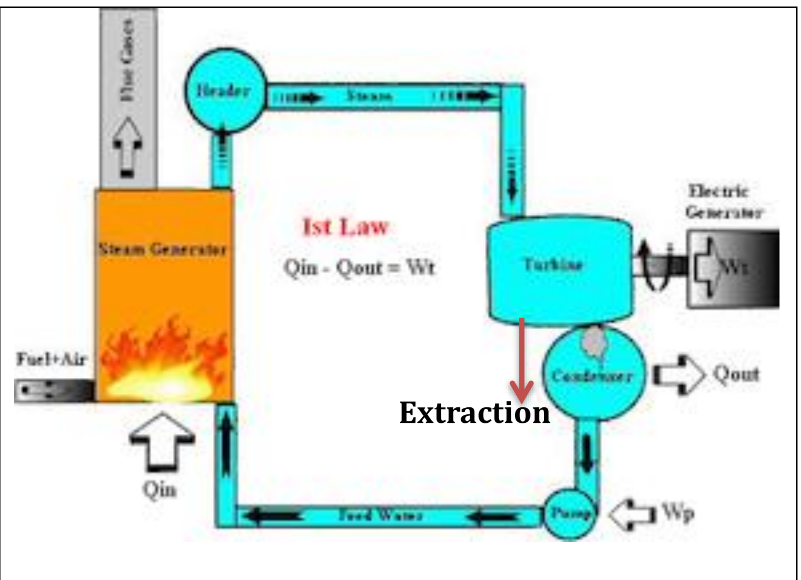
W_t = Power output

$Q_{total} = M_{fuel} * HHV$

HHV=High Heat Value

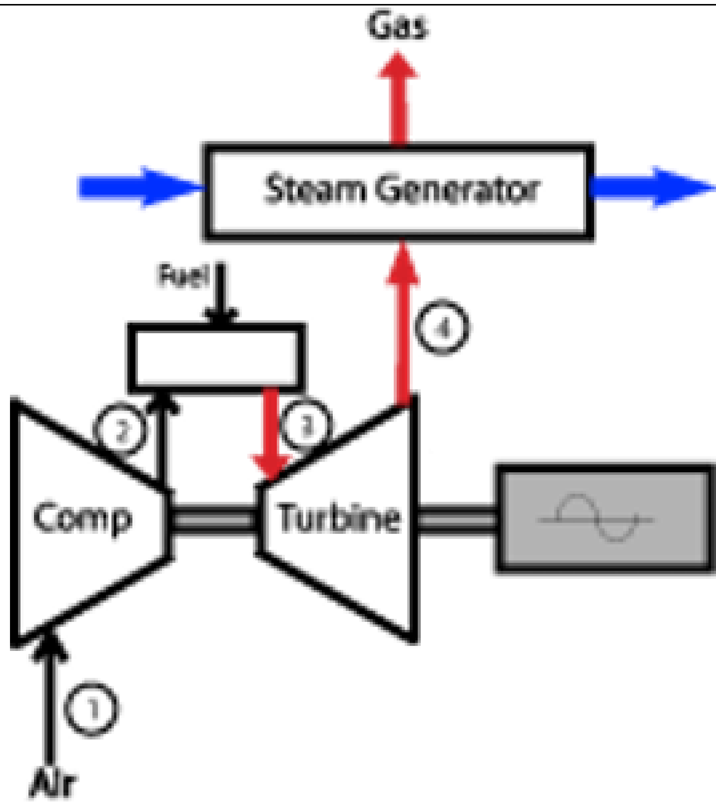
Boiler Efficiency:

$$\eta_{boiler} = \frac{Q_{in}}{Q_{total}}$$



Performance analysis and cycle efficiencies of Rankine cycle explained in the preceding section can be applied in the above case as well. Therefore, this section mainly focuses on Brayton cycle using gas turbine and combustion gases for generating steam in a heat exchanger.

A typical Brayton cycle model shown below is a model of a simple gas-turbine engine. Atmospheric air is compressed and mixed with fuel in a combustion chamber at a fixed pressure and then expand the combustion gases in a turbine to generate power. The exhaust gases are then sent to a boiler where process steam is generated and supplied to the downstream equipment.

Applying first law to the steady flow turbine work: 3w4 = $C_p (T_3 - T_4)$ Applying first law to the steady flow compressor: 1w2 = $C_p (T_1 - T_2)$ Applying first law to the combustion chamber: 2q3 = $C_p (T_3 - T_2)$ Applying first law to the heat rejection in boiler: 4q1 = $C_p (T_4 - T_{1a})$	

11.2. CHP (Brayton) state properties

Four states and the process of above Brayton cycle are given below.

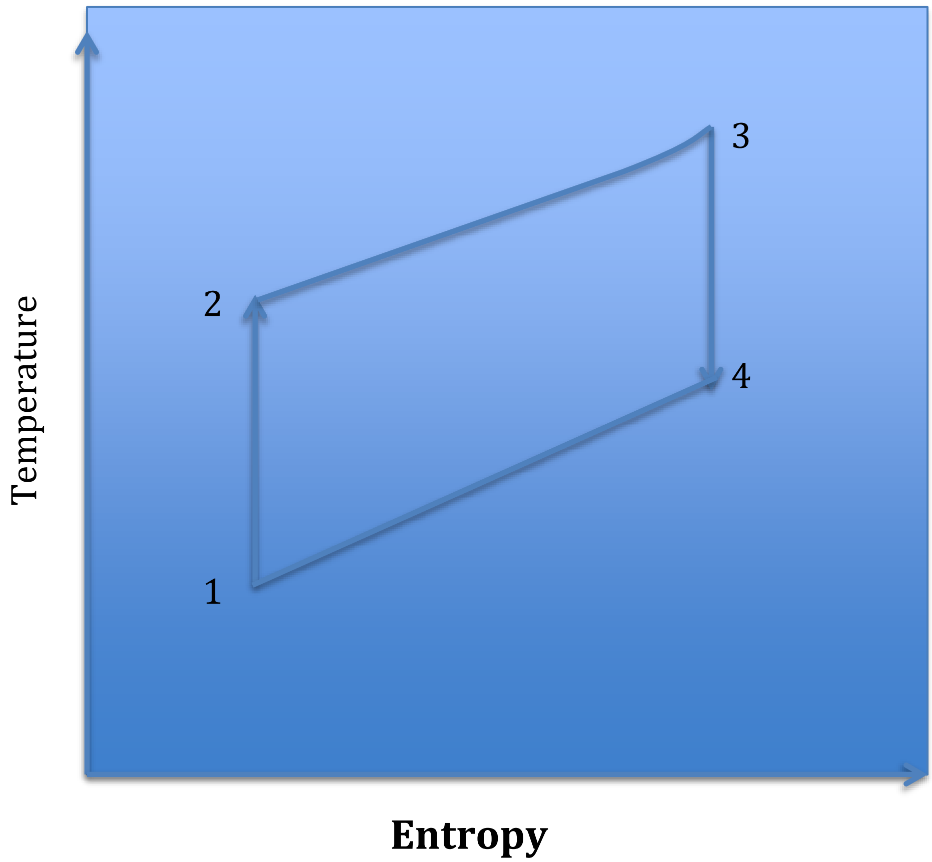
1-2 Isentropic compression of the gas in a steady flow compressor.

2-3 Fuel combustion process in a combustion chamber

3-4 Isentropic expansion of the gas in a steady-flow turbine.

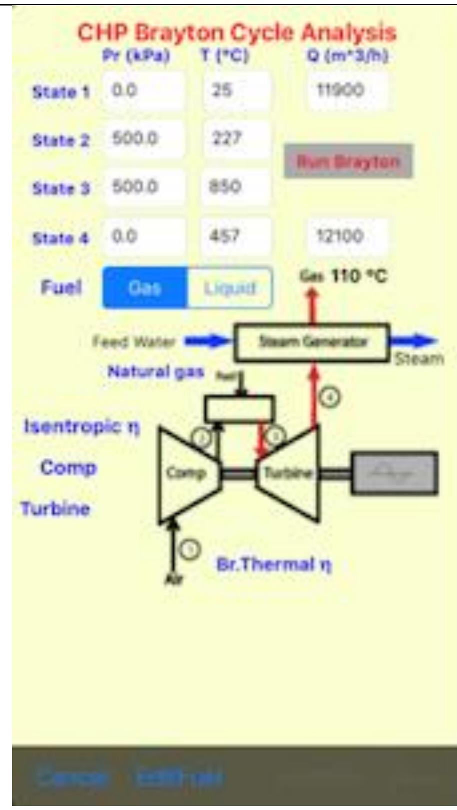
4-1 Rejection of heat from the gas in a steam generator (boiler).

The temperature-entropy state diagram shown below illustrates the above process that occurs in the Brayton cycle.



The combustion process 2-3 is isobaric process corresponding to compressor exit pressure. Similarly, turbine 4-1 is also isobaric process at turbine exit pressure.

App suite property data sheet is shown for the steam generator (Boiler) and Brayton cycle with default values. This can be edited for more specific application and result can be then saved for further analysis.

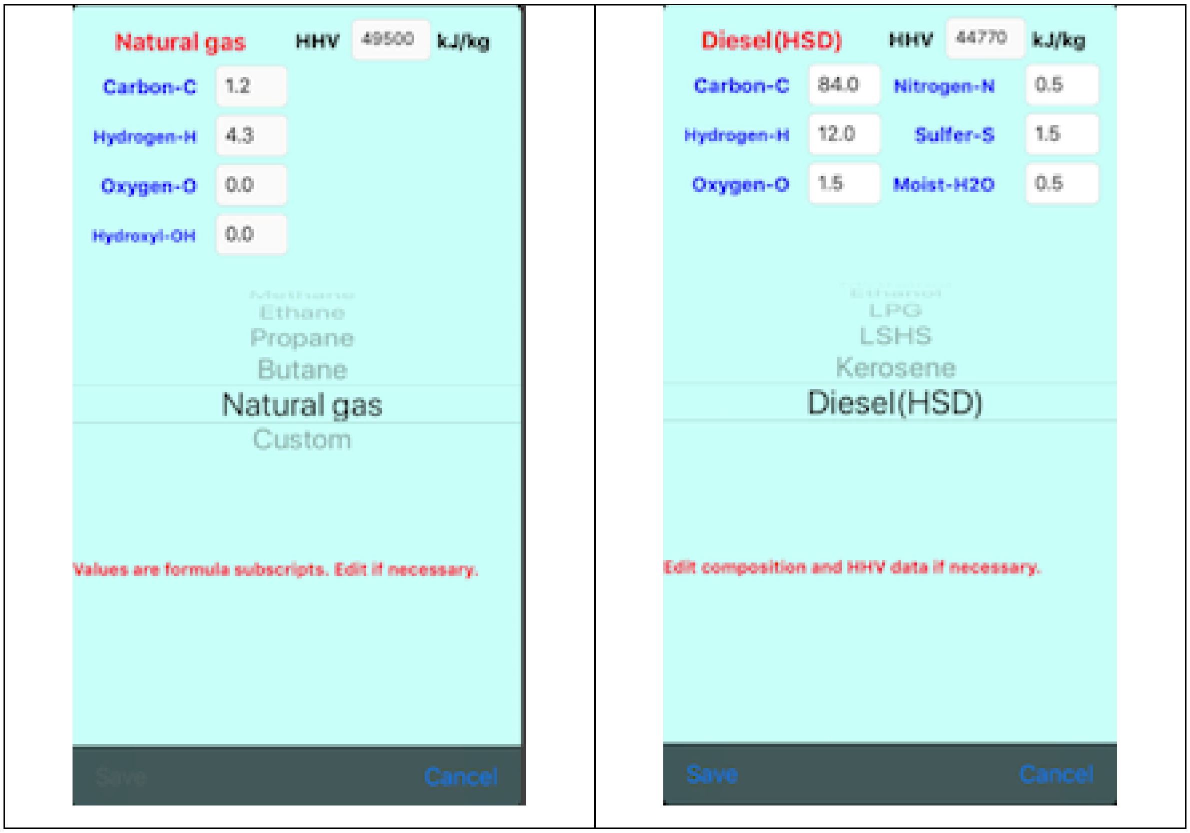


Air enters the turbo compressor at ambient temperature and pressure conditions and flow rate is shown under state 1 properties. Similarly, all four states properties are defined. Fuel enters the chamber mixes with air and combustion takes place under constant pressure as indicated in state2 and 3 with increase in temperature.

Fuel properties are defined for the most common type of fuel such as gas or liquid and given below. App suite provides the properties for these fuels with composition and high heating values (HHV).

Gas: Methane, Ethane, Propane, Butane, Natural gas and custom.

Liquid: Furnace oil (#1 to #6), Methanol, Ethanol, LPG, LSHS, Kerosene and Diesel.



The gases that are not in the list one can use it with custom properties by editing the composition as well as its high heating value.

11.3. CHP (Brayton) performance summary

Brayton cycle components are all steady flow devices and applying first law we get:

Work done by the compressor (isentropic) $_1w_2 = C_p\,(T_1 - T_2)$

Heat added in the combustion (isobaric) $_2q_3 = C_p\,(T_3 - T_2)$

Work done by the turbine (isentropic) $_3w_4 = C_p\,(T_3 - T_4)$

Heat rejected in the boiler (isobaric) $_4q_1 = C_p\,(T_4 - T_{1a})$

Thermal efficiency of the Brayton cycle can be written as:

$$Thermal\ \eta = 1 - \frac{4q1}{2q3} = 1 - \frac{T4 - T1}{T3 - T2}$$

Rewriting above expression for isentropic expansion

$$Thermal\ \eta = 1 - \frac{1}{r^{\frac{k-1}{k}}}$$

Where:

Pressure ratio r = P2/P1

Specific heat ratio k = Cp/Cv

Isentropic expansion: $\frac{T2}{T1} = r^{\frac{k-1}{k}} = \frac{T3}{T4}$

From the above expression one can deduce that thermal efficiency of the Brayton cycle is a monotonic increasing function of the pressure ratio of the turbo compressor.

The above expression is under ideal operating conditions and in practice the actual temperature rise after compression could be higher than the theoretical adiabatic condition. Similarly, the turbine expansion may not be truly adiabatic. This is designated as isentropic efficiency of the compressor and turbine and given as:

$$Isentropic\ \eta = \frac{Theoretical\ temperature\ change}{Actual\ tempearture\ change}$$

Based on the mass flow rate of the air one can generate an ideal heat balance sheet of a CHP cycle as given below.

Total heat input = m*Cp*(T3 – T1)

Turbine output = m*Cp*(T3 –T4)

Steam generation heat: m*Cp*(T4 – Texit)

Flue gas heat: m*Cp*(Texit – T1)

11.4. CHP (Brayton) Case Examples

11.4.1. A CHP plant uses natural gas to generate power and process heat at 15 bar and 220 °C temperature. Ambient air enters compressor at 25 °C and discharges at 500 kPa-gage and 227 °C to the combustion chamber. Turbine inlet and exit temperatures are at 850 °C and 457 °C. If boiler flue gas exit temperature is at 110 °C determine:

- Brayton cycle performance
- Flue gas flows at 200 m^3/min determine the net power
- Steam generation potential with CHP heat balance sheet.

Given:

Air flow rate: $\frac{12000*m^3}{3600*sec} * \frac{1.2*kg}{m^3} = 4.0\ kg/sec$

Specific heat ratio: k = 1.4; Specific heat capacity Cp = 1.0 kJ/kg

Pressure ratio: $r = \frac{(500+101.3)\ kPa}{101.3\ kPa} = 5.94$

Isentropic Efficiencies:

$$T_{2s} = T_1 * r^{\frac{k-1}{k}} = (273.15 + 25) * r^{\frac{0.4}{1.4}} = 495.9$$

$$Compressor\ \eta_s = \frac{T_{2s} - T_1}{T_2 - T_1} = \frac{197.77\ K}{202\ K} = 0.98$$

$$Turbine\ \eta = \frac{T_3 - T_4}{T_3 - T_{4s}} = \frac{1123.15 - 730.15}{1123.15 - 675.08} = 0.88$$

Brayton Thermal Efficiency:

$Brayton\ \eta = 1 - \frac{1}{r^{\frac{k-1}{k}}}$= 1 – 0.731 = 0.399 = 39.9%

Heat balance:

Total heat input: m*Cp*(T3 – T1) = 3300 kW

Turbine output Tw: m*Cp*(T3 –T4) = 1572 kW

Compressor work Cw: m*Cp*(T2-T1) = 808 kW

Net Power: $T_w - C_w$ = 1572 – 808 = 764 kW

Steam generation heat: $m*Cp*(T_4 - T_{exit})$ = 1388 kW

Flue gas heat: $m*Cp*(T_{exit} - T_1)$ = 340 kW

Turbine + Steam + Flue gas heat = 1572 + 1388 + 340 =3300 kW

Steam generation potential:

From app suite steam table for properties of 1601 kPa, 220 °C steam:

Super heat enthalpy hs = 2843 kJ/kg

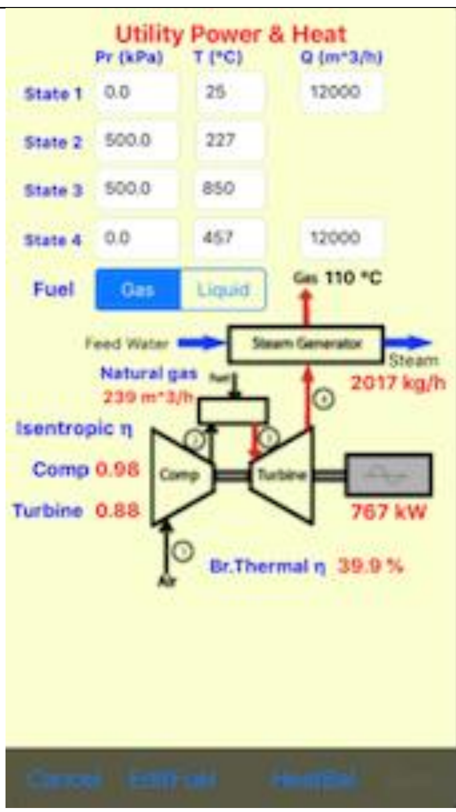
Enthalpy of saturated Water at 201 °C , hw: 856.9 kJ/kg

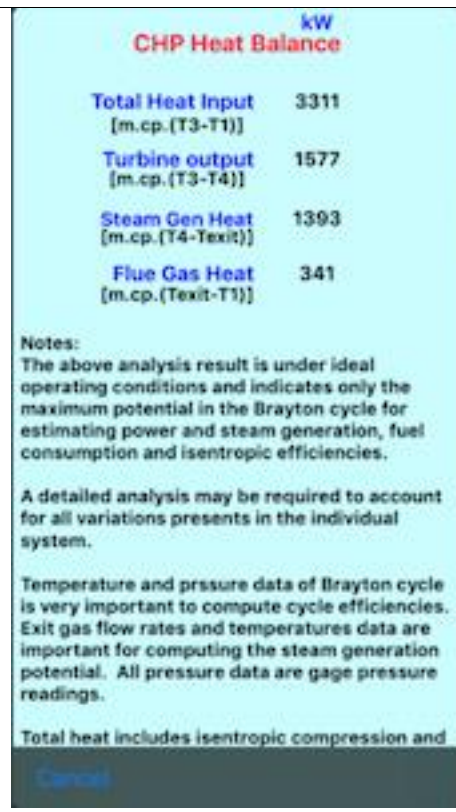
Sensible heat h1 = $Cp*(T_{sat} - T_{feed\ water})$ = 4.20(201 – 82) = 500 kJ/kg

Enthalpy / kg of steam = (2843 – 856.9) + 500 = 2485 kJ/kg

Steam generation potential = 1388/2485 = 0.56 kg/sec = 2017 kg/hour

App suite analysis:





11.4.2. A Brayton gas and Rankine steam power plant work in tandem to generate power. Ambient air enters the Brayton cycle at 80 °F. The compressor has a pressure ratio of 8:1, and maximum combustion chamber temperature is 2000 °F. Turbine exhaust gases leave the heat exchanger at 500 °F. Steam is generated at 900 psia pressure and 1100 °F enters the steam turbine and exit to a condenser at 10 psia. Determine the performance of both Brayton and Rankine cycle under ideal conditions assuming unit of air mass entering the compressor determine mass of steam and heat added and heat rejected with overall thermal efficiency.

Given State Properties:

Brayton 1 – 2- 3 - 4

State 1: 14,7 psi, 80°F (101.3 kPa, 26.7 °C) r = 8

State 3: 117.6 psi, 2000°F (811 kPa, 1094 ° C)

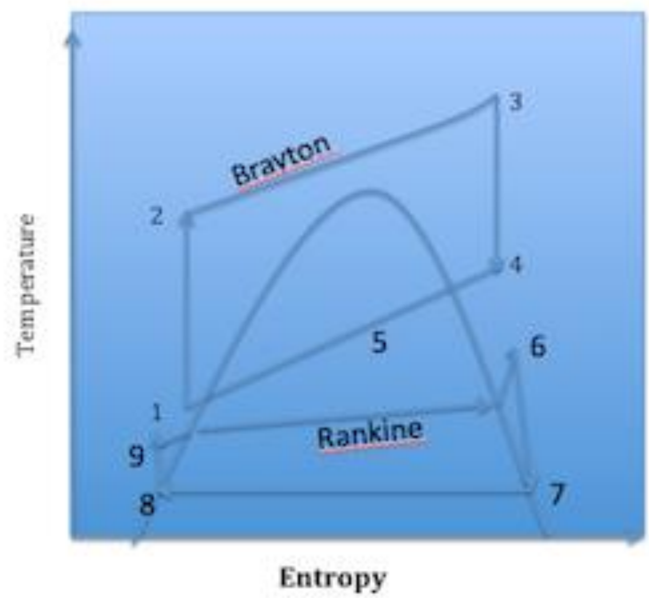
State 5: 14.7psia, 500 °F

5 ->6: Heat Exchanger

Rankine 6 – 7 – 8 – 9

State 6: 900 psia, 850 °F (6205 kPa, 593.3 °C)

State 7: 10 psia



(Ref: Thermodynamics by E. E. Anderson, page 409, with *correction.)

Following table summarizes state properties of both Brayton and Rankine cycle applying ideal gas isentropic conditions for compression and expansion for Brayton and isentropic expansion for Rankine cycle.

State	**P**	**T**	**h**	**s**
1	14.7 psia (101.3 kPa)	540 °R (300°K)		
2	117.6 psia (978 kPa)	978 °R (543 °K)		
3	117.6 psia (978 kPa)	2460 °R (1367 °K)		
6	900 psia (6206 kPa)	1100 °F (593 °C)	1564.4 Btu/lbm	1.703 Btu/lbm. °R
7	10 psia (69 kPa)	193 °F (89.6 °C)	1088.4 Btu/lbm	1.703 Btu/lbm. °R
8	10 psia		161.3 Btu/lbm	
9	900 psia		163.9 Btu/lbm	

Assuming M_{gas} = 1 lbm/sec = $\frac{M_{gas}}{\rho}\frac{60.sec}{min} = \frac{60}{0.075} = 800\ cfm$

$$M_{steam} = \frac{h_4 - h_5}{h_6 - h_9} = \frac{C_p(T_4 - T_5)}{h_6 - h_9} = \frac{0.24(1150 - 500)}{1565 - 163.9} = 0.113\ lbm$$

Applying first law to the Brayton cycle net heat added to the CHP cycle:

$$2Q3 = h_3 - h_2 = C_p(T_3 - T_2) = 0.24(2460 - 978) = 355\frac{Btu}{lbm - gas}$$

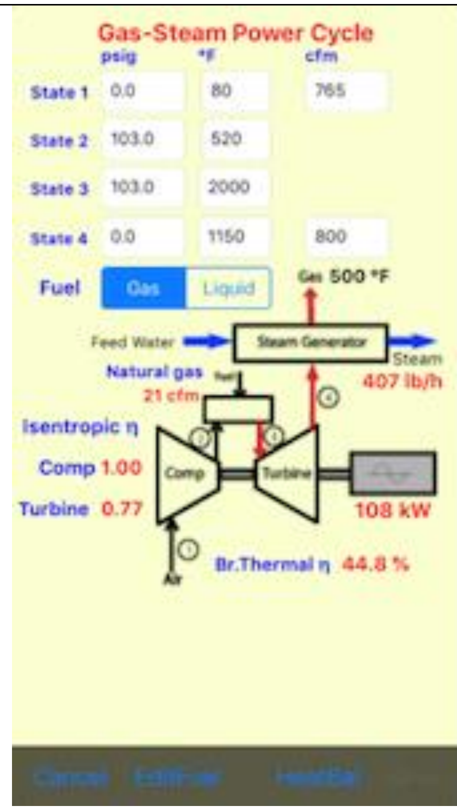
Heat rejected in Brayton cycle: $5Q1 = C_p(T_1 - T_5) = -101\frac{Btu}{lbm}$

Similarly heat rejected in Rankine cycle:

$$7Q8 = M_{steam}(h_8 - h_7) = 0.113(161.3 - 1088.4) = -105\ Btu/lbm$$

Overall Thermal efficiency $\eta = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{105+101}{355} = 0.42$

App Suite Analysis of Brayton:



Steam generation potential = 453 lbm/hour = 0.113 lbm/sec which is consistent with above analysis.

***Correction:**

In the reference book T4 is computed for isentropic expansion in the turbine as: $T_4 = 2460 * \left(\frac{14.7}{117.6}\right)^{\frac{0.4}{1.4}} = 1360\ °\mathrm{R} = 900\ °\mathrm{F}$ and calculated

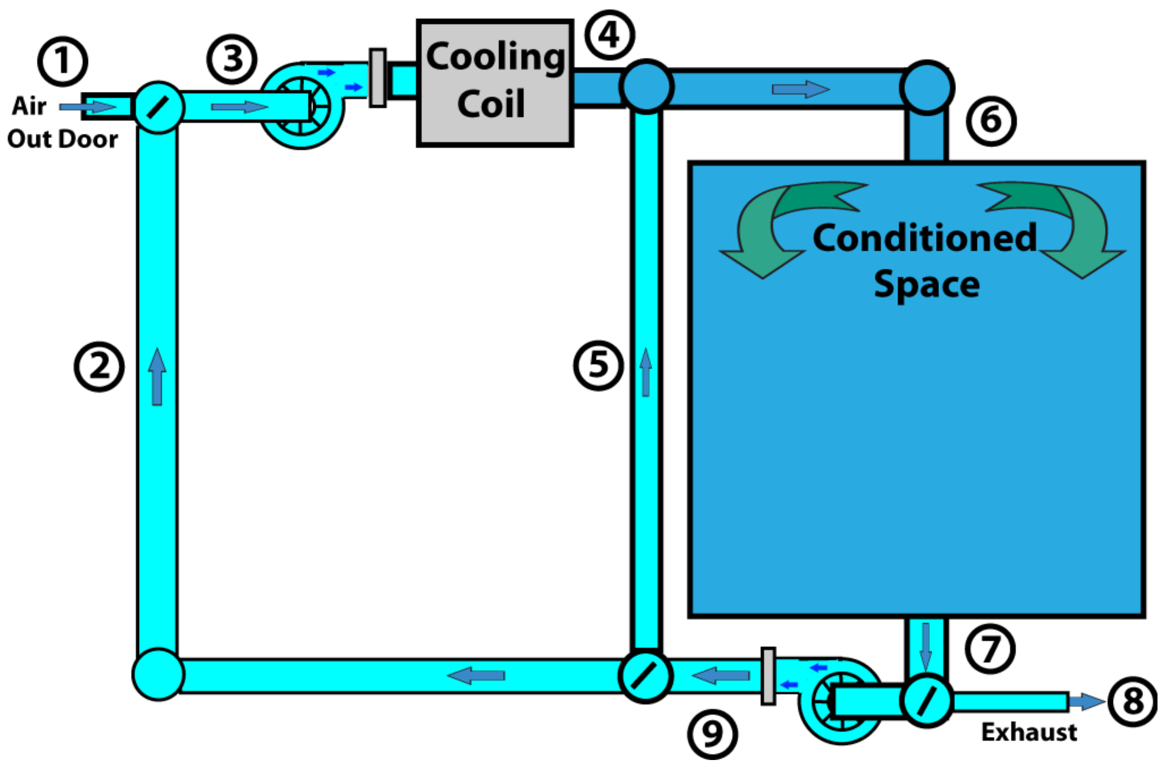
$$M_{steam} = \frac{h_4 - h_5}{h_6 - h_9} = \frac{C_p(T_4 - T_5)}{h_6 - h_9} = \frac{0.24(1360 - 500)}{1565 - 163.9} = 0.147\ lbm$$

Here is the mistake T_4 is 1360 °R (900 °F) and T_5 is 500 °F. Therefore, made correction for T_4 as 1150 °F gas turbine exit temperature since steam is at 1100 °F. Hence Turbine isentropic efficiency is 77%.

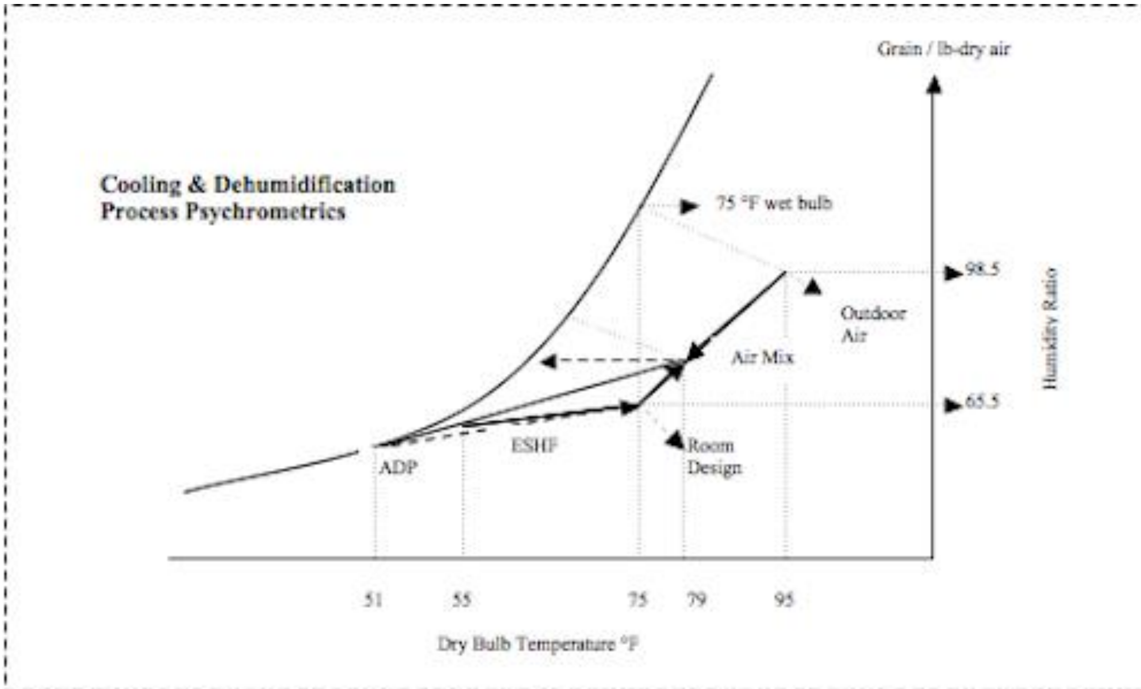
12. HVAC Cooling Cycle

12.1. Generic Design Analysis Model

The generic cooling cycle model consisting of conditioned space (office, residential or shopping mall etc.), outdoor ventilation air is mixed with return air and then passed through the cooling coil as shown below. The conditioned space is maintained at ASHRAE recommended comfort level such as 77 °F dry-bulb temperature with 50% relative humidity. The circuit is also consisting of dampers and mixing points to control the mass flow rates. State numbers are assigned in sequence from outdoor air to return air. Part of the return air is bypassed the coil to maintain a constant temperature differential of (20 °F) with respect to the conditioned space. (Refer Appendix 15.2.8 ASHRAE psychrometric)



Humid air properties (Section 2.5) of each state with psychometric process with typical data of a generic model are described below.



In the above process outdoor ventilation air at 95 °F (35 °C) dry bulb and 75 °F (23.9 °C) wet bulb mixes with the return air from the conditioned space maintained at 75 °F dry bulb and 50% humidity. Depending upon mass flow of the return air the resulting mix enters the cooling coil at 79 °F. The dew point temperature of this resulting mixture depends upon the amount of heat it needs to release along with reduction of moisture content from the outdoor air to maintain the temperature and humidity level of the conditioned space. This is a two-step process.

Step 1: Initially sensible cooling of the mixture take place that drops the dry bulb temperature until hits the saturation curve of psychrometric chart shown by the horizontal dotted line.

Step 2: Once reaching saturation curve any further drop in the temperature takes place along this curve by condensing moisture from the mixture reducing the humidity ratio as well. This dehumidification process takes place until it hit the apparatus dew point (ADP) of the cooling coil. Therefore, technically temperature and humidity ratio of the exiting air is same as ADP. However, since a small quantity of outdoor air mix bypasses the coil that increases the temperature and humidity ratio based on percent bypass, which is one of the design factors. The other design factors used in the analysis are listed below.

Room Sensible Heat Factor

It is the ratio of room sensible heat and room total heat (sensible + latent heat due to moisture content) added inside the conditioned space. This heat load is estimated under 10 different categories covered in the next section 12.2.

Outside Air Total Heat (OATH)

This is the total sensible and latent heat added through supply of ventilation air and other infiltrations that occurs in the conditioned space. There are industry standards for supply of ventilation air.

Effective Sensible Heat Factor (ESHF)

$$ESHF = \frac{Room\,Sensible\,HeatLoad\,(RSH) + cbf \cdot Outside\,Air\,Sensible\,Heat(OASH)}{Room\,Total\,Heat\,(RTH) + cbf \cdot Outside\,Air\,Total\,Heat(OATH)}$$

Where:

cbf – coil bypass factor (5 to 15% of total air entering the cooling coil)

$$ESHF = \frac{cp\left(T_{room} - T_{adp}\right)}{cp\left(T_{room} - T_{adp}\right) + h_g\left(w_{room} - w_{adp}\right)}$$

Hence T_{adp} is solved iteratively for the given ESHF of the conditioned space and for the given coil bypass factor.

Grand Total Heat (GTH)

This is the total cooling load of the HVAC cooling system based on the total room heat (RTH) load and out door ventilation air (OATH) and usually specified in Tons in USCS system and kW in SI system:

GTH = RTH + OATH

1Ton = 12000 BTU/hour

1 kW = 3412 BTU/hour

12.2. Cooling Load Estimation

HVAC cooling loads are estimated using following 10 broad categories.

1. Solar Heat Gain Through Glass
2. Heat Gain Through Wall Conduction
3. Heat Gain Through Roof Conduction
4. Heat Gain Through Floor Conduction
5. Heat Gain Through Ceil Conduction
6. Heat Gain Through Door Conduction
7. Heat Gain Through Partition Conduction
8. Internal Heat Gain - People
9. Internal Heat Gain – Electrical
10. Infiltration

Categories 2 to 7 contribute mainly to sensible heat gain by simple conduction, which is function of material properties used in the respective component and temperature differential of outside and conditioned space. Categories 8 to 10 contribute to both sensible and latent heat gains. This is briefly described in the following sections in respective groups.

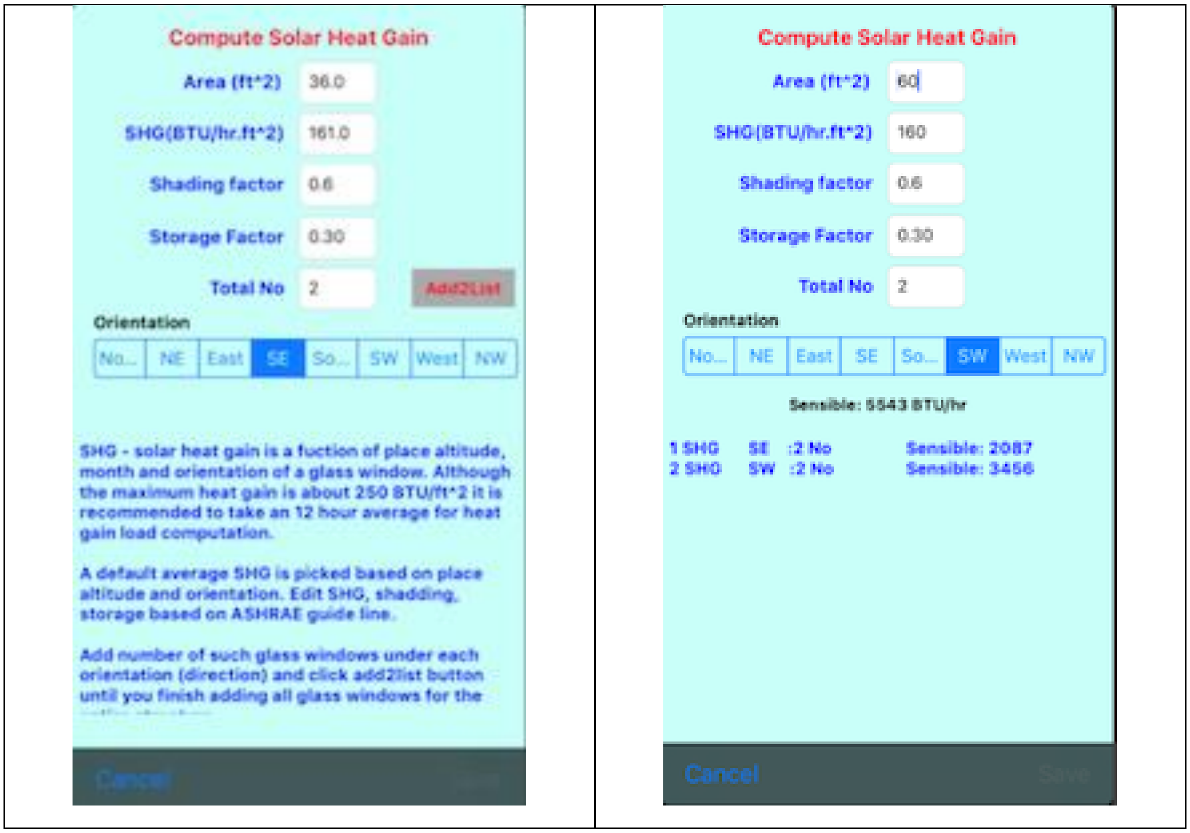
A sample cooling load summary with data sheet taken from app-suite.

12.2.1. *Solar Heat Gain*

The solar heat gain through glass depends upon the solar intensity during the design month, time, and orientation of the glass structure. One can also use the average intensity factor with individual glass area. Thus, total solar heat gain is calculated using following equation. ASHRAE handbook provides data for different storage and shading factors.   However, one can also use default values provided in the app suite for general estimation.

$$Soalr\ Heat\ Gain\ (SHG) = Area * Solar\ intensity * storage * shading$$

A sample solar heat gain data sheet is given below from app-suit.



12.2.2. Conduction Through *Wall/Roof/Floor/Ceil/Partition/Door*

The heat gain through conduction is primarily dependent upon the overall heat transfer coefficient (U) of the respective component, and the ELTD (Equivalent Load Temperature Difference). Where ELTD is a function of orientation, month and time and ASHRAE handbook gives this value along with recommended U value. However, app suite provides default values and heat gain is calculated as:

$$Heat\ gain = U.A.ELTD$$

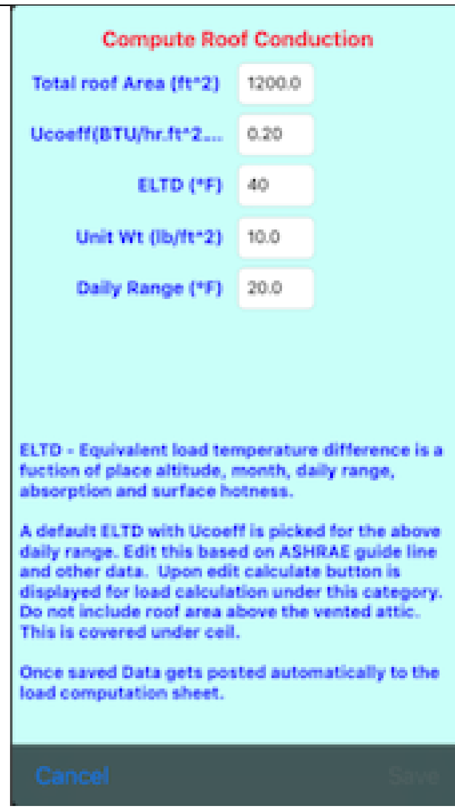
If the wall/roof/floor is a composite material structure, then the heat transfer coefficient can be obtained by adding individual material resistances to obtain the conductance. For example, a wall consists of 4" common brick with resistance R1+0.25" cement plaster with resistance

R2+ two plaster board of 0.5" thick with resistance R3 and 2" air gap with resistance R4. The overall conductance U coefficient is given as:

$$U = \frac{1}{R_1 + R_2 + R_3 + R_4}$$

ASHRAE handbook provides resistance value for the most common construction materials with its bulk density to obtain the overall conductance of any composite structure.

A sample conduction data sheet for wall and roof is shown below.

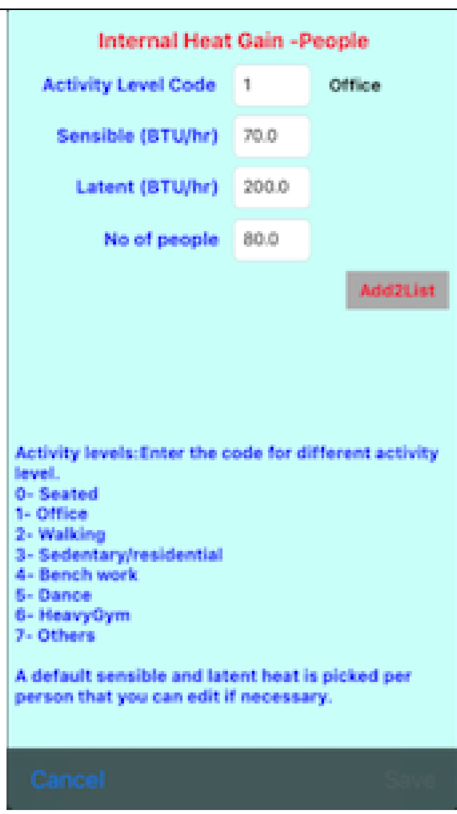


12.2.3. Internal Heat Gain – People

Heat gain through people within the space is mainly due to the rate of metabolism in human body. Though this rate varies from person to person, a general recommended value of both sensible and latent heat for different activity levels are given below for one person.

Activity Level	Sensible Heat Gain Btu/hour (kW)	Latent Heat Gain Btu/hour (kW)
Seated	225 (0.065)	200 (0.059)
Office	240 (0.07)	275 (0.081)
Residential	300 (0.088)	480 (0.141)
Bench work	300 (0.088)	510 (0.149)
Dance	320 (0.094)	520 (0.152)
Heavy Gym	635 (0.186)	920 (0.269)

A sample data sheet for internal heat gain by people is shown below.



Total heat gain is calculated depending upon the number of people engaged in different activity levels. App suite uses the above values and users can refer to other guidelines to compute the total internal heat gain under this category.

12.2.4. Heat Gain – Electrical Appliances

Electrical appliances used within conditioned space can contribute substantial heat starting from simple fluorescent lamp to electrical motors. There are seven most common appliances given below with recommended values for both sensible and latent heat for calculating the total heat gain.

Appliance	Sensible Heat Gain Btu/hour (kW)	Latent Heat Gain Btu/hour (kW)
Lamp	100 (0.030)	
Fluorescent Tube	154 (0.036)	
Toaster	1500 (0.440)	
Coffee maker	3500 (1.02)	400 (0.117)
Kettle	1600 (0.47)	2450 (0.718)
Fryer	4000 (1.17)	2050 (0.60)

12.2.5. Heat Gain – Infiltration

Infiltration through the structure depends upon the ambient conditions as well as wind speed. The pressure differential is sufficient to cause flow across various joints and gaps in the structure thereby adding additional heat loads. The basic assumption is whatever air comes in escapes out there by contributing only to the heat while flow rate remains unchanged. There are empirical formulae to compute the heat gain due to infiltration available in standard handbook such as ASHRAE and other HVAC reference documents. Here is one for a commercial building with sensible and latent heat gain using standard computation formulae. Total filtration area is computed for any given structure based on area of windows, doors and part of the structure that is susceptible for leakage air.

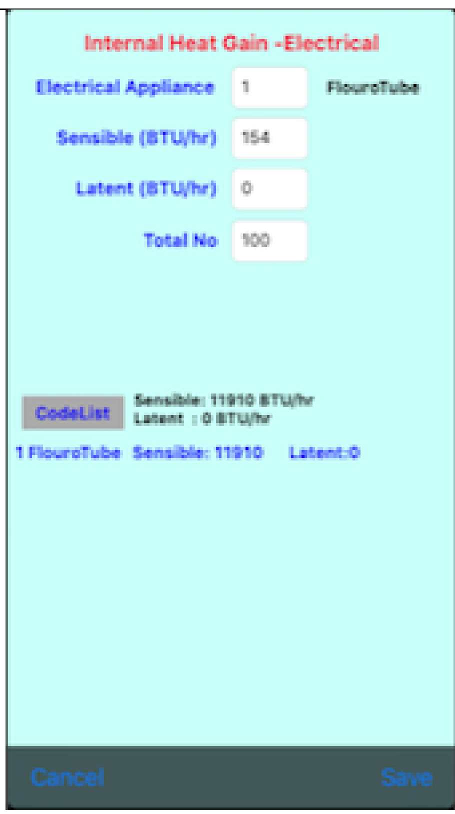
Infiltration rate q: 1.8 cfm/ft^2

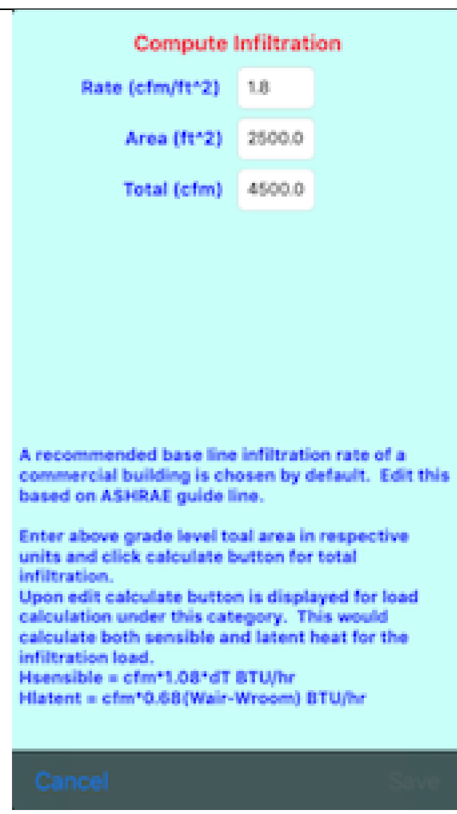
Qair = Total area * q

Sensible heat gain = Qair*1.08*ΔT BTU/hour

Latent Heat gain = Qair*0.68(W_{air} – W_{room}) BTU/hour

A sample data sheet for electrical and infiltration is shown below.





12.3. Analysis and Result Summary

HVAC cooling cycle analysis is done in following steps.

Step 1

Estimate the total cooling load of the conditioned space with respect to its design conditions (77 °F, 50% relative humidity) discussed in the preceding section.

Room Total Heat = Total Sensible Heat (TSH) + Total Latent Heat (TLH)

Step 2

Determine outside ventilation air sensible (OASH) and latent heat (OALH) load given as follows based on ventilation airflow rate.

$$OASH = \frac{Q_{vent} * C_p * \Delta T}{v} \frac{BTU}{hour} (kW)$$

$$OALH = \frac{Q_{vent}(W_{vent} - W_{room}) * h_g}{v} \frac{BTU}{hour} (kW)$$

Step 3

Determine Effective Sensible Heat Factor (ESHF) and

$$ESHF = \frac{Room\ Sensible\ HeatLoad\,(RSH) + cbf \cdot Outside\ Air\ Sensible\ Heat(OASH)}{Room\ Total\ Heat\,(RTH) + cbf \cdot Outside\ Air\ Total\ Heat(OATH)}$$

Step 4:

Determine Apparatus Dew Point (ADP) as shown in section 12.1. Analytically, T_{adp} is solved using the following ESHF equation. And one can also determine graphically by drawing a slope line equivalent to ESHP that intersects the saturation curve in a psychrometric chart.

$$ESHF = \frac{cp(T_{room} - T_{adp})}{cp(T_{room} - T_{adp}) + h_g(w_{room} - w_{adp})}$$

Step 5:

Determine dehumidification air quantity to offset the room sensible heat load with 20°F (ΔT) supply-temperature differential as well as apparatus differential.

Effective Room Sensible Heat $ERSH = RSH + cbf * OASH$

$$Q_{room-air} = \frac{v * ERSH}{C_p * \Delta T} cfm$$

Step 6:

Determine total air passing through the cooling coil based on ADP which is much lower than the room supply temperature.

$$Q_{coil} = \frac{v * ERSH}{C_p * (T_{room} - T_{adp})} cfm$$

Step 6:

Determine the air quantity need to recirculate from return air i.e. state 5 in the model diagram.

$$Q_5 = Q_{room-air} - Q_{coil}$$

Step7:

Determine the evaporator/cooling coil capacity based on grand total heat as given below.

GTH = (RTH + OATH) BTU/hour

$$Cooling\ Coil\ Capacity = \frac{GTH}{12000} Ton$$

Result Summary:

HVAC cooling cycle analysis results can be summarized as follows.

1. Cooling load in Ton or kW depending upon the units chosen.
2. Airflow rates across the cooling coil and conditioned space.
3. Supply and return line fan capacity.
4. Amount of recirculation necessary to maintain a constant temperature differential.
5. Moisture removal in the dehumidification process.

12.4. Case Example

12.4.1 An office complex is air-conditioned to offset the estimated sensible heat gain of 202600 BTU/hour and latent heat gain of 41980 BTU/hour. Outdoor ventilation air at 95 °F dry bulb and 75 °F wet bulb temperature enters the system @ 2000 cfm is mixed with return air and then passed through the cooling coil. The conditioned space is maintained at 75 °F dry-bulb temperature with 50% relative humidity. The circuit is consisting of dampers and mixing points to control the mass flow rates. Part of the return air is bypassed the coil to maintain a constant temperature differential of 20 °F with respect to the conditioned space. Determine humid air properties of each state, Effective Sensible Heat Factor, Apparatus Dew Point, dehumidification air quantity and grand total heat.

Given:

Room Sensible Heat (RSH) = 202600 BTU/hour
Room Latent Heat (RLH) = 41980 BTU/hour
Room Total Heat (RTH) = RSH + TLH = 244580 BTU/hour

Outside Air Sensible Heat (OASH)

$$QASH = \frac{q \cdot cp \cdot \Delta T}{v} = \frac{2000 cfm \cdot 0.24 \frac{BTU}{lb \cdot {}^\circ F} \cdot (95-75)^\circ F}{13.68 \frac{ft^3}{lb}} = 42105 \frac{BTU}{hr}$$

Outside Latent Heat Load (OALH)

$$OALH = \frac{(w_{oa} - w_r) h_g \cdot q}{v} = \frac{(98.5-65.5) \frac{grain}{lb} \frac{lb}{7000 grain} \cdot 1075 \frac{BTU}{lb} \cdot 2000 \frac{ft^3}{\min}}{13.68 \frac{ft^3}{lb}} = 44454 \frac{BTU}{hr}$$

Outside Air Total Heat (OATH) = OASH + OALH = 86560 BTU/hr.

A. Effective Sensible Heat Factor:

$$ESHF = \frac{Room\,Sensible\,HeatLoad\,(RSH) + cbf \cdot Outside\,Air\,Sensible\,Heat(OASH)}{Room\,Total\,Heat\,(RTH) + cbf \cdot Outside\,Air\,Total\,Heat(OATH)}$$

$$ESHF = \frac{202600 + 0.15 \times 42105}{244580 + 0.15 \times 86560} = 0.811$$

B. Apparatus Dew Point

The ‘Apparatus Dew Point’ is the point on the saturation line where the ESHF line from room design intersects on the psychrometric chart and found to be 51 °F

C. Dehumidification Air Quantity

The following is the dehumidified air required to offset the room sensible heat load with 20°F supply temperature differential as well as apparatus differential.

$$Q_{supply} = \frac{v \cdot ERSH}{cp \cdot (\Delta T_{supply})} = \frac{13.68 \frac{ft^3}{lb} \cdot (202600 + 0.15 \cdot 42105) \frac{BTU}{hr}}{0.24 \frac{BTU}{lb \cdot {}^\circ F} 20^\circ F} = 9923 \cdot cfm.$$

$$Q_{App} = \frac{v \cdot ERSH}{cp \cdot (T_{room} - T_{ADP})(1 - cbf)} = \frac{13.68 \cdot ft^3 \cdot lb^{-1} (202600 + 0.15 \cdot 42105) BTU \cdot hr^{-1}}{0.24 \cdot BTU \cdot lb^{-1} {}^\circ F^{-1} \cdot (75 - 51.1)^\circ F \cdot 0.85} = 9770 \cdot cfm$$

Since air quantity required to pass through the apparatus is less than room air supply required, the difference in quantity is re-circulated from the room return air.

$Q_{bypass} = Q_{supply} - Q_{app} = 9923 - 9770 = 152$ cfm

The apparatus entering temperature is calculated as follows.

$$T_3 = T_{room} + \frac{q_{vent} \cdot (T_1 - T_{room})}{Q_{Apr}} = 75 + \frac{2000 \cdot (95 - 75)}{9770} = 79.1°F$$

The apparatus exit temperature is calculated based on the coil bypass factor as follows.

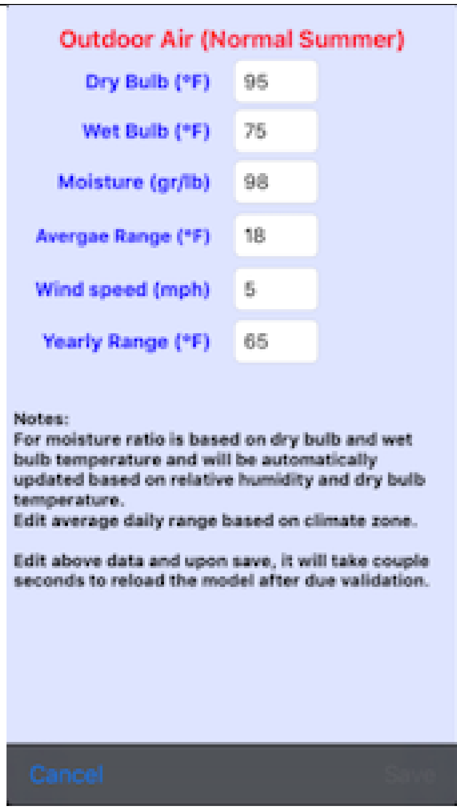
$$T_4 = T_{adp} + cbf\left(T_3 - T_{adp}\right) = 51.1 + 0.15(79.1 - 51.1) = 55.3°F$$

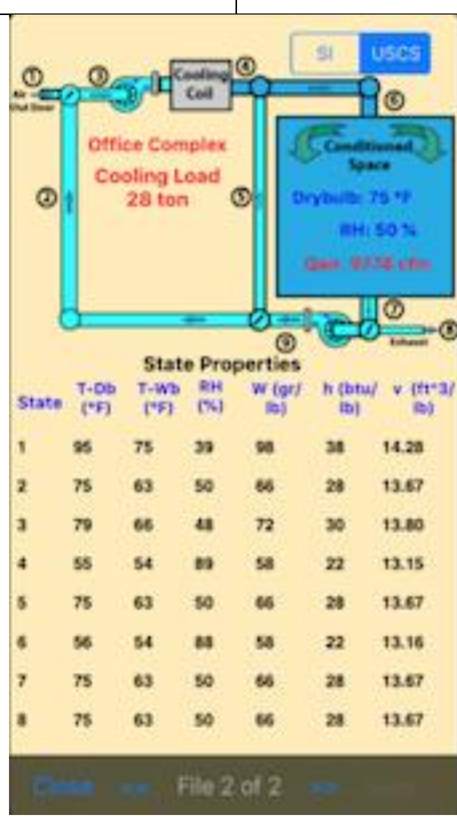
D. Grand Total Heat

GTH = RTH + OATH = 331140 BTU/hour

$$\textbf{Cooling Load} = \frac{GTH}{12000} = 28\ \textbf{Ton}$$

App Suite HVAC Cooling Cycle Analysis



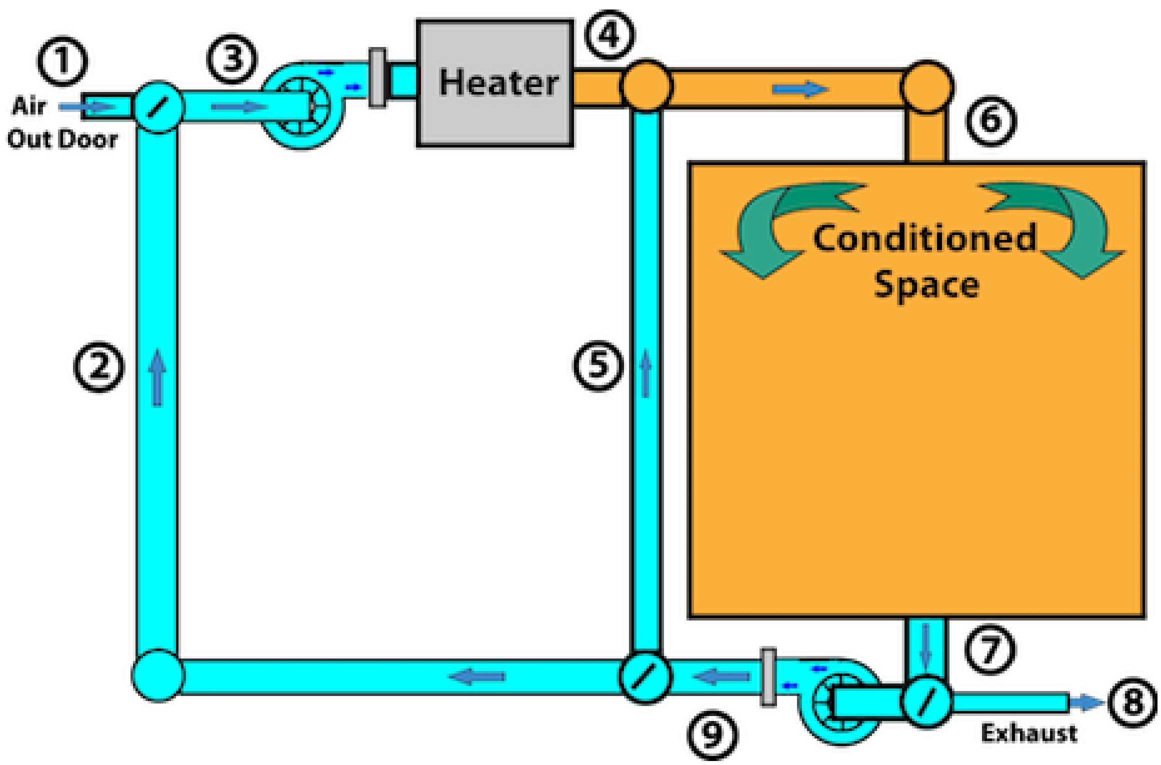


State	T-Db (°F)	T-Wb (°F)	RH (%)	W (gr/lb)	h (btu/lb)	v (ft^3/lb)
1	95	75	39	98	38	14.28
2	75	63	50	66	28	13.67
3	79	66	48	72	30	13.80
4	55	54	89	58	22	13.15
5	75	63	50	66	28	13.67
6	56	54	88	58	22	13.16
7	75	63	50	66	28	13.67
8	75	63	50	66	28	13.67

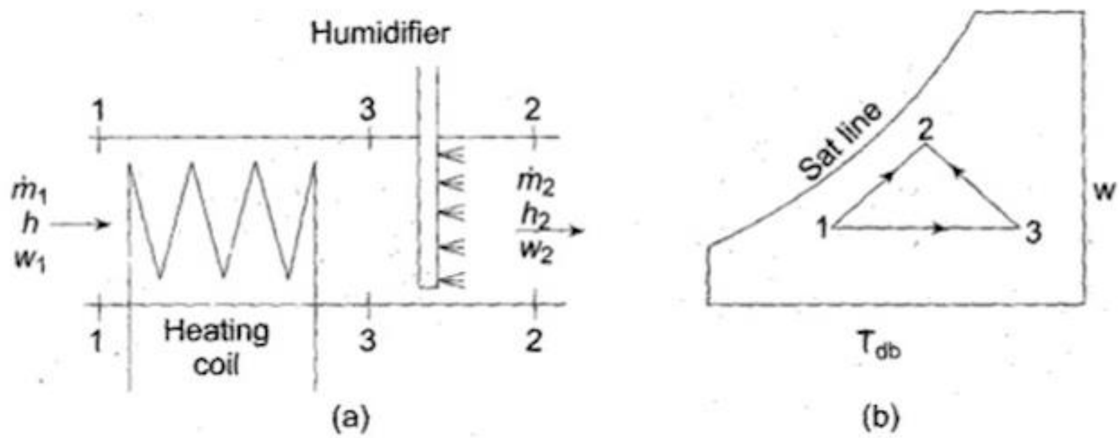
13. HVAC Heating Cycle

13.1. Generic Design Analysis Model

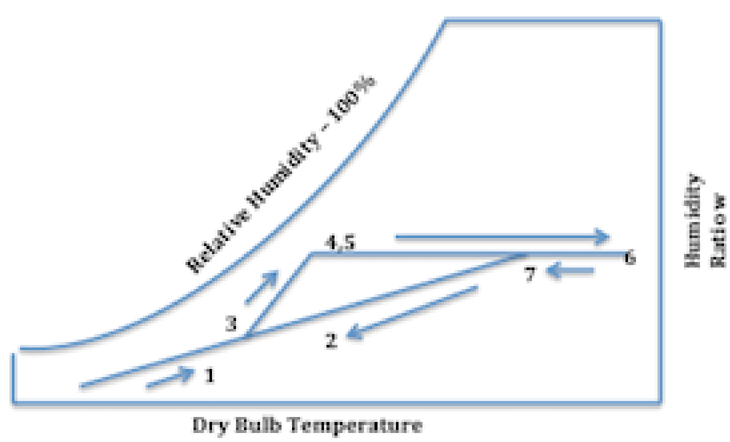
A generic model comprises of heating space, heater and fans that circulate a predetermined quantity of air to maintain temperature and humidity as per the recommended standards. Dry bulb 70 °F with 50% relative humidity is considered most comfortable for human activities. The circuit consists of dampers and mixing points to control the mass flow rates as shown in the following model. State numbers are assigned in sequence from outdoor air to return air. Part of the return air is bypassed back to the conditioned space mixed with conditioned outdoor air depending upon total heat lost or gained with in the conditioned space. (Refer Appendix 15.2.9 for ASHRAE pyschrometric)



Similar procedure is followed given in section 12.1 in performing psychrometric analysis of the humid air except humidification may be required depending upon the room design condition and condition of outdoor air as shown below.



Psychrometric chart for generic heating model is shown below

State 1: Outdoor air State 2: Return air State 3: Mixed air State 4: Heating and humidification. State 5: Bypass air State 6: Supply air State 7: Room Design Condition State 8: Exhaust air State 9: Return air fan.	

13.2. Heating Load Estimation

HVAC heating loads are estimated using the same 10 broad categories covered under HVAC cooling and given as follows.

1. Solar Heat Gain Through Glass
2. Heat Loss Through Wall Conduction
3. Heat Loss Through Roof Conduction
4. Heat Loss Through Floor Conduction
5. Heat Loss Through Ceil Conduction
6. Heat Loss Through Door Conduction
7. Heat Loss Through Partition Conduction
8. Internal Heat Gain - People
9. Internal Heat Gain – Electrical
10. Infiltration Heat Loss

Categories 2 to 7 contribute mainly to sensible heat loss by simple conduction, which is function of material properties used in the respective component and temperature differential of outside and conditioned space. Categories 8 and 9 contribute to both sensible and latent heat gains. Category 10 here mainly heat loss due to infiltration of cold air into the conditioned space.

13.2.1. *Solar Heat Gain*

The solar heat gain through glass depends upon the solar intensity during the design month, time, and orientation of the glass structure. One can also use the average intensity factor with individual glass area. Thus, total solar heat gain is calculated using following equation. ASHRAE handbook provides data for different storage and shading factors. However, one can also use default values provided in the app suite for general estimation as discussed in section 12.2.1.

$$Soalr\ Heat\ Gain\ (SHG) = Area * Solar\ intensity * storage * shading$$

13.2.2. Conduction Through *Wall/Roof/Floor/Ceil/Partition/Door*

The heat loss through conduction is primarily dependent upon the overall heat transfer coefficient (U) of the respective component, and the ELTD (Equivalent Load Temperature Difference). Where ELTD is a function of orientation, month and time and ASHRAE handbook gives this value along with recommended U value. However, app suite provides default values and heat loss is calculated as:

$$Heat\ loss = U.A.ELTD$$

U- value for composite structure can be estimated using individual material resistance based on thermal conductivity and thickness and then computed overall conductance of the composite structure consisting of four different materials as follows.

$$U = \frac{1}{R_1 + R_2 + R_3 + R_4}$$

This is very similar to one shown in section 12.2.2 for the cooling cycle.

13.2.3. Internal Heat Gain – People

Heat gain through people within the space is mainly due to the rate of metabolism in human body. Though this rate varies from person to person, a general recommended values of both sensible and latent heat for different activity levels are shown in section 12.2.3 is applicable in this case as well.

13.2.4. Heat Gain – Electrical Appliances

Electrical appliances used with in conditioned space can contribute substantial heat starting from simple fluorescent lamp to electrical motors. There are seven most common appliances with recommended values for both sensible and latent heat is shown in section 12.2.4 is applicable in this case for calculating the total heat gain.

13.2.5. Heat Loss – Infiltration

Infiltration through the structure depends upon the ambient conditions as well as wind speed. The pressure differential is sufficient to cause flow across various joints and gaps in the structure thereby adding additional heat loads. The basic assumption is whatever outdoor air gets in it contributes only towards heat loss while flow rate remains unchanged. There are empirical formulae to compute the heat gain due to infiltration available in standard handbook such as ASHRAE and other HVAC reference documents. App suite follows the same procedure shown in section 12.2.5 in estimating heat loss due to infiltration.

13.3. Analysis and Result Summary

HVAC heating cycle analysis is done in following steps.

Step 1

Estimate the total heating load of the conditioned space with respect to its design conditions (72 °F, 50% relative humidity) under 10 different categories explained in the previous section.

Room Total Heat = Total Sensible Heat (TSH) + Total Latent Heat (TLH)

Step 2

Determine outside ventilation air sensible (OASH) and latent heat (OALH) load given as follows based on ventilation airflow rate.

$$OASH = \frac{Q_{vent} * C_p * \Delta T}{v} \frac{BTU}{hour} (kW)$$

$$OALH = \frac{Q_{vent}(W_{vent} - W_{room}) * h_g}{v} \frac{BTU}{hour} (kW)$$

Step 3

Determine Effective Sensible Heat Factor (ESHF) and most cases it can be treated as 1.0 for heating and humidification since this factor has no significance in the pyshrometric process.

Step 4.

The following equation calculates room sensible heat load that need to be offset by room airflow supply at given temperature differential.

$$ERSH = RSH + cbf \cdot OASH$$

$$Q_{supply} = \frac{v * ERSH}{C_p * \Delta T}$$

Step 5:

Determine apparatus transfer point (ATP), Airflow rate, and Humidification rate. The 'Apparatus Transfer Point' is a hypothetical temperature point at which the heat is transferred to the air from the coil at constant rate. Therefore, the amount of return air required to pass through the heating coil depends upon this temperature and given by the following equation.

$$Q_{coil} = \frac{v * ERSH}{C_p * \left(T_{room} - T_{atp}\right) * (1 - cbf)} cfm$$

.

The humidification is carried out by means of a spray or injecting steam into the dry air to raise its humidity ratio. The humidifier rate is the rate at which moisture is added to reach the desired humidity ratio. This is given by the following expression.

$$Humidifier\ rate = \frac{RLH + OALH}{\frac{Q_{coil}}{v} * h_g}$$

Step 6:

Determine the heater coil capacity based on grand total heat as given below.

$$GTH = (RTH + OATH)\ kW$$

Result Summary:

HVAC heating cycle analysis results can be summarized as follows.

1. Heating load in BTU/hr or kW depending upon the units chosen.
2. Airflow rates across the heater and conditioned space.
3. Supply and return line fan capacity.
4. Amount of recirculation necessary to maintain a constant temperature differential.
5. Humidifier capacity.

13.4. Case Example

A 150 m^2 office annex circulates conditioned air to maintain inside temperature at 21 °C with 50% humidity during a winter day when outside air is at 2 °C dry bulb and -2 °C wet bulb.   Determine the quantity of air and humidifier rate to offset sensible heat loss of 7.2 kW and latent heat loss of 0.7 kW. Ventilation air is maintained at 340 m^3/hour.

A. Conditioned space heat load:

Room sensible heat RSH: 7.2 kW

Room latent heat RLF: 0.7 kW

Total room heat loss = 7.9 kW

B. Outside air heat Load

$$OASH = \frac{Q_{vent} * C_p * \Delta T}{v} = \frac{340 * \frac{m^3}{3600*sec} * 1.0 * \frac{kJ}{Kg*K} * (21-2)}{0.843 * \frac{m^3}{kg}} 2.12kW$$

$$OALH = \frac{Q_{vent}(W_{vent} - W_{room}) * h_g}{v}$$

$$= \frac{340 * \frac{m^3}{3600*sec} * (7.82 - 1.65) * \frac{gm}{kg} * 2555 \frac{kJ}{kg}}{0.843 * \frac{m^3}{kg}} = 1.77kW$$

Total OATH = OASH + OALH = 2.12 + 1.77 = 3.89 kW

C. Determine ESHF

Assuming coil bypass factor of 5% ESHF is written as:

$$ESHF = -\frac{RSH + cbf * OASH}{RTH + cbf * OATH} = -\frac{7.2 + 0.05 * 2.12}{7.9 + 0.05 * 3.89} = -0.90$$

D. Room sensible heat

Calculate room sensible heat assuming 5% coil bypass

Cbf = 0.05

ERSH = RSH + cbf * OARH = 7.2 + 0.05*2.12 = 7.30 kW

E. Determine room airflow rate assuming 20 °F temperature differential.

$$Q_{supply} = \frac{v * ERSH}{C_p * \Delta T} = \frac{0.843 * 7.30}{1.0 * 11.2 * 3600} = 0.549 \frac{m^3}{sec} = 1978 \frac{m^3}{hour}$$

F. Humidifier rate to maintain design condition at 50% RH

Humidity ratio at design room $w_{room} = 7.82 * gm/kg - dryair$

Humidity ratio at mixer $w_3 = \frac{w_2 * m_2 + w_1 * m_1}{m_2 + m_1} = 6.25 * gm/kg - dryair$

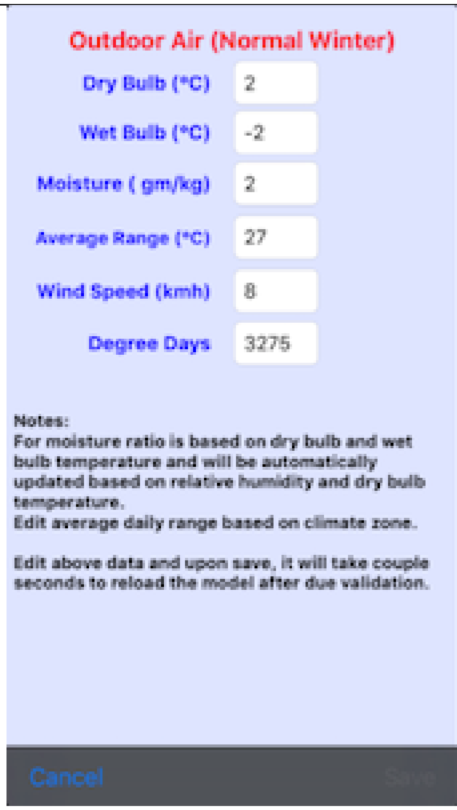
$$Humidifier\ rate = \ w_{room} - w_3 = \frac{7.82 - 6.25}{1000} = 0.0015 * \frac{kg}{kg - air}$$

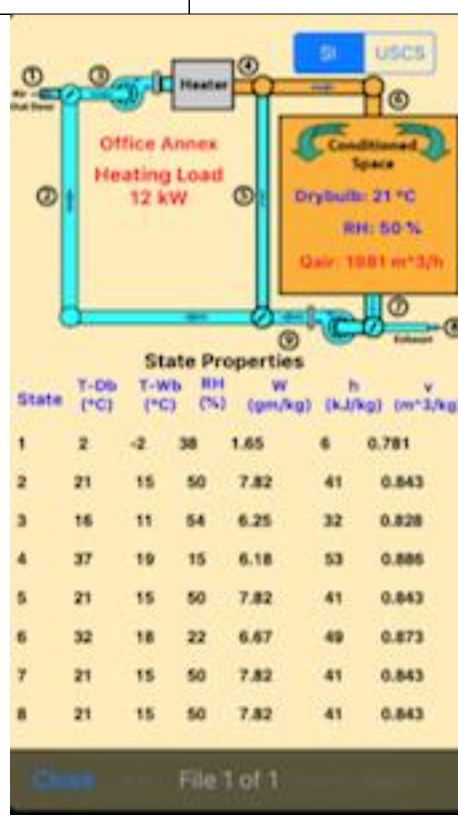
G. Heating Load

Grand Total Heat load

GTH = RTH + OATH = 7.9 + 3.89 = 11.79 kW

App Suite Analysis





14. Plant Maintenance System

14.1 Total Productive Maintenance Philosophy

The philosophy of Total Productive Maintenance (TPM) was first introduced in Japan in 80's following Total Quality Management (TQM) in 70's. TPM and TQM have one thing in common, i.e., "Changing Corporate Culture." Also advocate small group activities but differ in subject management. In a manufacturing industry 'Quality' refers to the final product or output while 'Equipment' that produces is a cause or input. Thus, effective management of these input and output can enhance overall productivity in any organization.

TPM aims to run any production system at its maximum effectiveness by eliminating waste or losses. More specifically it aims at eliminating major losses grouped under following three major categories and can be quantified as:

A. Downtime loss (Availability)
B. Speed loss (Performance efficiency)
C. Loss due to defects in the final product (Rate of quality output).

Overall Equipment Effectiveness = A x B x C

The other objective of TPM is in pursuit of economic life cycle cost. Although it is like "Terotechnology" advocated in 70's in UK but differs in scope. Terotechnology encompasses concept to customer that includes design, manufacturer, engineering, and then users. TPM focuses mainly on management of physical assets through autonomous small group activities with assigned responsibilities based on policies and objectives determined by a corporate level committee, department managers, supervisors and group leaders.

This is illustrated using a case study from a medium scale pulp and paper plant and data for six months as given below.

Maintenance Record:

Total available hours: 4272

Equipment failures: 422.9

Setup and changes: 137.1 (wire/felt change)

$$\boldsymbol{Availabilty\ (A) = \frac{4272 - 560}{4272} = 0.869}$$

Production Record:

Theoretical cycle time: 0.4 hours/Metric ton (Based on 60 TPD)

Actual production hours: 3721.5 hours

Production Quantity: 7773 Metric tons

$$\boldsymbol{Performance\ Efficiency\ (B) = \frac{0.4 * 7773}{3712.5} = 0.837}$$

Quality Control Record:

Total Salable product: 7621 Metric tones

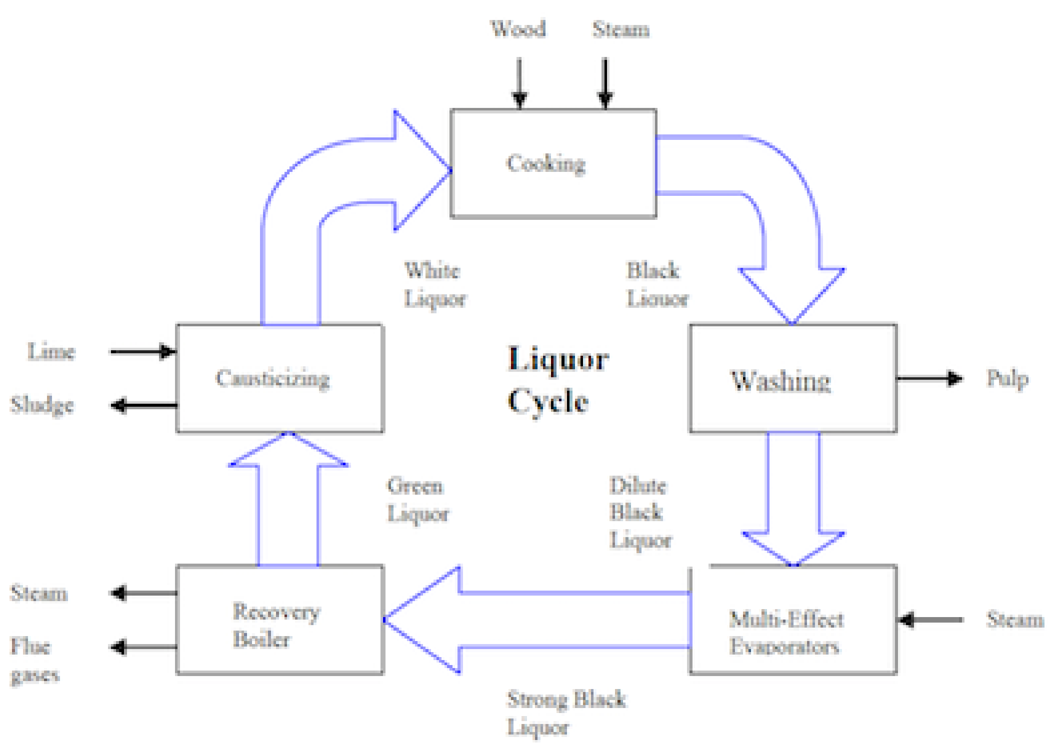
$$\boldsymbol{Rate\ of\ Quality\ Product\ (C) = \frac{7621}{7773} = 0.98}$$

The overall production system effectiveness can be computed as:

$$\boldsymbol{Effectiveness = A * B * C = 0.867 * 0.837 * 0.98 = 71.2\%}$$

The above data was collected during a project assignment to improve overall system and cost effectiveness. The cost effectiveness in a pulp and paper industries that employs "Kraft" process for chemical wood pulping depends upon energy efficiency measures taken as well. This is briefly described below.

The Kraft process uses cooking liquor mainly comprises of sodium hydroxide, carbonate, sulfide, sulfate and known as white liquor. The composition and concentration vary depending upon type of wood chips. Under certain temperature (170 °C / 340 °F) and pressure (6.9 bar /100 psig) white liquor reacts to remove all non-cellulosic components from the wood fibers. In the process it becomes dark brownish in color called weak black liquor, which contains lignin (organic) and inorganic compounds. Lignin is highly combustible and that can be fired in a boiler after concentrating in multi-effect evaporators. The bottom ash from the boiler contains the inorganic compounds is then converted into white liquor in causticizing plant. This completes the liquor cycle as shown below.



Energy consumption in the liquor cycle (current figures*)

Process	**Steam MJ/ADt**	**Electricity kWh/ADt**
Cooking	1.7	40
Washing	0	30
Evaporation	3.1	30
Recovery Boiler	-15.8	-655
Causticizing	0	50

(*Pulp & Paper Technical Association Canada) ADt- Airdry metric ton

The recovery boiler produces high-pressure steam that can run a backpressure turbine to generate electricity and exhaust steam at pressure and temperature required by the pulping process. Thus, there is a net gain in energy generation enough to cover all downstream equipment including paper mill. Hence a good energy management coupled with TPM can ensure overall system and cost effectiveness.

Process equipment failures resulting in downtime needs a closer look in maximizing effectiveness. In the current 21st century a great deal of automation and sophistication are built into this equipment to minimize operator intervention for continuous production. Therefore, it also needs same degree of sophistication in diagnosis and care of this process equipment. With advent of Artificial Intelligence and expert system TPM needs to integrate and incorporate AI in monitoring and diagnostic for life extension to achieve overall cost effectiveness.

14.2 Maintenance Standards and Application

One of the primary objectives of TPM is to detect and restore failures well in advance by employing modern monitoring and diagnostic tools. Especially in chemical and process industries where considerable cost is associated with unplanned stoppages otherwise expected to run continuously. The process equipment that is subjected to adverse operating conditions is broadly classified as static and rotating. Each one of them got their own life expectancy depends upon the severity of application. Mostly static equipment that is under chemical environment subjected to severe corrosive wears. Therefore, industry specific standards need to be applied for restoration and annual turn around maintenance and NDT inspection (Table 15.1.1) of all static equipment such as pipes, pressure vessels and tubes, and storage tanks etc. For rotating equipment, general vibration standards can be applied to ensure its operation and structural integrity. Some of the applicable standards are listed below.

- ISO 2372: For general machine severity (Measurements are taken on bearing housings)
- ISO 3945: For large rotating machines (Measurement taken on shaft)
- VDI 2056: Same as ISO 2372 (Appendix 15.1.2, 15.2.10)
- CDA/MS/NVSH 107: Canadian Maintenance Standards for new and old machines.

After every installation and commissioning of any rotating equipment whether it is new or old should comply above applicable standards before putting that equipment on production.

14.3 Cause and Consequence Analysis under Hazardous Conditions

There is a considerable overlap between the term reliability, safety, hazard, and risk. If the purpose is to determine safety, the reliability analysis is then extended to determine whether any failure or operability can cause possible damage to the system or to the people working around. For example, reliability analysis may come up with the probability that a chemical reactor may overheat due to malfunctioning of pump or heat exchanger or utilities. In other words, it's failure modes, effects and criticality analysis (FMECA). If this were extended to study how frequently such overheating would result into explosion, we would be looking at the safety (Hazard) problem or risk analysis. There are two kinds of risk associated.

1. Risk to Life
    a. Mechanical risk
    b. Process risk (Loss of containment or leakage)
2. Risk to plant and profit
    a. Equipment damage
    b. Equipment outage

The criticality classification is done based on above risk factors and given as follows.

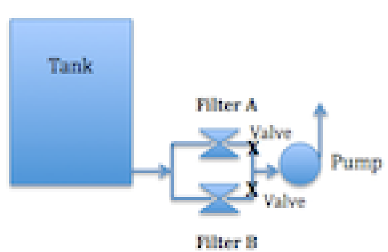
Category	**Description**	**Cost M$**	**Injury**	**FATAL**
1. Negligible	Minor damage	< 0.1	None	None
2. Marginal	Serious damage	0.1-0.5	Potential	None
3. Substantial	Major damage	0.5-1	Some	Probable
4. Critical	Damage over wide area	1 - 5	Serious	1+

5. Catastrophic	Wide spread damage	>5	Severe	25

Failure and effect (FEA) analysis some time referred as cause and consequence analysis in a hazardous process environment shown with an example.

Hazardous chemical flows through a pump from a storage tank, filter, and valve. The setup is as shown in the following figure. One can analyze deviations in process parameters such as flow, pressure, and temperature etc. using set of guidewords its cause and consequences.

FEA Analysis



1. None (No Flow)

Causes	Deviation	Consequences	Action
A. Tank empty	No Flow	Mechanical seal failure	Low level alarm & trip
B. Valve shut off	-do-	-do-	Low feed trip
C. Filter block	-do-	-do-	Low feed trip

Finally, one needs to access the effects based on series of events and their probability of occurrences. For example, in the above case pump mechanical seal failure due to no flow is considered as initiating event and probability of occurrence be P(a).

If the protective system is in a failed state and let the probability if its occurrence be P(b) and in which also operator non-intervention taking place in this succeeding event and let the probability is P(c).

The top event is the hazardous incident occurs and that can be categorized under above consequence criticality categories (1 to 5).

In the above example, the top event will be category 1 under no flow condition i.e., minor damage, no injuries or fatalities.

Probability of this category 1 hazardous event is: P(a)*P(b)*P(C)

This is just an example how any accident take place in series of events based on equipment/component failure rate. Failure rate for most electronic and mechanical component is published by respective industries for their product by performing reliability and accelerated life testing such as L10 life for bearings. One can use this failure rate data to come up with fatal accident rate (FAR) under category 5.

FAR/10^6 hours for certain specific industry type is listed below.

1. Chemical and process Industry (USA): 3
2. Chemical and process Industry (UK): 4
3. Coal Mining: 40
4. Construction: 67

(Please note this is not current. Data published during 1980's.)

14.4 Maintenance Information System

TPM succeed only when it is supported by a proper information system for planning and execution of all maintenance tasks. Now there is a gamut of technology and platforms available that provides these services depends upon economic significance of maintenance function in any organization. This includes both hardware and software support at different infrastructure and architectural level such as AWS, Azure, SAP, SAS, etc. But all have one thing in common when comes to maintain certain basic maintenance records and reporting procedures listed as follows.

A. Maintenance records
   a. Equipment data including engineering drawing.
   b. Equipment history.
   c. Spare parts inventory.
   d. Inspection checklists and monitoring schedules.
   e. Gnat chart/PERT charts for major overhaul and annual turn around.
   f. Failure investigation records of all categories 3 and above events.
B. Maintenance procedures
   a. Work order procedures for breakdown and PM activities.
   b. Spare parts ordering and procurement.
   c. Costing procedures.
   d. Failure investigation procedures.
   e. Standard operating procedure/tag out lock out.
   f. Management reporting.

As every maintenance personnel carry a smart phone, one can share and report instantly any information pertaining to critical maintenance activities. In addition, engineers can have native apps for onsite analysis as well as performance trend monitoring of critical process equipment like described in this manual. Hence autonomous maintenance philosophy advocated by TPM is now a reality.

Supervisors or leaders of small group activities can form their own group just like WhatsApp group to communicate and exchange all info to speed up the assigned tasks. Thus, all delays in logistics part can be cut down drastically. This is one way to maximize effectiveness and reduce cost due to delays in execution.

15. Appendix

15.1 Tables

15.1.1. NDT for corrosion inspection

Technique Corrosion type	*Radiography*	*Ultrasonic*	*Eddy current*	*Magnetic particle*	*Dye penetrant*
Hidden wastage of unknown mechanism	Best general technique	Limited	Limited	N.A.	N.A.
General wastage	Poor	Best general technique	Limited	N.A.	N.A.
Pitting corrosion	Best general technique	Can detect	Good method	N.A.	N.A.
Intergranular corrosion	Can detect	Best general technique	Good method	Can detect	Can detect
Dezincification	Can detect	Best general technique	N.A.	N.A.	N.A.
Corrosion fatigue	Best general technique	Good method	Good method	Good method	Can detect
Stress corrosion cracking	Good method	Best general technique	Good method	Good method	Good method
Hydrogen embrittlement cracking	Can detect	Best general technique	Good method	Good method	Good method

Reference: R. A. Collacott

15.1.2. VDI 2056/ISO 2372 Vibration Standard

Vibration severity ranges		*Examples of evaluation stages for individual groups of machines*			
Range classification	*Effective velocity in $mm\ s^{-1}$ rms at the range*	*Group K*	*Group M*	*Group G*	*Group T*
0.28	0.28	Good			
0.45	0.45		Good		
0.71	0.71				
1.12	1.12	Usable		Good	
1.8	1.8		Usable		Good
2.8	2.8	Still acceptable		Usable	
4.5	4.5		Still acceptable		Usable
7.1	7.1			Still acceptable	
11.2	11.2				Still acceptable
18	18	Not acceptable			
28	28		Not acceptable	Not acceptable	Not acceptable
45	45				

Group K = Individual drive units of prime movers and processing machines which are rigidly fixed to the entire machine in the operating condition; particularly mass-produced electric motors up to about 15 kW.

Group M = Medium size machines, particularly electric motors from 15 to 75 kW capacity, without special foundations: also rigidly mounted drive components and machines (up to about 300 kW) with rotating parts only on special foundations.

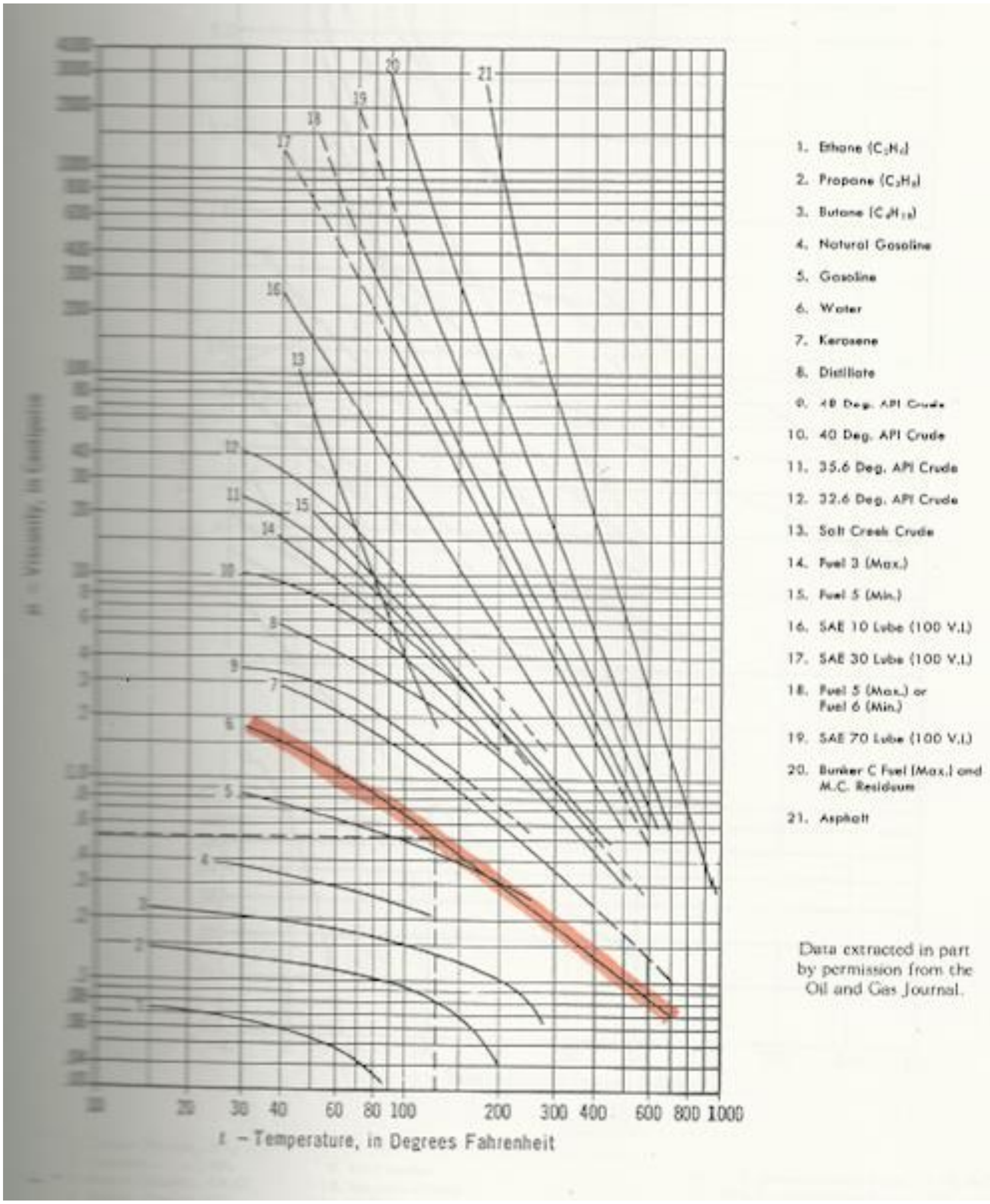
Group G = Larger machines, prime movers or processing machines with rotating parts only, mounted on rigid or heavy foundations with a high natural frequency of vibration.

Group T = Larger prime movers and processing machines with rotating masses only, mounted on foundations with a low natural frequency of vibration, e.g. turbine groups, particularly those on foundations built on light construction principles.

Reference: R.A. Collacott

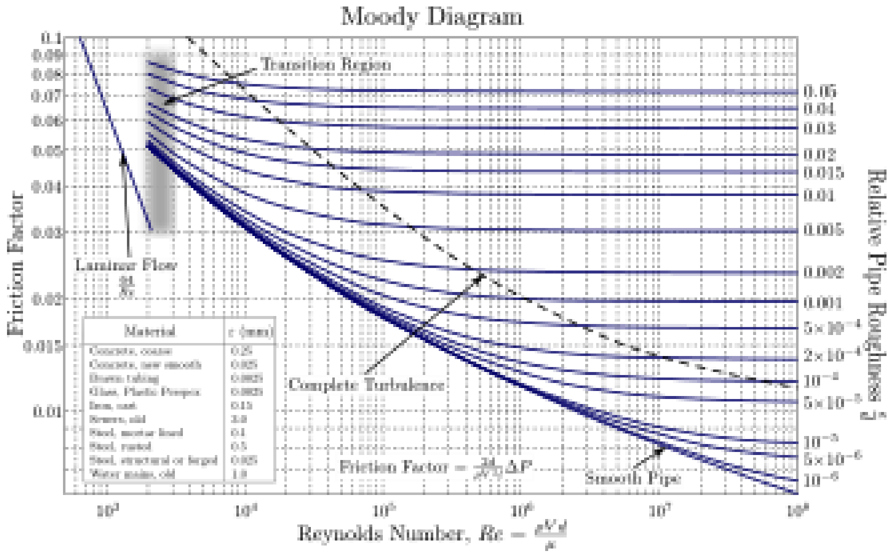
15.2 Charts

15.2.1. Viscosity Temperature chart for common petroleum fluids

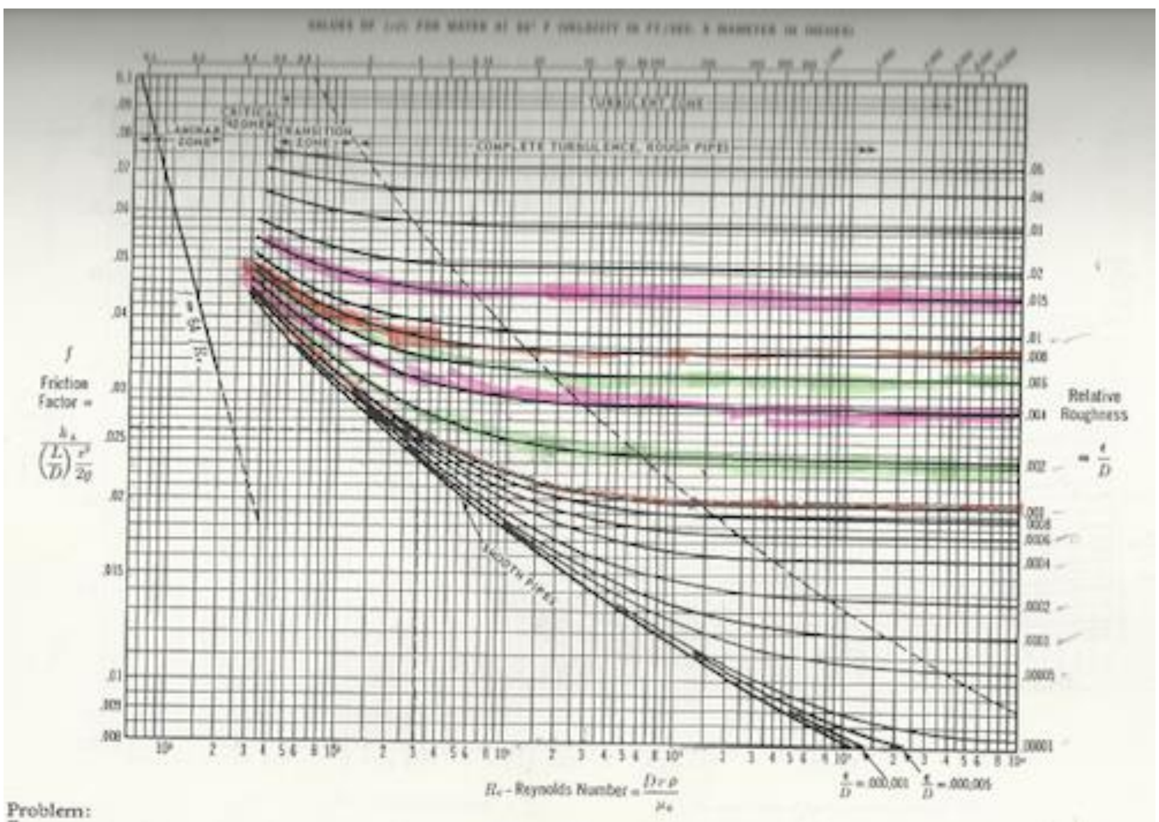


Reference: Crane Engineering, US publication. (Water is shown in color)

15.2.2. Moody Diagram



15.2.3. Moody Diagram for Water



Reference: Crane Engineering, US publication, exclusively for water.

15.2.4 Fluid Pressure Drop Example 1 and 2

Simplified Flow Formula for Compressible Fluids

Pressure Drop, Rate of Flow, and Pipe Size

The simplified flow formula for compressible fluids is accurate for fully turbulent flow; in addition, its use provides a good approximation in calculations involving compressible fluid flow through wrought iron or commercial steel pipe for most normal flow conditions.

If velocities are low, friction factors assumed in the simplified formula may be too low; in such cases, the formula and nomograph shown on pages 3-10 and 3-21 may be used to provide greater accuracy.

The Darcy formula can be written in the following form:

$$\Delta P_{100} = W^2\left(\frac{0.000\,336 f}{d^5}\right)\bar{V} = (W^2 10^{-6})\left(\frac{336\,000 f}{d^5}\right)\bar{V}$$

$$C_1 = W^2 10^{-6} \qquad C_2 = \frac{336\,000 f}{d^5}$$

The simplified flow formula can then be written:

$$\Delta P_{100} = C_1 C_2 \bar{V} = \frac{C_1 C_2}{\rho}$$

$$C_1 = \frac{\Delta P_{100}}{C_2 \bar{V}} = \frac{\Delta P_{100}\,\rho}{C_2} \qquad C_2 = \frac{\Delta P_{100}}{C_1 \bar{V}} = \frac{\Delta P_{100}\,\rho}{C_1}$$

C_1 = discharge factor from chart at right.
C_2 = size factor, from table on next page.

The limitations of the Darcy formula for compressible flow, as outlined on page 3-3, apply also to the simplified flow formula.

Example 1

Given: Steam at 345 psig and 500 F flows through 8-inch Schedule 40 pipe at a rate of 240,000 pounds per hour.

Find: The pressure drop per 100 feet of pipe.

Solution:
$C_1 = 57$
$C_2 = 0.146$
$\bar{V} = 1.45$ ....page 3-17 or A-16
$\Delta P_{100} = 57 \times 0.146 \times 1.45 = 12$

Example 2

Given: Pressure drop is 5 psi with 100 psig air at 90 F flowing through 100 feet of 4-inch Schedule 40 pipe.

Find: The flow rate in standard cubic feet per minute.

Solution:
$\Delta P_{100} = 5.0$
$C_2 = 5.17$
$\rho = 0.564$ ....page A-10
$C_1 = (5.0 \times 0.564) \div 5.17 = 0.545$
$W = 23\,000$
$q'_m = W \div (4.58\, S_g)$ ....page B-2
$q'_m = 23\,000 \div (4.58 \times 1.0) = $ **5000** scfm

Values of C_1

For C_1 values and an example on "determining pipe size", see the opposite page.

Reference: Crane Engineering, US publication

15.2.5. Fluid Pressure Drop Example 3

Simplified Flow Formula for Compressible Fluids

Pressure Drop, Rate of Flow, and Pipe Size — continued

Values of C_2

Nominal Pipe Size Inches	Schedule Number	Value of C_2
⅛	40 s	7 920 000.
	80 x	26 200 000.
¼	40 s	1 590 000.
	80 x	4 290 000.
⅜	40 s	319 000.
	80 x	718 000.
½	40 s	93 500.
	80 x	186 100.
	160	430 000.
	... xx	11 180 000.
¾	40 s	21 200.
	80 x	36 900.
	160	100 100.
	... xx	627 000.
1	40 s	5 950.
	80 x	9 640.
	160	22 500.
	... xx	114 100.
1¼	40 s	1 408.
	80 x	2 110.
	160	3 490.
	... xx	13 640.
1½	40 s	627.
	80 x	904.
	160	1 656.
	... xx	4 630.
2	40 s	169.
	80 x	236.
	160	488.
	... xx	899.
2½	40 s	66.7
	80 x	91.8
	160	146.3
	... xx	380.0
3	40 s	21.4
	80 x	28.7
	160	48.3
	... xx	96.6
3½	40 s	10.0
	80 x	13.2
4	40 s	5.17
	80 x	6.75
	120	8.94
	160	11.80
	... xx	18.59

Nominal Pipe Size Inches	Schedule Number	Value of C_2
5	40 s	1.59
	80 x	2.04
	120	2.69
	160	3.59
	... xx	4.93
6	40 s	0.610
	80 x	0.798
	120	1.015
	160	1.376
	... xx	1.861
8	20	0.133
	30	0.138
	40 s	0.146
	60	0.163
	80 x	0.185
	100	0.211
	120	0.252
	140	0.289
	... xx	0.317
	160	0.333
10	20	0.039 7
	30	0.042 1
	40 s	0.044 7
	60 x	0.051 4
	80	0.056 9
	100	0.065 2
	120	0.075 3
	140	0.090 5
	160	0.105 2
12	20	0.015 7
	30	0.016 8
	... s	0.017 5
	40	0.018 0
	... x	0.019 5
	60	0.020 6
	80	0.023 1
	100	0.026 7
	120	0.031 0
	140	0.035 0
	160	0.042 3
14	10	0.009 49
	20	0.009 96
	30 s	0.010 46
	40	0.010 99
	... x	0.011 55
	60	0.012 44
	80	0.014 16
	100	0.016 57
	120	0.018 98
	140	0.021 8
	160	0.025 2

Nominal Pipe Size Inches	Schedule Number	Value of C_2
16	10	0.004 63
	20	0.004 83
	30 s	0.005 04
	40 x	0.005 49
	60	0.006 12
	80	0.007 00
	100	0.008 04
	120	0.009 26
	140	0.010 99
	160	0.012 44
18	10	0.002 47
	20	0.002 56
	... s	0.002 66
	30	0.002 76
	... x	0.002 87
	40	0.002 98
	60	0.003 35
	80	0.003 76
	100	0.004 35
	120	0.005 04
	140	0.005 73
	160	0.006 69
20	10	0.001 41
	20 s	0.001 50
	30 x	0.001 61
	40	0.001 69
	60	0.001 91
	80	0.002 17
	100	0.002 51
	120	0.002 87
	140	0.003 35
	160	0.003 85
24	10	0.000 534
	20 s	0.000 565
	... x	0.000 597
	30	0.000 614
	40	0.000 651
	60	0.000 741
	80	0.000 835
	100	0.000 972
	120	0.001 119
	140	0.001 274
	160	0.001 478

Note

The letters s, x, and xx in the columns of Schedule Numbers indicate Standard, Extra Strong, and Double Extra Strong pipe respectively.

Example 3

Given: An 85 psig saturated steam line with 10,000 pounds per hour flow is permitted a maximum pressure drop of 10 psi per 100 feet of pipe.

Find: The smallest size of Schedule 40 pipe suitable.

Solution: $\Delta P_{100} = 10$ $\quad \bar{V} = 4.4$ ....page 3-17 or A-13

$C_1 = 0.4$ $\quad C_2 = 10 \div (0.4 \times 4.5) = 5.56$

Reference to the table of C_2 values above shows that the 4-inch size is the smallest Schedule 40 pipe having a C_2 value less than 5.56.

The actual pressure drop per 100 feet of 4-inch Schedule 40 pipe is:

$$\Delta P_{100} = 0.4 \times 5.17 \times 4.4 = 9.3$$

Reference: Crane Engineering, US publication

15.2.6. Gas flow through orifice

Flow of Compressible Fluids Through Nozzles and Orifices

The flow of compressible fluids through nozzles and orifices can be determined from the following formula, or, by using the nomograph on the next page. The nomograph is a graphical solution of the formula.

$$w = 0.525\, Y d_1^2 C \sqrt{\Delta P \rho_1} = 0.525\, Y d_1^2 C \sqrt{\frac{\Delta P}{\bar{V}_1}}$$

$$W = 1891\, Y d_1^2 C \sqrt{\Delta P \rho_1} = 1891\, Y d_1^2 C \sqrt{\frac{\Delta P}{\bar{V}_1}}$$

(Pressure drop is measured across taps located 1 diameter upstream and 0.5 diameter downstream from the inlet face of the nozzle or orifice)

Example 1

Given: A differential pressure of 11.5 psi is measured across taps located 1 diameter upstream and 0.5 diameter downstream from the inlet face of a 1.000-inch I.D. nozzle assembled in a 2-inch Schedule 40 steel pipe, in which, dry carbon dioxide (CO_2) gas is flowing at 100 psig pressure and 200 F.

Find: The flow rate in cubic feet per hour at standard conditions (scfh).

Solution:

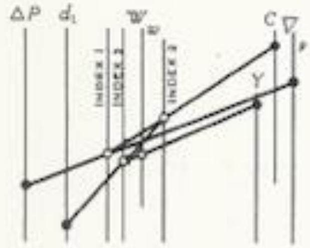
1. $R = 35.1$
2. $S_g = 1.516$ .... for CO_2 gas; page A-8
3. $k = 1.28$

Steps 3 through 7 are used to determine the Y factor.

4. $P'_1 = P + 14.7 = 100 + 14.7 = 114.7$
5. $\Delta P / P'_1 = 11.5 \div 114.7 = 0.1003$
6. $d_2 = 2.067$ .... 2" Sched 40 pipe; page B-16
7. $\beta = 1.00 \div 2.067 = 0.484$
8. $Y = 0.93$ .... page A-21
9. $C = 1.02$ .. turbulent flow assumed; page A-20
10. $T = 460 + t = 460 + 200 = 660$
11. $\rho_1 = 0.71$ .... page A-10

	Connect		Read
12.	$\Delta P = 11.5$	$\rho_1 = 0.71$	Index 1
13.	Index 1	$C = 1.02$	Index 2
14.	Index 2	$d_1 = 1.000$	Index 3
15.	Index 3	$Y = 0.93$	$W = 5000$

16. $q'_h = 44\,000$ scfh .... page B-2
17. $\mu = 0.018$ .... page A-5
18. $R_e = 860\,000$ or 8.6×10^5 .... page 3-2
19. $C = 1.02$ is correct for $R_e = 8.6 \times 10^5$ ... page A-20
20. When the C factor assumed in Step 9 is not in agreement with page A-20, for the Reynolds number based on the calculated flow, it must be adjusted until reasonable agreement is reached by repeating Steps 9 through 19.



Example 2

Given: A differential pressure of 3 psi is measured across taps located 1 diameter upstream and 0.5 diameter downstream from the inlet face of a 0.750-inch I.D. square edged orifice assembled in 1-inch Schedule 40 wrought iron pipe, in which, dry ammonia (NH_3) gas is flowing at 40 psig pressure and 50 F.

Find: The flow rate in pounds per second and in cubic feet per minute at standard conditions (scfm).

Solution:

1. $R = 90.8$
2. $S_g = 0.587$ .... for NH_3 gas; page A-8
3. $k = 1.19$

Steps 3 through 7 are used to determine the Y factor.

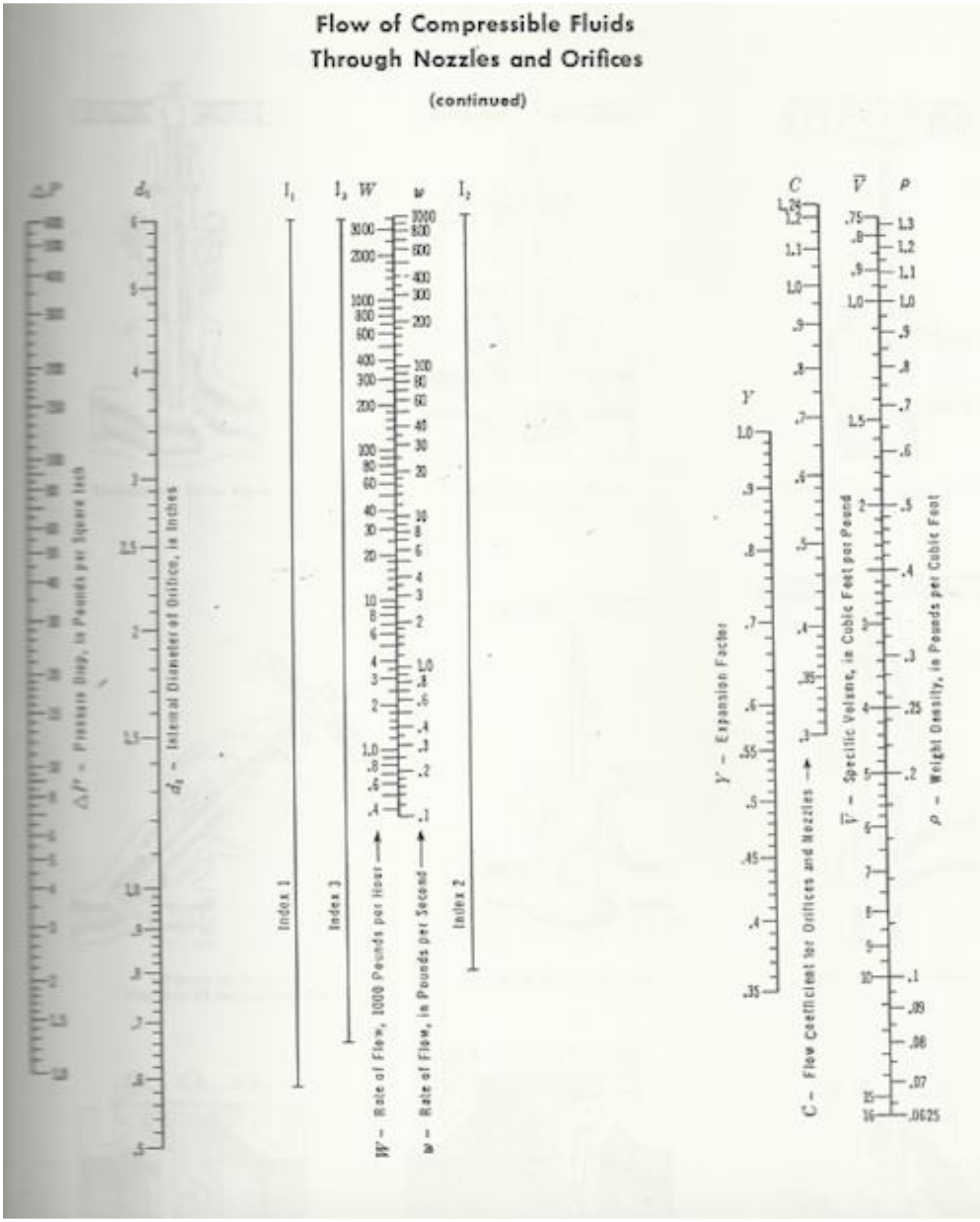
4. $P'_1 = P + 14.7 = 40 + 14.7 = 54.7$
5. $\Delta P / P'_1 = 3.0 \div 54.7 = 0.0549$
6. $d_2 = 1.049$ .... 1" Sched 40 pipe; page B-16
7. $\beta = 0.750 \div 1.049 = 0.716$
8. $Y = 0.98$ .... page A-21
9. $C = 0.71$ .. turbulent flow assumed; page A-20
10. $T = 460 + t = 460 + 50 = 510$
11. $\rho_1 = 0.17$ .... page A-10

	Connect		Read
12.	$\Delta P = 3.0$	$\rho_1 = 0.17$	Index 1
13.	Index 1	$C = 0.71$	Index 2
14.	Index 2	$d_1 = 0.75$	Index 3
15.	Index 3	$Y = 0.98$	$w = 0.145$
16.	Index 3	$Y = 0.98$	$W = 520$

17. $q'_m = \dfrac{W}{4.58\, S_g} = \dfrac{520}{4.58 \times 0.587} = 195$ .. page B-2
18. $\mu = 0.010$ .... page A-5
19. $R_e = 310\,000$ or 3.10×10^5 ...... page 3-2
20. $C = 0.71$ is correct for $R_e = 3.10 \times 10^5$ .. page A-20
21. When the C factor assumed in Step 9 is not in agreement with page A-20, for the Reynolds number based on the calculated flow, it must be adjusted until reasonable agreement is reached by repeating Steps 9 through 20.

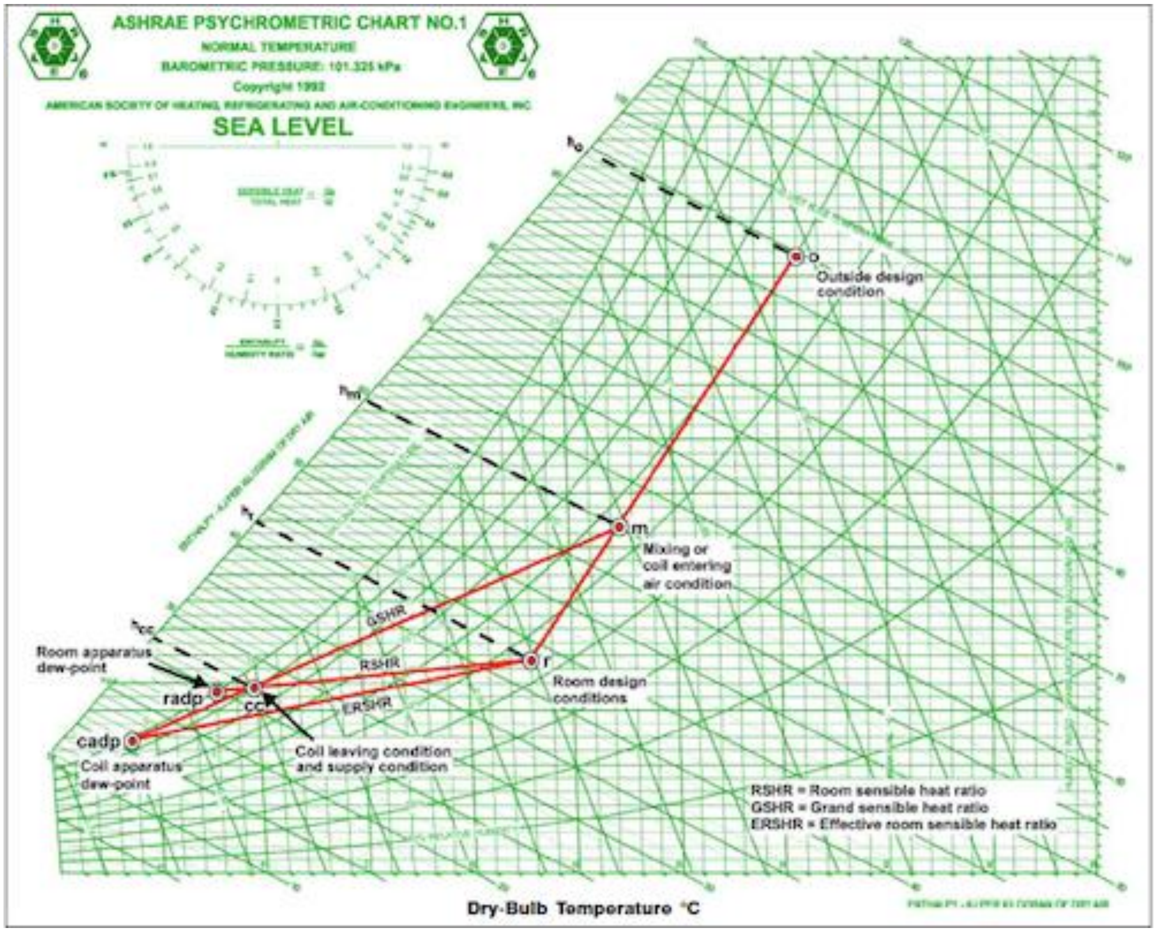
Reference: Crane Engineering, US publication

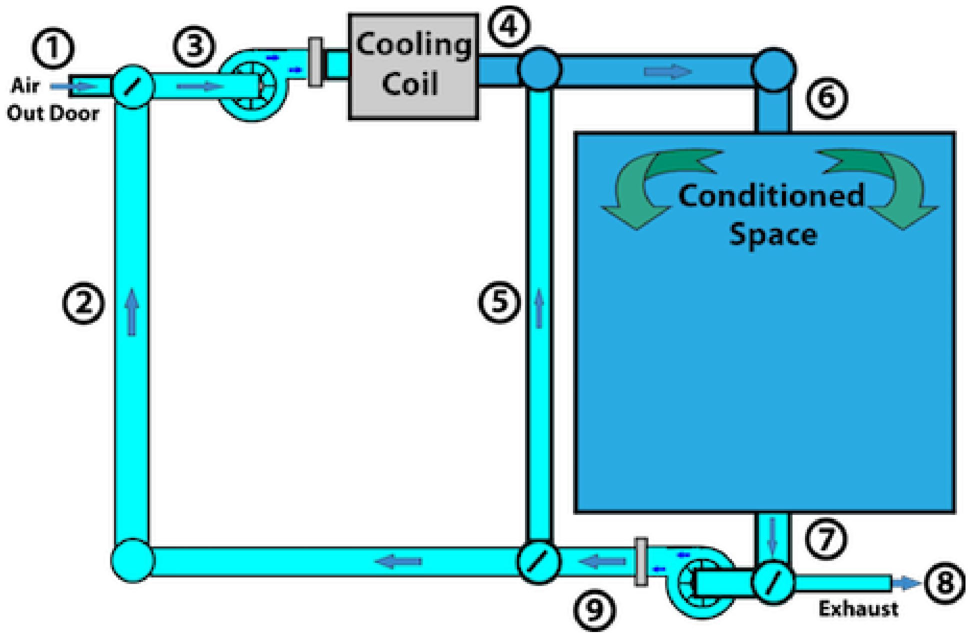
15.2.7 Nomogram for compressible flow through orifices



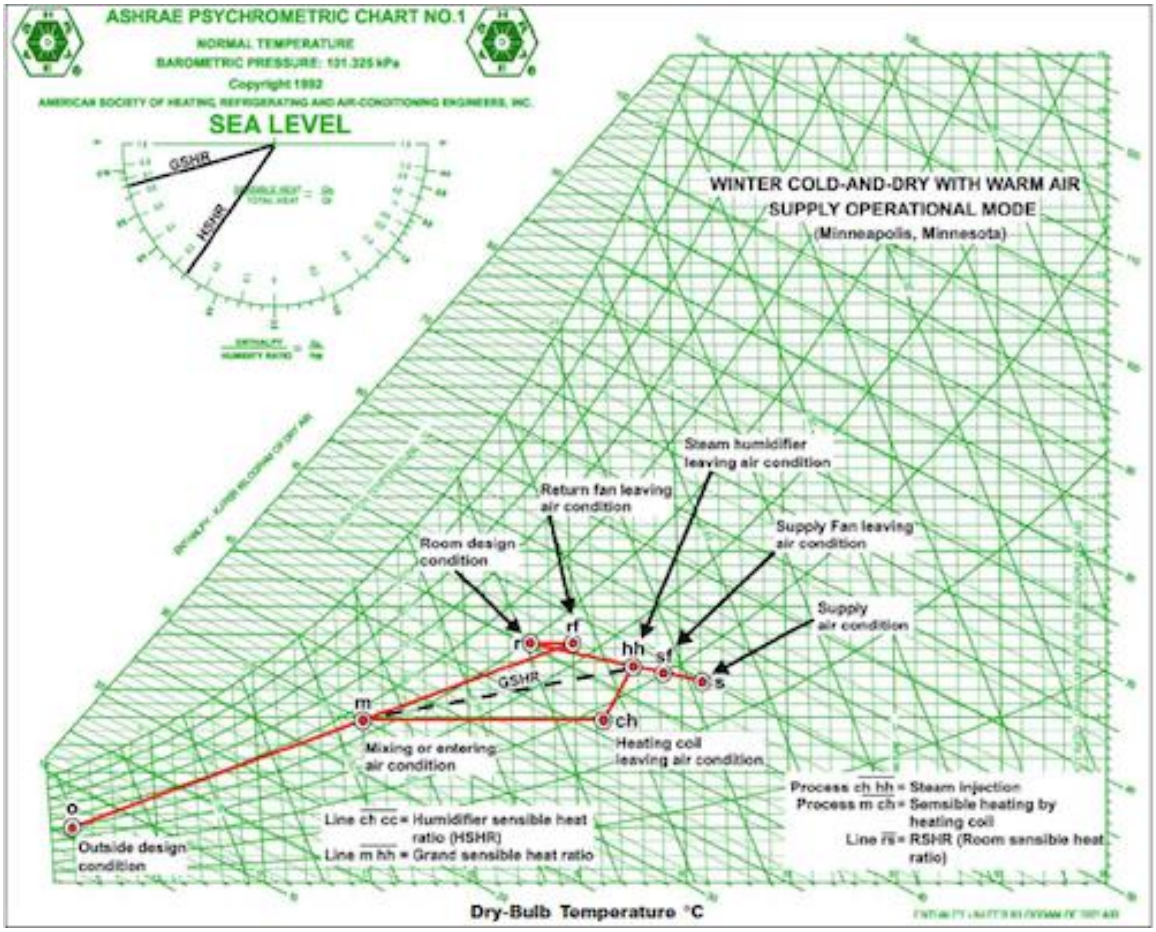
Reference: Crane Engineering, US publication

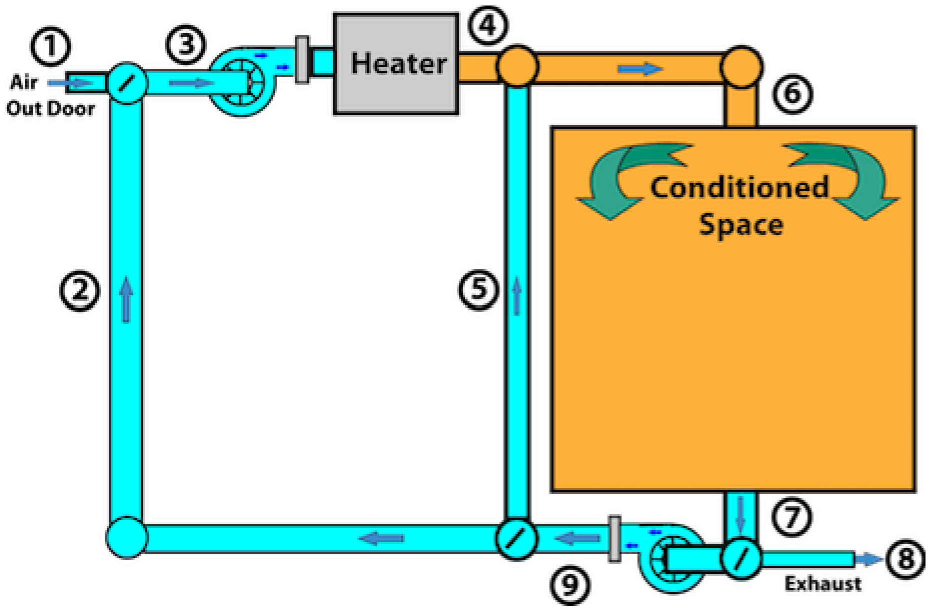
15.2.7. ASHRAE Cooling model with Psychrometrics



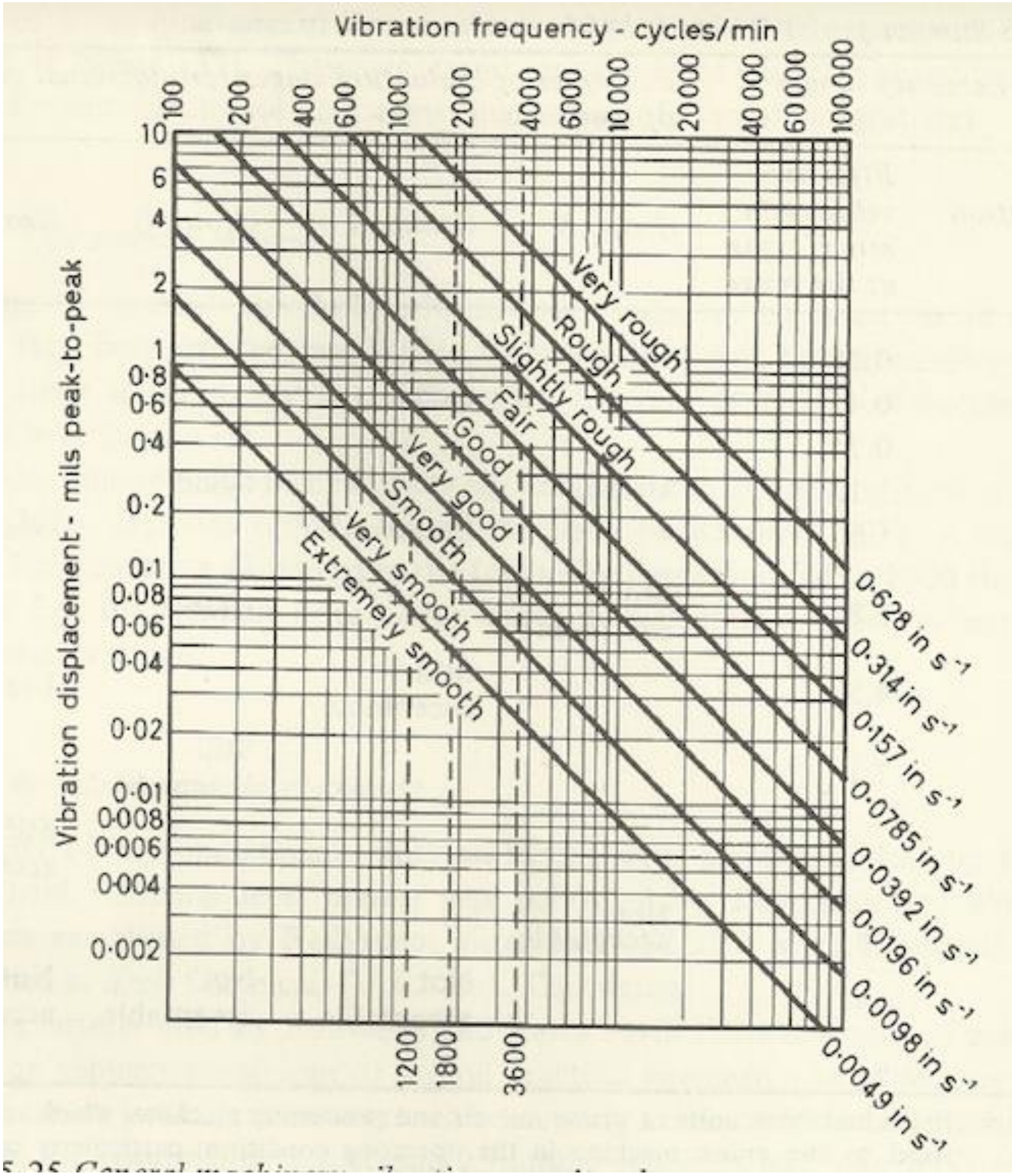


15.2.8. ASHRAE Heating model with Psychrometrics





15.2.10. General machinery vibration severity chart



Reference: R.A.Collacott

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15.4 Index

About Author:

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Belong to a class of engineers (KREC-1974) when there were no calculators existed. After initial exposure on manufacturing of precision machine tool components at HMT ancillary, quest for further advancement got into National Productivity Council's (NPC) pioneering institution for productivity and industrial engineering in 1977. It was a 15 long years of gratifying experience with constant enrichment of professional skill development as well as serving and contributing to the growth of productivity in various industrial sectors. In 1992, dawn of Personal Computers (PC) had already begun and that is when migrated to US for advancement of self and family. Did masters with focus was on computer aided design, modeling and simulation techniques for fluid power systems as received initial exposure on this subject during career in NPC as a part of professional training in Sweden.

As computers becoming more and more powerful with 32-bit processors, a lot of product development software tools started to emerge in which HyPneu was one of them where was associated for about five years. This was used in many proto-type design validation projects like a virtual laboratory cutting down the development time into half. In early 80's a new car model would take about 4 years to develop that was cut into half by the end of 90's. Advent of Internet everything started to grow exponentially like a second industrial revolution. This has followed by iPhone technology in 2007 and now everyone uses like a handheld computer. Hence, published smart phone apps splitting the old PC version discussed in this manual and guide. Here is a short intro on author's curricula vitae.