TAMPERE UNIVERSITY OF TECHNOLOGY

TONI HAKALA

CALCULATION TOOL FOR FAN COIL UNIT

Master of Science Thesis

Examiner: Professor Hannu Ahlstedt
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ABSTRACT

TAMPERE UNIVERSITY OF TECHNOLOGY
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The aim of this Master of Science Thesis was to develop an MS Excel-based calculation tool for cooling and heating capacities in Halton Group´s new product, the Fan Coil Unit. The Fan Coil Unit is commonly used in passenger cabins of cruisers to cool or heat the cabin air. The number of customer orders of the Fan Coil Unit has been increasing, therefore, Halton Group's subdivision Marine had a need to get this tool to save time, which had earlier gone to calculations for each unit order separately.

The main components of Fan Coil Unit are a fan, a coil, a filter, a drip pan and an electric heater. The operating principle of the Fan Coil Unit is that circulated cabin air will be cooled in coil section by using water as refrigerant. After the coil, cooled underinflated air goes through the fan section and gets slightly overpressured. Then the air goes through the electric heater section, which is usually turned off in case of cooling. After the electric heater, the circulated air duct is combined with a primary air duct to form a chamber, wherein the circulated air from the cabin and pre-cooled primary air are mixed. This mixture is called the supply air. Flow rate of primary air is usually around 30 per cent of total airflow rate. Thereafter, supply air is blown into the top of the cabin. Supply air temperature is usually 5-10 degrees lower than cabin temperature.

Cooling demand consists of two kinds of heat loads: latent and sensible heat load. The part of the heat load which is caused by evaporation, is called the latent heat load. The latent heat load comes from for example the cabin, when the passenger's skin is sweating and from the shower facilities when water vapor is released into the cabin air. Sensible heat is heat exchanged by a body or thermodynamic system that has as its sole effect a change of temperature.

This Thesis is divided into two parts. The first part will offer a basic theoretical background for the features that will be dealt with in the second part. The basic principles of Fan Coil Unit, heat transfer properties between air and water and other mathematical functions that have been used in this work are presented in the first part. The structure of the calculation tool and operations manual, are presented in the second part.

As a result of this work, the calculation tool was developed and found to be appropriate for designing as well as improving the Fan Coil Unit.

TIIVISTELMÄ

TAMPEREEN TEKNILLINEN YLIOPISTO
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Tämän Diplomityön tarkoitus oli kehittää Microsoft Excel-pohjainen jäähdytys- ja lämmityskapasiteettien laskentatyökalu Halton -konsernin uudelle tuotteelle, puhallinkonvektorille (Fan Coil Unit). Puhallinkonvektoria käytetään risteilijöiden matkustajahyteissä, olosuhteista riippuen joko jäähdyttämään tai lämmittämään hytti-ilmaa. Työkalu tuli tarpeeseen, koska uuden tuotteen tilausmäärät ovat kasvaneet ja näin ollen myös käsin laskemiseen kuluva aika on kasvanut niin suureksi, ettei se ole enää ollut yritykselle kannattavaa.

Puhallinkonvektorin pääkomponentit ovat puhallin (fan), jäähdytyspatteri (coil), suodatin (filter), tippakaukalo (drip pan) sekä sähkövastus (electric heater). Puhallinkonvektorin idea jäähdytystapauksessa on, että hytti-ilma kulkeutuu laitteessa olevan jäähdytyspatterin läpi, jossa jäähdyttävänä materiana kiertävä vesi vastaanottaa lämpöenergiaa kiertoilmasta, saaden näin kiertoilman lämpötilan laskemaan. Tämän jälkeen jäähdytetty kiertoilma kulkee puhaltimen ja lämmitysvastuksen läpi kammioon, jossa siihen sekoittuu konvektorin erillisestä liitoksesta tuleva esijäähdytetty tuoreilma. Yleensä tämän tuoreilman määrä on noin 30 prosenttia koko kierrätettävän ilman määrästä. Lopuksi kiertoilman ja tuoreilman seos puhalletaan hytin yläosaan 5-10 °C hyttilämpötilaa alemmassa lämpötilassa.

Jäähdytystarve koostuu kahdenlaisista lämpökuormista, latentista ja tuntuvasta lämpökuormasta. Latenttilämmöllä tarkoitetaan höyrystymisen aiheuttamaa lämpömäärää. Höyrystymistä tapahtuu muun muassa ihmisten iholla normaalin hikoilun vaikutuksesta ja suihkutiloissa, joista vapautuu vesihöyryä hytti-ilman sekaan nostaen näin hytti-ilman suhteellista kosteutta. Tuntuvalla lämmöllä vastaavasti tarkoitetaan hytti-ilman ja jäähdytetyn ilman lämpötilaeroa, joten sen määrä voidaan mitata suoraan lämpötilamittarilla.

Työ jakaantuu kahteen osaan. Kirjallisuusosassa selvitetään puhallinkonvektorin toimintaa ja perusperiaatteita, ilman ja veden välisiä lämmönsiirto-ominaisuuksia, sekä muita työhön liittyviä matemaattisia kaavoja. Työn tutkimusosassa käydään läpi laskentatyökalun rakennetta, toimintaa ja käyttöohjetta.

Tämän työn tuloksena voidaan todeta laskentatyökalun olevan sille asetettuihin tavoitteisiin soveltuva ja hyödyllinen puhallinkonvektorin kehittämisessä.

PREFACE

This Master of Science Thesis has been carried out at the Halton Corporation in Lahti, Finland. The supervisor and examiner of the thesis have been Professor Hannu Ahlstedt.

First of all, I would like to thank Professor Hannu Ahlstedt for providing me this interesting topic and for his guidance. I also want to thank personnel in Halton Marine and specially Jukka Maksimainen and Fabrice Deldreve for their help, because this thesis would not have been possible to make without them.

My last gratitude goes to my family, to whom I am very grateful for the sacrifices they have made in allowing me to study and for supporting and encouraging me in every decision during this thesis work.

Thank you!

Tampere, February 2012

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LIST OF ABBREVIATIONS AND TERMS

Δp	Air pressure drop [Pa]
Δt	Air temperature change [°C]
β_{coil}	Contact factor of coil [-]
ε	Effectiveness [-]
η	Efficiency [-]
ϕ	Fin parameter [-]
λ	Thermal conductivity [W/(m×K)]
δ_f	Fin thickness [m]
ρ_{air}	Density of air [kg/m^3]
ρ_{water}	Density of water [kg/m^3]
ν	Kinematic viscosity [m^2/s]
$\dot{v}$	Volumetric flow rate [m^3/s]
A	Area [m^2]
A_f	Face area of coil [m^2]
A_r	Total external surface area per tube row [m^2]
A_t	Total external surface area of the coil [m^2]
C	Heat capacity [J/K]
C_p	Specific heat at constant pressure [J/(kg×K)]
C_r	Critical heat capacity rate of fluids [-]
d	Diameter [m]
g	Moisture content [kg_{water}/kg_{air}]
h	Enthalpy [kJ/kg]
h_a	Heat transfer coefficient on the air side [W/(m^2×K)]
h_{fg}	Latent heat of evaporation [kJ/kg_{water}]
h_g	Enthalpy of water vapour [kJ/kg]
$LMTD_{as}$	Logarithmic mean temperature difference between the air stream and the mean coil surface temperature [°C]
$LMTD_{aw}$	Logarithmic mean temperature difference between the air stream and the water flow [°C]
$\dot{m}$	Mass flow rate [kg/s]
m_{cond}	Condensed water [kg_{water}/kg_{air}]
M	Molecular mass [g/mol]
NTU	Number of Transfer Units [-]
p_{atm}	Barometric pressure [Pa]
p_s	Pressure of water vapour [Pa]
p_{ss}	Saturation vapour pressure [Pa]
P_{fan}	Power of fan [W]
P_{lat}	Latent capacity [W]
P_{sen}	Sensible capacity [W]
P_{water}	Total capacity of water [W]
Q , q	Rate of heat transfer [W]
r	Radius [m]
r_f	Circular fin radius [m]
R	Thermal resistance [m^2×K/W]
R_a	Thermal resistance of the air film [m^2×K/W]
Re	Reynolds number [-]
R_f	Thermal resistance of the fins [m^2×K/W]

R_t	Thermal resistance of the tubes [m^2×K/W]
R_w	Thermal resistance of the water film [m^2×K/W]
$RH\%$, Φ	Humidity ratio of the air [%]
R_o	Universal gas constant [J/(kmol×K)]
$Rows$	Amount of rows in coil [-]
S	Sensible heat/total heat -ratio of coil [-]
t_d	Dew-point temperature of an air stream [°C]
T_{out}	Outside air temperature [°C]
T_{room}	Room temperature [°C]
T_{sm} , t_c	Mean coil surface temperature [°C]
T_{supply}	Supply air temperature [°C]
T_{wa}	Entering water temperature [°C]
T_{wb}	Leaving water temperature [°C]
T_{wm}	Mean water temperature [°C]
U_{coil}	U-value of coil [W/m^2]
v	Velocity [m/s]
v_f	Face velocity of coil [m^2/s]
V_m	Molar volume [m^3/kmol]

ASHRAE	American Society of Heating, Refrigerating and Air-conditioning Engineers
CIBSE	The Chartered Institution of Building Services Engineering
FCU	Fan Coil Unit
HVAC	Heating, Ventilation, and Air Conditioning
VBA	Visual Basic for Applications

1 INTRODUCTION

1.1 Fan Coil Unit

A Fan Coil Unit (abbreviated as FCU) is a simple device consisting of a heating or cooling coil and fan. It is part of an HVAC (Heating, Ventilation, and Air Conditioning) system found in residential, commercial, and industrial buildings. Typically a Fan Coil Unit is not connected to ductwork, and is used to control the temperature in the space where it is installed, or serve multiple spaces. It is controlled either by a manual on/off switch or by thermostat.

Due to their simplicity, Fan Coil Units are more economical to install than ducted or central heating systems with air handling units. However, they can be noisy because the fan is within the same space. Unit configurations are numerous including horizontal (ceiling mounted) or vertical (floor mounted). A Fan Coil Unit may be concealed or exposed within the room or area that it serves.

An exposed Fan Coil Unit may be wall mounted, freestanding or ceiling mounted, and will typically include an appropriate enclosure to protect and conceal the Fan Coil Unit itself, with return air grille and supply air diffuser set into that enclosure to distribute the air.

A concealed Fan Coil Unit will typically be installed within an accessible ceiling void or services zone. The return air grille and supply air diffuser, typically set flush into the ceiling, will be ducted to and from the Fan Coil Unit and thus allows a great degree of flexibility for locating the grilles to suit the ceiling layout and/or the partition layout within a space. It is quite common for the return air not to be ducted and to use the ceiling void as a return air plenum.

The coil receives hot or cold water from a central plant, and removes heat from or adds heat to the air through heat transfer. Traditionally Fan Coil Units can contain their own internal thermostat, or can be wired to operate with a remote thermostat.

Fan Coil Units circulate hot or cold water through a coil in order to condition a space. The unit gets its hot or cold water from a central plant, or mechanical room containing equipment for removing heat from the closed-loop. The equipment used can consist of machines used to remove heat such as a chiller and equipment for adding heat to the building's water such as a boiler or a commercial water heater.

Fan Coil Units are divided into two types: Two-pipe Fan Coil Units or four-pipe Fan Coil Units. Two-pipe Fan Coil Units have one supply-, and one return pipe. The supply pipe supplies either cold or hot water to the unit depending on the time of year. Four-pipe Fan Coil Units have two supply pipes and two return pipes. This allows either hot or cold water to enter the unit at any given time. Since it is often necessary to heat and cool different areas of a building at the same time, due to differences in internal heat loss or heat gains, the four-pipe Fan Coil Unit is most commonly used.

Depending upon the selected chilled water temperatures and the relative humidity of the space, it is likely that the cooling coil will dehumidify the entering air stream, and as a by product of this process, it will at times produce a condensate which will need to be carried to drain. The Fan Coil Unit will contain a purpose designed drip pan with drain connection for this purpose.

Speed control of the fan motors within a Fan Coil Unit is effectively used to control the heating and cooling output desired from the unit. Some manufacturers accomplish speed control by adjusting the taps on an AC transformer supplying the power to the fan motor. [6]

1.2 Calculation Tool

The calculation tool was developed by using a Microsoft Excel –program. It has been developed to facilitate cooling and heating capacity calculations in Fan Coil Unit, as well as helping with other Fan Coil Unit design problems. The tool was divided to six different pages. Pages are named *FCU, Fan, Coil, Prints, Main,* and *Charts*. All adjustments for FCU can be done from the first page, *FCU*. Therefore it is the most important page when using the tool. *Fan* -page is for managing different fans in the tool, as well as *Coil* –page is for managing different coils. All of the most important prints are seen in *Prints* –page, and it is possible to print out different kind of prints if needed. *Main* –page contains most of the calculations that have been used in the tool. *Charts* –page contains charts, which are related to the tool.

2 HALTON GROUP

Halton is passionate about indoor environments. They offer business-enhancing products, systems, and services for comfortable, energy-efficient, and safe environments to customers who value people's wellbeing. Halton is involved from target-setting to facility use and focuses on creating positive indoor environment experiences for people.

Halton solutions range from public and commercial buildings to industry, commercial kitchen and restaurant applications. Halton is also one of the most recognized indoor climate solution providers for marine and offshore applications. Areas of expertise and product ranges cover air diffusion, airflow management, fire safety, kitchen ventilation, air purification and indoor environment management. [11]

Halton Group figures:

- Family-owned Group has founded 1969
- International company, own operations in 19 countries
- Factories in Finland, France, United Kingdom, Hungary, Norway, USA, Canada, Malaysia, China and Germany
- Sales was 146 million Euros in 2011
- Employs more than 1000 people globally

Halton Group’s structure consists of five strategic business areas:

- Halton Indoors concentrates on indoor climate solutions for public buildings, special emphasis being on solutions for office, hotel and health facilities.
- Halton Foodservice provides indoor climate solutions for commercial kitchens and restaurants.
- Halton Marine offers solutions for safety and comfort aboard ships and offshore installations.
- Halton Clean Air manufactures advanced air cleaning solutions for building industry and private homes.
- Halton New Ventures provides solutions for indoor environment management.

2.1 Halton Marine

Halton Marine offers the latest technology for cabin and galley ventilation, fire safety, airflow management and air distribution systems. They are one of the world's leading suppliers of HVAC for marine, and focus on solutions that provide the highest standards of safety and comfort aboard cruise ships, navy and offshore installations.

Halton Marine sales offices are located to Finland, France, China, Norway and USA. Factories are located to Finland and China, and distributors they have in 20 countries.

Marine subdivision has divided to three different segments:

- Cruise & Ferry
- Oil & Gas and
- Navy

Halton Marine products and solutions have been designed specifically for marine, offshore and naval applications. Good indoor air quality in demanding conditions result of paying attention to many details and their successful adjustment. Regular, controlled indoor air and effective air-conditioning improve comfort, efficiency and ensure safety.

Investment in Halton Marine's quality products and leading-edge technical and functional solutions are always profitable in the long run. In the following list has listed some of Halton Marine’s main products. [11]

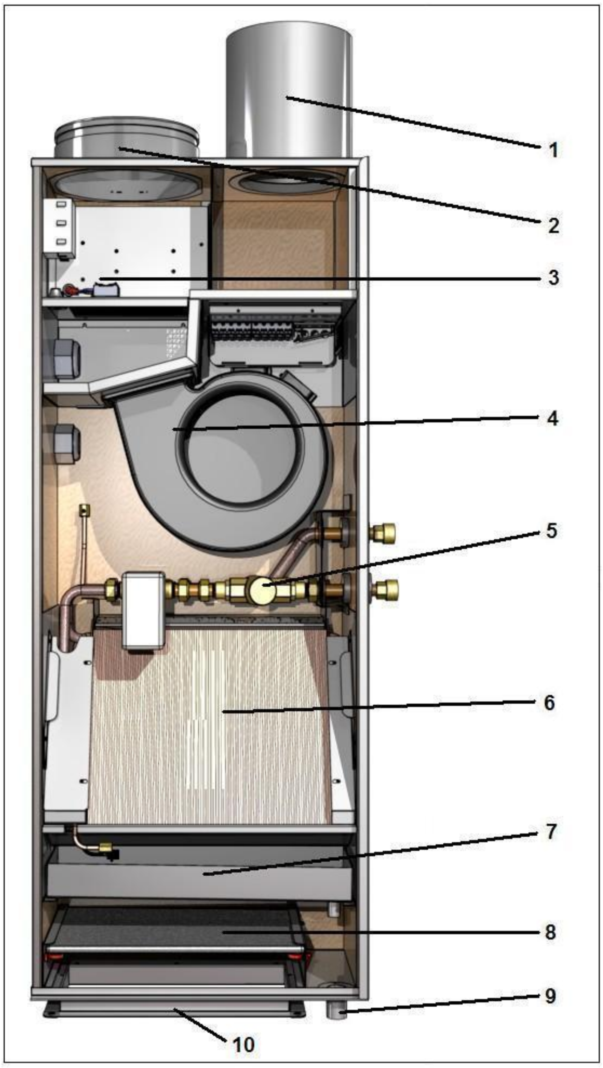
- Marine & Offshore fire dampers
- Flow control dampers
- Non-return and pressure relief dampers
- Blast valves
- Galley water wash hoods
- Galley hoods and canopies
- Cabin units and Fan Coil Units
- Droplet separators
- Outdoor louvres
- Valves
- Diffusers
- Grilles

2.1.1 Halton Fan Coil Unit

The Halton Fan Coil Unit is a cabin ventilation solution for demanding marine applications. Fan Coil Unit has been specifically designed for silent cabin comfort with sophisticated air treatment and control. An advanced digital room temperature system with stepless fan speed control and cooling/heating power regulation completes the solution. The manufacturing method and innovative, compact design allow Fan Coil Units to be modified for any situation.

The operating principle of the Fan Coil Unit is that circulated cabin air will be cooled in coil section by using water as refrigerant. Commonly, inlet and outlet water temperatures are about 7 °C and 12 °C, while cabin air is set to 22-24 °C. After the coil section, cooled underinflated air passes through the fan section and gets slightly overpressured. Then air passes through electric heater section, which is usually turned off in case of cooling. After the electric heater, the circulated air duct is combined with a primary air duct to form a chamber, wherein the circulated air from the cabin and pre-cooled primary air are mixed. This mixture is called the supply air. In supply air, the ratio of the overall amount of primary and circulated air are about 30 and 70 per cents. Finally, supply air is blown with fan into the top of the cabin. Supply air temperature is recommended to be 5-10 degrees lower than cabin temperature.

The main components and sections of Fan Coil Unit are presented in Figure 2.1.



1. Primary air inlet
2. Supply air duct
3. Electric heater
4. Fan
5. Water valve
6. Coil
7. Drip pan
8. Filter
9. Condense water outlet
10. Grille

Figure 2.1. *The components of Fan Coil Unit.*

Technical data for Halton Fan Coil Unit:

- Fan Coil Unit is capable of distributing airflows from 150 m^3/h to 500 m^3/h.
- Operating voltage 230 VAC +-10 %, max. 8 A, 50/60 Hz.
- Tubes and fins of coil made of copper.
- Galvanized steel casing and mineral wool insulation.
- Silent and stepless fan operation.
- Electric heaters: 400 W + 800 W.
- Total measured cooling capacity: up to 1470 W.
- Quick water connections and integrated electric connections.
- Air connections tailored according to customer needs.
- Dimensions: 430 mm x 960 mm x 240 mm.
- Weight: 32 kg.

3 THEORY OF HEAT TRANSFER

All equations and functions, which are used in calculation tool, are presented in this part. First of all, has presented some of the basic laws related to air and water vapour mixtures. Psychrometry of air conditioning processes, comfort and design conditions, air cooler coils, airflow in ducts and fan performance are presented as well.

3.1 Properties of air and water vapour mixtures

The most important thing for the student of psychrometry to understand from the outset is that the working fluid under study is a mixture of two different gaseous substances. One of these, dry air, is itself a mixture of gases, and the other, water vapour, is steam in the saturated or superheated condition. Some of the most important standards are,

- Density of air, ρ_{air} 1,296 kg/m^3 for dry air at 101325 Pa and 0 °C.
- Density of water, ρ_{water} 999,9 kg/m^3 at 0 °C and
- Barometric pressure, p_{atm} 101325 Pa in 0 °C. [3][10]

3.1.1 The general gas law

The general gas law is expressed as:

$$p \times V = m \times R \times T \tag{3.1.1}$$

where *p* is the pressure of the gas in *Pa*, *V* is the volume of the gas in m^3, *m* is the mass of the gas in *kg*, *R* is a specific gas constant in *J/(kg×K)* and *T* is the absolute temperature of the gas in *K*.

Avogadro´s hypothesis argues that equal volumes of all gases at the same temperature and pressure contain the same number of molecules. Accepting this and taking as the unit of mass the kilomole *(kmol)*, a mass in kilograms numerically equal to the molecular mass of the gas, a value for the universal gas constant can be established:

$$p \times V_m = R_o \times T \tag{3.1.2}$$

where V_m is the molar volume in $m^3/kmol$ and is the same for all gases having the same values of *p* and *T*. Using the values *p* is *101325 Pa* and *T* is *273,15 K*, it has been experimentally determined that V_m equals *22,41* $m^3/kmol$. Hence the universal gas constant, R_o, is determined

$$R_o = \frac{p \times V_m}{T} = \frac{101325\,Pa * 22{,}41 m^3 / kmol}{273{,}15\,K} = 8314{,}41 \frac{J}{kmol \times K}$$

Specific gas constants of dry air and steam are expressed as

$$R_a = \frac{R_o}{M_a} = \frac{8314{,}41}{28{,}97} = 287 \frac{J}{kg \times K} \tag{3.1.3}$$

$$R_s = \frac{R_o}{M_s} = \frac{8314{,}41}{18{,}02} = 461 \frac{J}{kg \times K} \tag{3.1.4}$$

where M_a and M_s are molecular masses of dry air and steam. A suitable transposition of the general gas law gives expressions for density, pressure and volume. [10][15]

3.1.2 Dalton's law of partial pressure

Dalton’s law may be stated as follows:

If a mixture of gases occupies a given volume at a given temperature, the total pressure exerted by the mixture equals the sum of the pressures of the components, each being considered at the same volume and temperature.

It is possible to show that if Dalton's law holds, each component of the mixture obeys the general gas law. As a consequence, it is sometimes more convenient to re-express the law in two parts: [3][10]

(1) The pressure exerted by each gas in a mixture of gases is independent of the presence of the other gases, and

(2) The total pressure exerted by a mixture of gases equals the sum of the partial pressures.

3.1.3 Saturation vapour pressure

There are two requirements for the evaporation of liquid water to occur:

- Thermal energy must be supplied to the water, and
- The vapour pressure of the liquid must be greater than that of the steam in the environment.

These statements need some explanation.

Molecules in the liquid state are comparatively close to each other. They are nearer to each other than are the molecules in a gas and are less strongly bound together than those in a solid. The three states of matter are further distinguished by the extent to which an individual molecule may move. At a given temperature, a gas consists of molecules which have high individual velocities and which are arranged in a random fashion. A liquid at the same temperature is composed of molecules, the freedom of movement of which is much less, owing to the restraining effect which neighbouring molecules have on each other, by virtue of their comparative proximity. An individual molecule, therefore, has less kinetic energy if it is in the liquid state than it has in the gaseous state. Modern thought is that the arrangement of molecules in a liquid is not entirely random as in a gas, but that it is not as regular as it is in most, if not all, solids.

It is evident that if the individual molecular kinetic energies are greater in the gaseous state, then energy must be given to a liquid when it is changing to the gaseous phase. This explains the first stated requirement for evaporation.

As regards the second requirement, the situation is clarified if one considers the boundary between a vapour and its liquid. Only at this boundary can a transfer of molecules between the liquid and the gas occur. Molecules at the surface have a kinetic energy, which has a value related to the temperature of the liquid. Molecules within the body of the gas also have a kinetic energy, which is a function of the temperature of the gas. Those gaseous molecules near the surface of the liquid will, from time to time, tend to hit the surface of the liquid, some of them staying there. Molecules within the liquid and near to its surface will, from time to time, also tend to leave the liquid and enter the gas, some of them staying there.

It has been found that water in an ambient gas which is not pure steam but a mixture of dry air and steam, behaves in a similar fashion, and that for most practical purposes the relationship between saturation temperature and saturation pressure is the same for liquid water in contact only with steam. One concludes from this a very important fact: saturation vapour pressure depends solely upon temperature. [10]

3.1.4 Moisture content and relative humidity

Moisture content is defined as the mass of water vapour in *kg*, which is associated with one kilogram of dry air in an air-water vapour mixture.

$$Moisture\,content\;\; m = \frac{m_s}{m_a} \tag{3.1.5}$$

Relative humidity is a term used to describe the amount of water vapour in a mixture of air and water vapour. It is defined as the ratio of the partial pressure of water vapour in the air-water mixture to the saturated vapour pressure of water at those conditions. The relative humidity of air depends not only on temperature but also on pressure of the system of interest. Relative humidity is often used instead of absolute humidity in situations where the rate of water evaporation is important, as it takes into account the variation in saturated vapour pressure. [4]

Relative humidity is defined as

$$Humidity\,ratio\;\; \Phi = \frac{p_s}{p_{ss}} \times 100\% \tag{3.1.6}$$

where p_s is a pressure of water vapour and p_{ss} is the saturation vapour pressure

3.1.5 Dew point and specific volume of the mixture

The dew point is the temperature to which a given parcel of humid air must be cooled, at constant barometric pressure, for water vapour condense into liquid water. The condensed water is called dew when it forms on a solid surface. The dew point is a saturation temperature.

The dew point is associated with relative humidity. A high relative humidity indicates that the dew point is closer to the current air temperature. Relative humidity of 100 % indicates the dew point is equal to the current temperature and the air is maximally saturated with water. When the dew point remains constant and temperature increases, relative humidity will decreases. [5]

Specific volume of the mixture is the volume in m^3 of one kilogram of dry air mixed with water vapour. In the mixture each component occupies the same volume and is at the same temperature, but each exerts its own partial pressure. By Dalton's law sum of these partial pressures is the total (barometric) pressure of the mixture. The general gas law, may be transposed to express the specific volume:

$$V = \frac{m \times R \times T}{p} \tag{3.1.7}$$

This equation could be used to refer to the dry air, or to the water vapour, independently if Dalton's law is accepted. In doing so, the appropriate values for the mass, specific gas constant and partial pressure of the constituent considered must be used. [10]

3.1.6 Enthalpy of moist and dry air

The enthalpy, *h*, used in psychrometry is the specific enthalpy of moist air, expressed in $kJ/kg_{dry\ air}$, defined by the equation:

$$h = h_a + g \times h_g \tag{3.1.8}$$

where h_a is the enthalpy of dry air, g is the moisture content in $kg/kg_{,dry\ air}$, and h_g is the enthalpy of water vapour, both expressed in kJ/kg. An approximate equation for the enthalpy of dry air over the range 0 °C to 60 °C is

$$h_a = 1.007 \times T - 0.026 \tag{3.1.9}$$

and the following equation can be used for the enthalpy of water vapour:

$$h_g = 2501 + 1.84 \times T \tag{3.1.10}$$

Equations (3.1.9) and (3.1.10) can now be combined, as typified by equation (3.1.8), to give an approximate expression for the enthalpy of humid air:

$$h = (1.007 \times T \text{ - } 0.026) + g \times (2501 + 1.84 \times T) \tag{3.1.11}$$

Equation (3.1.11) is valid at a barometric pressure (101325 Pa). [10]

3.2 The Psychrometry of Air Conditioning Processes

This chapter provides a picture of the way in which the state of moist air alters as an air conditioning process takes place or a physical change occurs. Familiarity with the psychrometric chart is essential for a proper understanding of air conditioning.

Any point on the chart is termed a state point, the location of which, at a given barometric pressure, is fixed by any two of the psychrometric properties. It is customary and convenient to design charts at a constant barometric pressure because barometric pressure does not alter greatly over much of the inhabited surface of the earth. [10]

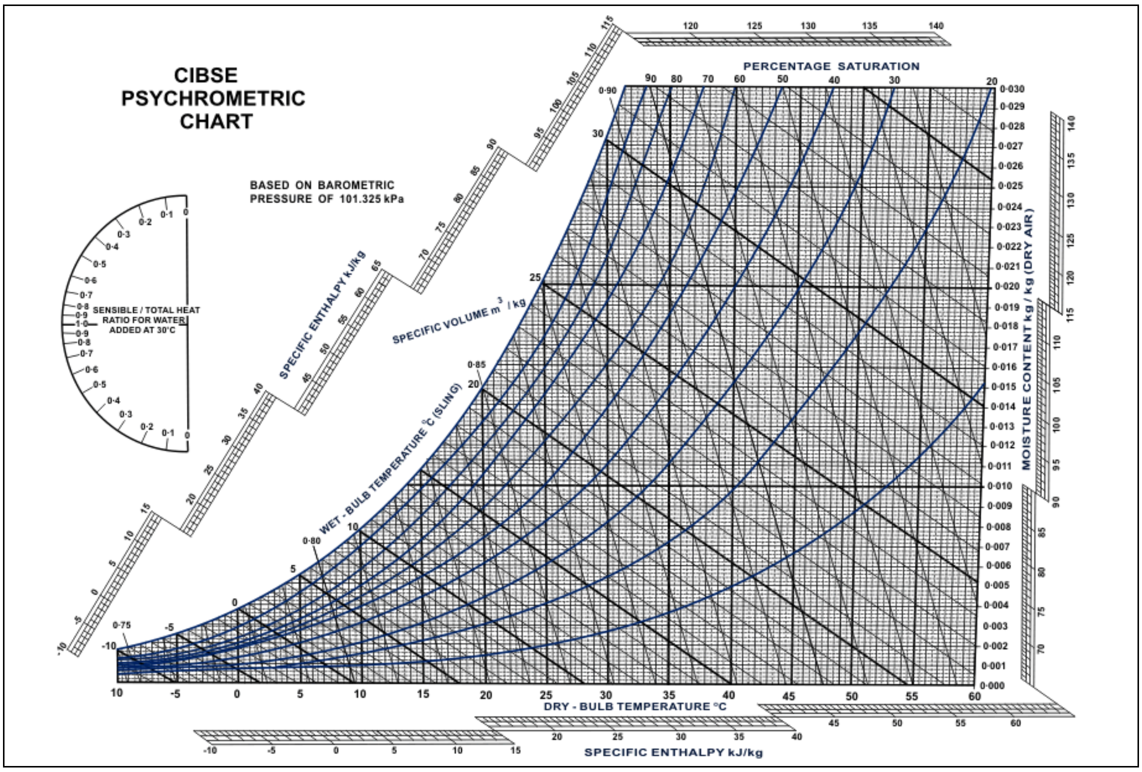


Figure 3.1. *Psychrometric chart by CIBSE. [16]*

The psychrometric chart published by the CIBSE (Figure 3.1) uses two fundamental properties, mass and energy, in the form of moisture content and enthalpy, as co-ordinates. As a result, mixture states lie on the straight line that joins the state points of the two constituents. Lines of constant dry-bulb temperature are virtually straight but divergent, only the isotherm for 30 °C being vertical. The reason for this is that to preserve the usual appearance of a psychrometric chart, in spite of choosing the two fundamental properties as co-ordinates, the co-ordinate axes are oblique, not rectangular. Hence, lines of constant enthalpy are both straight and parallel, as are lines of constant moisture content. Since both these properties are taken as linear, the lines of constant enthalpy are equally spaced as are the lines of constant moisture content. This is not true of the lines of constant humid volume and constant wet-bulb temperature, which are slightly curved and divergent. Since their curvature is only slight in the region of practical use on the chart, they can be regarded as straight without significant error resulting. In the sketches of psychrometric charts used throughout this text to illustrate changes of state, only lines of percentage saturation are shown curved. All others are shown straight, and dry-bulb isotherms are shown as vertical, for convenience. [10]

The chart also has a protractor, which allows the value of the ratio of the sensible heat gain to the total heat gain to be plotted on the chart. This ratio is an indication of the slope of the room ratio line and is of value in determining the correct supply state of the air that must be delivered to a conditioned space. The zero value for the ratio is parallel to the isotherm for 30 °C because the enthalpy of the added vapour depends on the temperature at which evaporation takes place, it being assumed that most of the latent heat gain to the air in a conditioned room is by evaporation from the skin of the occupants and that their skin surface temperature is about 30 °C. [10]

The psychrometric chart by CIBSE is such a system with oblique co-ordinates. For this chart then, a principle can be stated for the expression of mixture states. When two air streams mix adiabatically, the mixture state lies on the straight line which joins the state points of the constituents, and the position of the mixture state point is such that the line is divided inversely as the ratio of the masses of dry air in the constituent air streams. [10]

3.2.1 Sensible heating and cooling

Sensible heat transfer occurs when moist air flows over the coils of a sensible heater or cooler. In sensible cooling there is a following restriction: the lowest water temperature must not be so low that moisture starts to condense on the cooler coils. If such condensation does occur, through a poor choice of chilled water temperature, then the process will no longer be one of sensible cooling since dehumidification will also be taking place. [10]

The variations in the physical properties of the moist air, for the two cases, are summarized below:

	Sensible heating	Sensible cooling
Dry-bulb	*increases*	*decreases*
Enthalpy	*increases*	*decreases*
Humid volume	*increases*	*decreases*
Wet-bulb	*increases*	*decreases*
Percentage saturation	*decreases*	*increases*
Moisture content	*constant*	*constant*
Dew point	*constant*	*constant*
Vapour pressure	*constant*	*constant*

3.2.2 Dehumidification

There are four principal methods whereby moist air can be dehumidified:

(i) cooling to a temperature below the dew point,
(ii) adsorption,
(iii) absorption and
(iv) compression followed by cooling.

The first method forms the subject matter of this section.

Cooling to a temperature below the dew point has done by passing the moist air over a cooler coil provided with chilled water. Figure 3.2. shows on a sketch of a psychrometric chart what happens when moist air is cooled and dehumidified in this fashion. Since dehumidification is the aim, some part of the cooler coil must be at a temperature less than the dew point of the air entering the equipment. In the Figure 3.2, *td* is the dew point of the moist air and temperature *tc* corresponding to the point C on the saturation curve, this is termed the *mean coil surface temperature*.

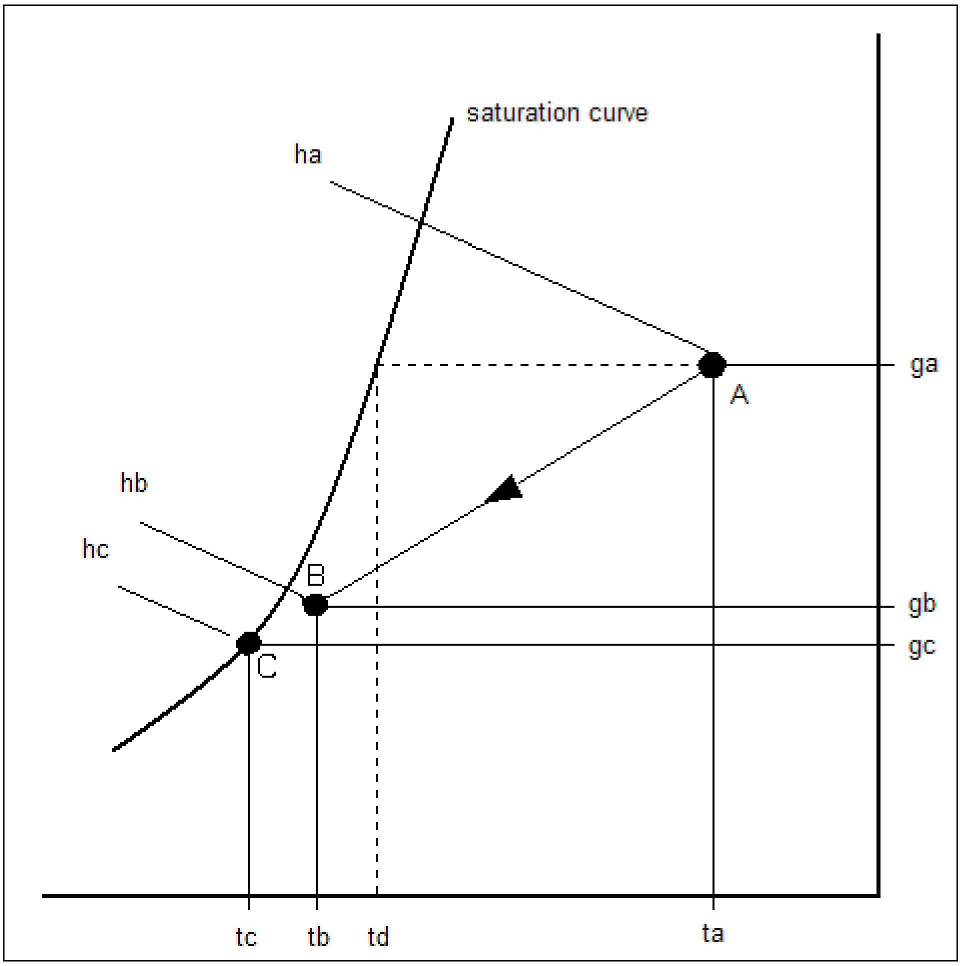


Figure 3.2. *Cooling and dehumidification by a cooling coil.*

For purposes of carrying out air conditioning calculations, it is sufficient to know the state A of the moist air entering the coil, the state B of the air leaving the coil, and the mass flow of the air.

It can be seen from Figure 3.2, that the moisture content of the air is reduced, as also is its enthalpy and dry-bulb temperature. The percentage saturation, of course, increases. Any of the cooler coil could not be 100 per cent efficient, which means that humidity on state B is never as much as 100 per cent. It is unusual to speak of the efficiency of a cooler coil. Instead, the alternative terms, contact factor and by-pass factor, are used. They are complementary values and contact factor, sometimes denoted by β, is defined as

$$\beta = \frac{g_a - g_b}{g_a - g_c} = \frac{h_a - h_b}{h_a - h_c} = \frac{t_a - t_b}{t_a - t_c} \tag{3.2.1}$$

Similarly, by-pass factor is defined as

$$(1-\beta) = \frac{g_b - g_c}{g_a - g_c} = \frac{h_b - h_c}{h_a - h_c} = \frac{t_b - t_c}{t_a - t_c} \tag{3.2.2}$$

Typical values of contact factor are 0.82 to 0.92 for practical coil selection. In hot, humid climates more heat transfer surface is necessary and higher contact factors are common. [10]

3.2.3 Cooling and dehumidification with reheat

When a cooler coil is used for dehumidification, the temperature of the moist air is reduced, but it is quite likely that under these circumstances this reduced temperature is too low. Although, we usually arrange that under conditions of maximum loads, both latent and sensible, the state of the air leaving the cooler coil is satisfactory. This is not so for partial load operation. The reason for this is that latent and sensible loads are usually independent of each other. Consequently, it is sometimes necessary to arrange for the air that has been dehumidified and cooled by the cooler coil to be reheated to a temperature consistent with the sensible cooling load; the smaller the sensible cooling load, the higher the temperature to which the air must be reheated. [10]

3.2.4 Pre-heat and humidification with reheat

Air conditioning units that handle primary air only may be faced in winter with the task of increasing both the moisture content and the temperature of the air they supply to the conditioned space. Humidification is needed because the outside air in winter has a low moisture content, and if this air were to be introduced directly to the room there would be a correspondingly low moisture content as well. The low moisture content may not be intrinsically objectionable, but when the air is heated to a higher temperature its relative humidity may become quite low. For example, outside air in winter might be at -1 °C, saturated. The moisture content at this state is only 3.484 $g/kg_{dry\ air}$. If this is heated to 20 °C dry-bulb, and if there is a moisture pick-up in the room of 0.6 $g/kg_{dry\ air}$, due to latent heat gains, then the relative humidity in the room will be as low as 28 %. This value may sometimes be seen as too low for comfort. The unit must increase the temperature of the air, either to the value of the room temperature if there is background heating to offset fabric losses, or to a value in excees of this if it is intended that the air delivered should deal with fabric losses. [10]

3.3 Comfort and Design Conditions

For comfort, indoor air quality may be said to be acceptable if not more than 50 % of the occupants can detect any odour and not more than 20 % experience discomfort, and not more than 10 % suffer from mucosal irritation, and not more than 5 % experience annoyance, for less than 2 % of the time. [10]

Air humidity affects the evaporation of water from the mucosal and sweating bodily surfaces, influencing its diffusion through the skin. Low humidities, with dew points less than 2 °C, tend to give a dry nose and throat, and eye irritation. A dusty environment can exacerbate low humidity skin conditions. [7][10]

3.3.1 The choice of inside design conditions

For a person to feel comfortable it appears that the following conditions are desirable:

(1) The air temperature should be between 22-24 °C.
(2) The average air velocity in the room should not exceed 0.15 m/s in an air conditioned room.
(3) Relative humidity should desirably lie between about 45 % and 60 %.
(4) The dew point should never be less than 2 °C.
(5) The temperature difference between the feet and the head should be as small as possible, normally not exceeding 1.5 °C.
(6) Floor temperatures should not be greater than 26 °C when people are standing and probably not less than 17 °C.
(7) The radiant temperature asymmetry should not be more than 5 °C vertically or 10 °C horizontally.
(8) The carbon dioxide content should not exceed about 0.1 %.

The two most important variables are dry-bulb temperature and air velocity, with mean radiant temperature of slightly less importance. Of these a comfort air conditioning system can only exercise direct automatic control over the dry-bulb temperature. A suitable choice of air velocity may be achieved by proper attention to the system of air distribution and acceptable values of mean radiant temperature should result from co-operation between the design engineer and the architect, aiming to eliminate objectionable radiant effects from sunlit windows in summer, cold windows in winter, cold exposed floors or walls, and excessive radiation from light fittings.

The choice of inside comfort design conditions for an air conditioned room depends on the physiological considerations already debated and on economic factors. The designer then examines the outside design state, the clothing worn by the occupants, their rate of working and the period of occupancy. An air temperature of 22 °C with about 50 % relative humidity is a comfortable choice for long-term occupancy by normally clothed, sedentary people but the humidity can be allowed to rise to 60 % or to fall towards 40 %, under conditions of peak summer heat gains if psychrometric, commercial or other practical considerations warrant it. [7][10]

3.3.2 Design temperatures and heat gains

The choice of inside and outside summer dry-bulb design temperatures affects the heat gains and hence influences the capital cost of the installation and its running cost, the latter implying the energy consumption of the system. Any change in the room design condition, particularly dry-bulb temperature, will have an effect on the comfort of the occupants. Similarly, any change in the chosen outside dry-bulb temperature will influence the system performance and the satisfaction given. Thus a relaxation of the outside design dry-bulb temperature to a lower value may give a small reduction in the sensible heat gains and the capital cost but this must be balanced against the fact that the system will only be able to maintain the inside design conditions for a shorter period of the summer. The relative merits of any such decisions must be carefully considered and the client advised. [10]

3.3.3 Sensible heat removal

If there is a continuous source of heat having an output of Q in a hermetically sealed room the temperature within room, T_{room} will rise until the flow of heat through the walls, of area A and thermal transmittance U, equals the output of the source:

$$Q = AU\,(T_{room} - T_{out}) \tag{3.3.1}$$

In which T_{out} is the outside air temperature. It then follows that,

$$T_{room} = T_{out} + Q/AU \tag{3.3.2}$$

and hence T_{room} will always exceed T_{out}. [10]

3.3.4 The specific heat capacity of humid air

The air supplied to a conditioned room in order to remove sensible heat gains occurring therein, is a mixture of dry air and superheated steam. It follows that these two gases being always at the same temperature because of the intimacy of their mixture, will rise together in temperature as both offset the sensible heat gain. They will, however, offset differing amounts of sensible heat because, first of all, their masses are different, and secondly, their specific heats are different too. [10]

Consider 1 kg of dry air with associated moisture content of g kg of superheated steam, supplied at temperature T_{supply} in order to maintain temperature T_{room} in a room in the presence of sensible heat gains of Q in kW. A heat balance equation can be written thus:

$$Q = 1 \times 1.012 \times (T_{room} - T_{\sup ply}) + g \times 1.890 \times (T_{room} - T_{\sup ply})$$

Where 1,012 and 1,890 are the specific heats at constant pressure of dry air and steam respectively. Rearrange the equation:

$$Q = (1.012 + 1.89 \times g) \times (T_{room} - T_{\sup ply}) \tag{3.3.3}$$

The expression (1.012+1.89×g) is sometimes called the specific heat of humid air. Taking into account the small sensible cooling or heating capacity of the superheated steam present in the supply air (or its moisture content) provides a slightly more accurate answer to certain types of problem. Such extra accuracy may not be warranted in most practical cases but it is worthy of consideration as an exercise in fundamental principles. [10]

3.3.5 Latent heat removal

If the air in a room is not at saturation, then water vapour may be liberated in the room and cause the moisture content of the air in the room to rise. Such a liberation of steam is effected by any process of evaporation as, for example, the case of insensible perspiration and sweating on the part of the people present. Since it is necessary to provide heat to effects a process of evaporation, it is customary to speak of the addition of moisture to a room as *kW* of latent heat rather than as *kg/s* of water evaporated. [10]

The heat gains occurring in a room can be considered in two parts: sensible gains and latent gains. The mixture of dry air and associated water vapour supplied to a room has therefore a dual role: it is cool enough initially to suffer a temperature rise up to the room dry-bulb temperature in offsetting the sensible gains, and its initial moisture content is low enough to permit a rise to the value of the room moisture content as latent heat gains are offset. [10]

If the mass of dry air supplied and its associated moisture content is known, then it is possible to calculate the rise in room moisture content corresponding to given latent heat gains:

$$Latent\ heat\ gain = m \times (g_{room} - g_{supply}) \times h_{fg} \tag{3.3.4}$$

where m is mass flow rate of supply air in $kg_{dry\ air}/s$, g_{room} and g_{supply} are the moisture contents of the room and supply air in $kg_{water}/kg_{dry\ air}$, and h_{fg} is the latent heat of evaporation in kJ/kg_{water}. [10]

3.3.6 Heat gain arising from fan power

The flow of air along a duct results in the air stream suffering a loss of energy. The energy dissipated through the ducting system is apparent as a change in the total pressure of the air stream and the energy input by the fan is indicated by the fan total pressure. [10]

Ultimately, all energy losses appear as heat (although, on the way to this, some are evident as noise, in duct systems). So an energy balance equation can be formed involving the energy supplied by the fan and the energy lost in the air stream. That is to say, the loss of pressure suffered by the air stream as it flows through the ducting system and past the items of plant (which offer a resistance to airflow) constitutes an adiabatic expansion which must be offset by an adiabatic compression at the fan. [10]

So, all the power supplied by the fan is regarded as being converted to heat and causing an increase in the temperature of the air handled, Δt, as it flows through the fan. A heat balance equation can be written:

$$\text{Air power} = \text{fan total pressure}(\text{N/m}^2) \times \text{volumetric flow rate}\,(\text{m}^3/\text{s})$$

The rate of heat gain corresponding to this is *the volumetric flow rate* $\times\ \rho \times C_p \times \Delta t$, where ρ and C_p are the density and specific heat of air, respectively. Hence

$$\Delta t = \frac{fan\ total\ pressure\ (N/m^2)}{\rho \times C_p}$$

The air quantities have cancelled, indicating that the rise in air temperature is independent of the amount of air handled, and using ρ is 1,296 kg/m^3 and C_p is 1007 $J/(kg \times K)$ we get

$$\Delta t = \frac{fan\ total\ pressure\ (N/m^2)}{1305} \tag{3.3.5}$$

Thus, the air suffers a temperature rise of 0,000766 *K* for each N/m^2 of fan total pressure. [3][10]

The energy the fan receives is in excess of what it delivers to the air stream, since frictional and other losses occur as the fan impeller rotates the air stream. The Power input to the fan shaft is termed the *Fan power* and the ratio of the air power to the fan power is termed the *total fan efficiency* and is denoted by η. Not all the losses occur within the fan casing. Some take place in the bearings external to the fan, for example. Hence, for the case where the fan motor is not in the air stream, full allowance should not be made. It is suggested that a compromise be adopted. [10]

If an assumption of 70 % is made for the fan total efficiency and if it is assumed that, instead of 30 %, only 15 % of the losses are absorbed by the air stream (since some are lost from the fan casing and the bearings) equation (3.3.5) becomes

$$\Delta t = \frac{fan\ total\ pressure\ (N/m^2)}{1305\ \times\ 0{,}85}$$

$$\Delta t = \frac{fan\ total\ pressure\ (N/m^2)}{1109} \tag{3.3.6}$$

Thus almost one thousandth of a degree rise in temperature for each N/m^2 of fan total pressure results from the energy input at the fan. In other words, a degree rise occurs for each *kPa* of fan total pressure. [10]

When the fan and motor are within the air stream, as is the case with many air handling units, all the power absorbed by the driving motor is liberated into the air stream. Full account must then be taken of the motor inefficiency as well as all the fan inefficiency. Assuming a total fan efficiency of 70 % and a motor efficiency of 90 % the temperature rise of the air stream is:

$$\Delta t = \frac{fan\ total\ pressure\ (N/m^2)}{1305 \times 0.7 \times 0.9}$$

$$\Delta t = \frac{fan\ total\ pressure\ (N/m^2)}{822} \tag{3.3.7}$$

This represents a temperature rise of about *1.3 K* for each kPa of fan total pressure. [3][10]

3.3.7 Other heat gains

Heat gains are either sensible, tending to cause a rise in air temperature, or latent, causing an increase in moisture content. In comfort air conditioning sensible gains originate from the following sources:

(1) Solar radiation through windows, walls and roofs.
(2) Transmission through the building envelope and by the natural infiltration of warmer air from outside.
(3) People.
(4) Electric lighting.
(5) Business machines and the like.

Latent heat gains are due to the presence of the occupants and the natural infiltration of more humid air from outside. In the case of industrial air conditioning there may be additional sensible and latent heat gains from the processes carried out. All the above sources of heat gain are well researched but a measure of uncertainty is introduced by the random nature of some, such as the varying presence of people and the way in which electric lights are switched. The thermal inertia of the building structure also introduces a problem when calculating the sensible heat gain arising from solar radiation. It follows that a precise determination of heat gains is impossible. Nevertheless, it is vital that the design engineer should be able to calculate the heat gains with some assurance and this can be done when generally accepted methods of calculation are followed, supported by sound common sense. [10]

3.4 Air Cooler Coils

A cooler coil is not merely a heater battery fed with chilled water or into which cold, liquid refrigerant is pumped. There are two important points of difference: at first, the temperature differences involved are very much less for a cooler coil than for a heater battery, and secondly, moisture is condensed from the air on to the cooler coil surface. With air heaters, water entering and leaving at 85 °C and 65 °C respectively may be used to raise the temperature of an air stream from 0 °C to 35 °C, resulting in a log mean temperature difference of about 53 °C for a counter flow heat exchange. With a cooler coil, water may enter at 7 °C and leave at 13 °C in reducing the temperature of the air stream from 26 °C to 11 °C, a log mean temperature difference of only 7.6 °C with counter flow operation. The result is that much more heat transfer surface is required for cooler coils and it is important that counter flow heat exchange be obtained. Chilled water coils are usually constructed of externally finned, horizontal tubes, so arranged as to facilitate the drainage of condensed moisture from the fins. Tube diameters vary from 8 to 25 mm, and copper is the material commonly used, with copper or aluminium fins. Copper fins and copper tubes generally offer the best resistance to corrosion, particularly if the whole assembly is electro-tinned after manufacture. Fins are usually of the plate type, although spirally wound and circular fins are also used. Cross flow heat exchange between the air and cooling fluid occurs for a particular row but, from row to row, parallel flow or counter flow of heat may take place, depending on the way in which the piping has been arranged. Counter flow connection is essential for chilled water coils in all cases. In direct expansion coils, since the refrigerant is boiling at a constant temperature the surface temperature is more uniform and there is no distinction between the parallel flow and counter flow, the logarithmic mean temperature difference being the same. [10]

All cooler coils should be divided into sections by horizontal, independently drained, condensate collection trays running across their full width and depth. Opinions seem to differ among manufacturers as to the maximum permissible vertical spacing between condensate drip pans. Clearly it depends on the sensible-total heat ratio (the smaller this is the greater the condensation rate), the spacing between the fins (the narrower the spacing the more difficult it is for the condensate to drain freely) and the face velocity (the faster the airflow the more probable the carryover of condensate). Fin spacing in common use are 316 to 476 per metre and the thicknesses used lie between 0.15 and 0.42 mm. Thinner fins, incidentally, tend to grip the tube less tightly at their roots and perhaps give poorer heat transfer. Fins may be corrugated or smooth, the former reducing the risk of carryover while improving the heat transfer by a small increase in the surface area of the fins. An analysis of manufacturers' data suggests that for cooling coils having sensible-total heat ratios of not less than 0.65. [10]

For sensible-total ratios less than 0.65 should not be used. Coils with sensible-total ratios exceeding 0.98 are virtually doing sensible cooling only and the risk of condensate carryover is slight. Water velocities in use are between 0.6 and 2.4 m/s, in which range the coils are self-purging of air. Water pressure drops are usually between 15 and 150 kPa and air pressure drops are dependent on the number of rows and the piping and finning arrangements. A coil that is doing no latent cooling offers about one-third less resistance to airflow. [10]

Careless handling in manufacture, delivery to site and erection often causes damage to the coil faces, forming large areas of turned-back fin edges that disturb the airflow, collect dirt from the air stream and increase the air pressure drop. The fins in such damaged areas must be combed out after installation before the system is set to work.

Other materials are sometimes used for air cooler coils but ordinary steel coils should never be used because of the rapid corrosion likely. Stainless steel is sometimes used but it is expensive and, because its thermal conductivity is less than that of copper, more heat transfer surface is required. [10]

Air cooler coils tend to be wide and short, rather than narrow and tall. This is because it is cheaper to make coils with this shape, there is being fewer return bend connections to make (where tubes emerge from the coil easing). It is also because a short coil drains condensate away more easily: with a tall coil there is the likelihood of condensate building up between the fins at the bottom of the coil, inhibiting airflow and heat transfer and increasing the risk of condensate carry-over into the duct system. [10]

A consequence of the wide shape of cooler coil faces is that airflow over them is likely to be uneven, the air stream tending to flow over the middle of the coil face. [10]

Galvanised steel casings are often used for coils with copper tubes and copper or aluminium fins. This is a poor combination since copper and zinc in conjunction with slightly acidic condensate favour electrolytic corrosion. If possible, other materials should be used for cooler coil casings. Drain cocks and air vents should always be provided for cooler coils using chilled water or brine. [10]

3.4.1 Parallel and counter flow

Although ordinary heat exchangers may be extremely different in design and construction and may be of the single- or two-phase type, their modes of operation and effectiveness are largely determined by the direction of the fluid flow within the exchanger. The most common arrangements for flow paths within a heat exchanger are counter flow and parallel flow. [10]

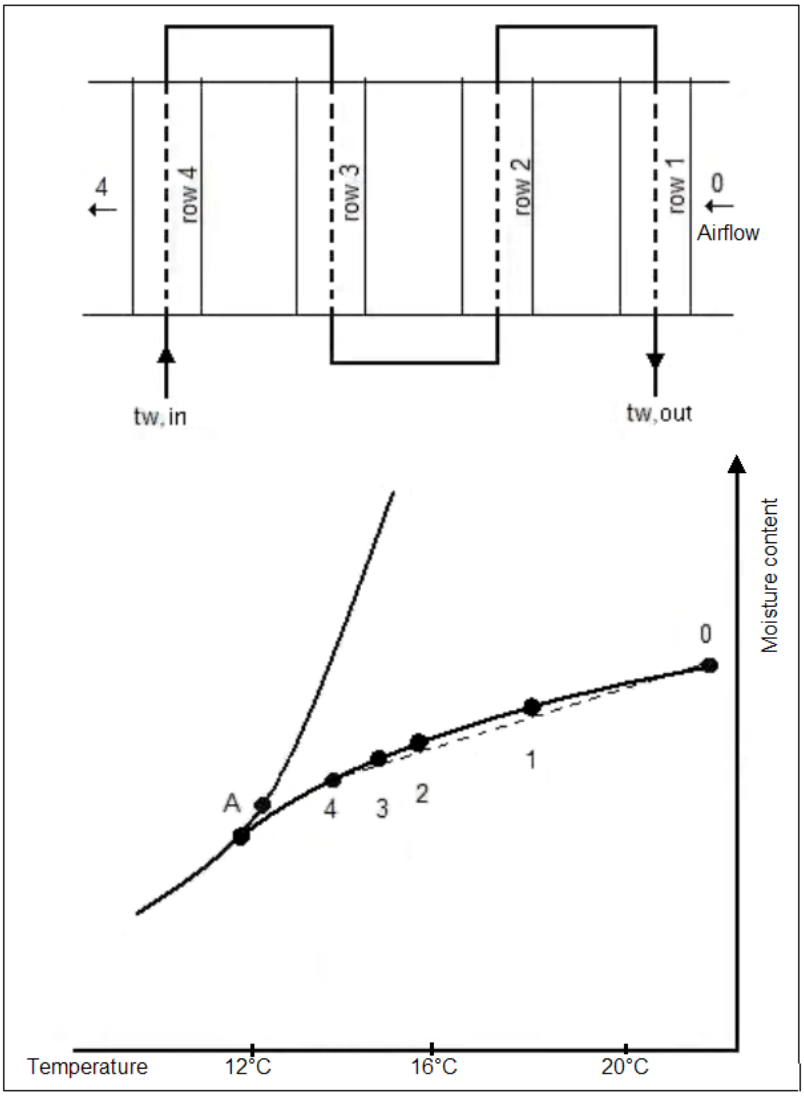


Figure 3.3. *Counter flow. The points 0, 4 and A are in a straight line and A lies on the 100 % saturation curve. A is the apparatus dew point and its temperature,* T_{sm}*, is the mean coil surface temperature for the whole four rows of the coil.*

A counter flow heat exchanger (see Figure 3.3) is one in which the direction of the water flow is opposite to the direction of the airflow, due to this the heat transfer is almost equal in every different rows of coil. A line joining the state points 0, 1, 2, 3 and 4 is a convex curve and represents the change of state of the air as its flows past the rows under counter flow conditions. A straight line joining the points 0 and 4 indicates the actual overall performance of the coil. This condition line replacing the condition curve, cuts the saturation curve at a point A when produced. [10]

Similar considerations apply when parallel flow is dealt with (Figure 3.4), but the result is different. A concave condition curve is obtained by joining the points 0, 1, 2, 3 and 4. Flow direction of water and air are same for both in parallel flow, which means that most of the heat transfer happens in the row number 1 and the least in row number 4. When comparing these two methods, could be said that, a lower leaving air temperature is achieved, greater heat transfer occurs, and the coil is more efficient, if it is piped for counter flow operation. Due of these things, counter flow configuration is commonly used method in cooling coils. [10]

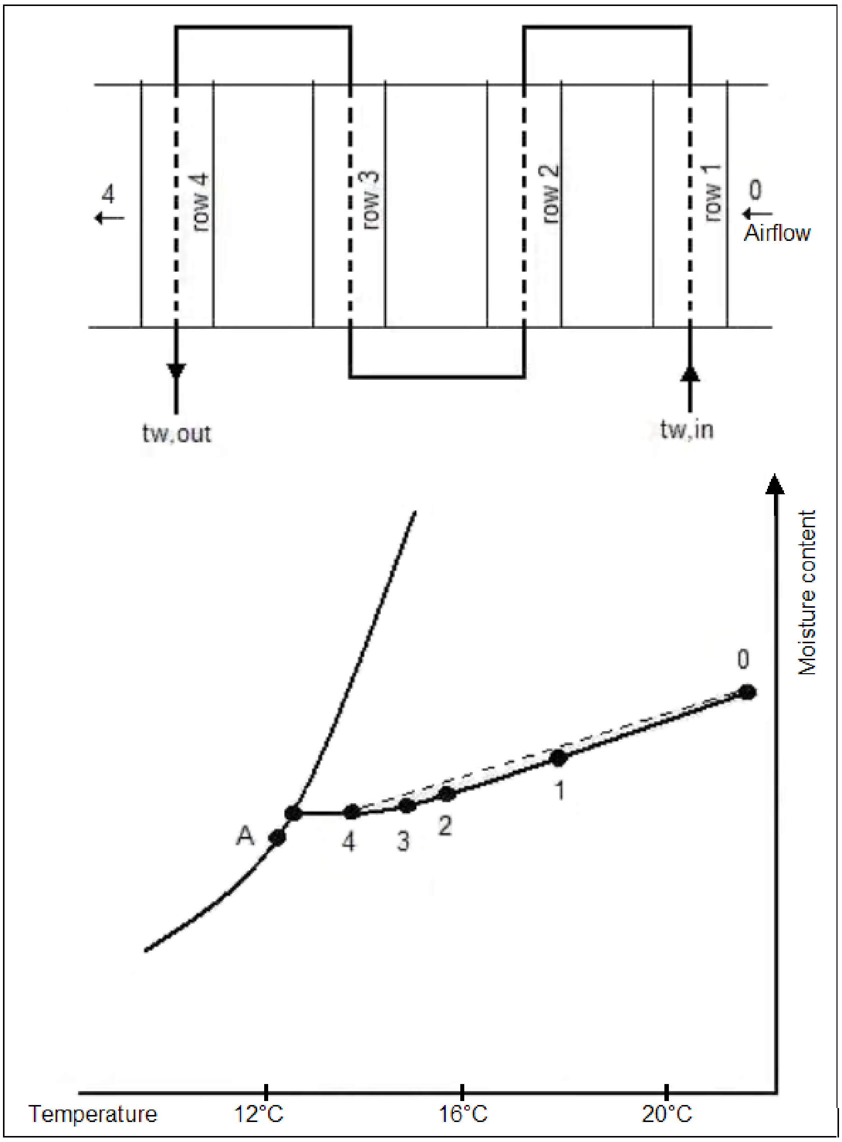


Figure 3.4. *Parallel flow. The points 0, 4 and A are in a straight line. A is the apparatus dew point and its temperature is the mean coil surface temperature, T_{sm}.*

3.4.2 Contact factor

Such a definition is not always useful-for example, in a cooler coil for sensible cooling only-and so it is worth considering another approach, in terms of the heat transfer involved, that is in some respects more informative though not precise.

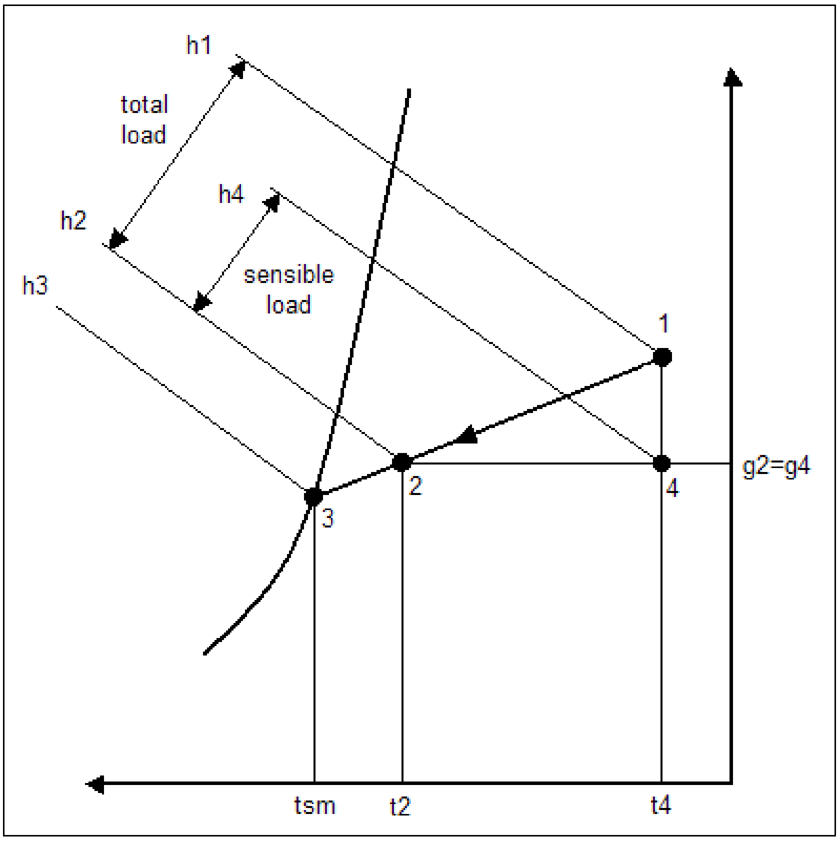


Figure 3.5. *The psychrometric relationship between sensible and total cooling load.*

Coils used for dehumidification as well as for cooling remove latent heat as well as sensible heat from the air stream. This introduces the idea of the ratio S, defined by the expression:

$$S = \frac{\textit{sensible heat removed by the coil}}{\textit{total heat removed by the coil}}$$

In terms of Figure 3.5 this becomes:

$$S = \frac{h_4 - h_2}{h_1 - h_2} \tag{3.4.1}$$

If the total rate of heat removal is Q_t when a mass of dry air m_a in *kg/s* is flowing over the cooler coil, then the sensible heat ratio *S*, can also be written as

$$S = \frac{\dot{m}_a \times C_p \times (t_1 - t_2)}{Q_t} \tag{3.4.2}$$

where C_p is the humid specific heat of the air stream.

Sensible heat transfer, Q_s, can be considered in terms of the outside surface air film resistance of the coil, R_a:

$$Q_s = 1.25 \times \dot{v} \times (t_1 - t_2) = \frac{LMTD_{as} \times A_t}{R_a} \tag{3.4.3}$$

where $\dot{v}$ is volumetric flow rate in m^3/s, A_t is the total external surface area of the coil in m^2 and $LMTD_{as}$ is the logarithmic mean temperature difference between the air stream and the mean coil surface temperature, T_{sm}.

$$LMTD_{as} = \frac{[(t_1 - T_{sm}) - (t_2 - T_{sm})]}{\ln[(t_1 - T_{sm})/(t_2 - T_{sm})]} \tag{3.4.4}$$

Equation (3.4.3) then becomes

$$Q_s = \frac{(t_1 - t_2)}{\ln[(t_1 - T_{sm})/(t_2 - T_{sm})]} \times \frac{A_t}{R_a} \tag{3.4.5}$$

and therefore

$$1.25 \times \dot{v} \times (t_1 - t_2) = \frac{(t_1 - t_2)}{\ln[(t_1 - T_{sm})/(t_2 - T_{sm})]} \times \frac{A_t}{R_a}$$

whence

$$\ln\left[\frac{(t_1 - T_{sm})}{(t_2 - T_{sm})}\right] = \frac{A_t}{R_a \times 1.25 \times \dot{v}} \tag{3.4.6}$$

Since $\dot{v}$ equals $A_f \times v_f$, where A_f and v_f are the face area and face velocity entering the cooler coil in *m/s*, respectively, equation (3.4.6) becomes

$$\ln\left[\frac{(t_1 - T_{sm})}{(t_2 - T_{sm})}\right] = \frac{A_t}{A_f \times R_a \times 1.25 \times v_f} = k \tag{3.4.7}$$

k is a constant for a given coil and face velocity but it takes no account of heat transfer through the inside of the tubes. We can now write:

$$\frac{(t_2 - T_{sm})}{(t_1 - T_{sm})} = \exp(-k)$$

If r is the number of rows and A_r is the total external surface area per row, A_t *is* $A_r \times r$ and k becomes $(A_r/A_f)(r)/(R_a \times 1{,}25 \times v_f)$. An approximate expression for the contact factor now emerges:

$$\beta = 1 - \exp\left(-\frac{A_r}{A_f} \times \frac{r}{1.25 \times R_a \times v_f}\right) \tag{3.4.8}$$

Note that the contact factor is independent of the psychrometric state and the coolant temperature, provided that the mass flow ratio of air to water remains constant. [10]

3.4.3 Heat and mass transfer to cooler coils

LMTD-method

Heat and mass transfer to a cooler coil involves three stages: heat flows from the air stream to the outer surface of the fins and tubes, it is then transferred through the metal of the fins and the wall of the tubing, and finally, it passes from the inner walls of the tubes through the surface film of the cooling fluid to the main stream of the coolant. In general, dehumidification occurs as well as cooling so that the behaviour of cooler coils cannot be described in simple terms. A very approximate approach, adopted by some manufacturers, is to establish a U-value for the coil that is given a bias to account for the extra heat flow by virtue of condensation. The duty is described by:

$$Q_t = U_t \times A_t \times LMTD_{aw} \tag{3.4.9}$$

Where U_t is the biased U-value, A_t the total external surface area and $LMTD_{aw}$ is the logarithmic mean temperature difference between the air stream and the water flowing, defined by:

$$LMTD_{aw} = \frac{\left[(t_l - t_{wb}) - (t_2 - t_{wa})\right]}{\ln\left[(t_l - t_{wb}) / (t_2 - t_{wa})\right]} \tag{3.4.10}$$

using the notation of Figure 3.5. The biased thermal transmission coefficient is expressed by

$$1/U_t = R = SR_a + R_m + R_w \tag{3.4.11}$$

where R_a is the thermal resistance of the air film when the external surface is dry, S is the sensible-total heat transfer ratio, R_m is the thermal resistance of the tubes R_t plus the fins R_f, and R_w is the resistance of the water film within the tubes. R_w must multiplied by the ratio A_t/A_i, where A_t is the total external surface area and A_i is the internal surface area, so that it refers to the total external surface area. R is the total thermal resistance, air to water. [10]

R_a is the reciprocal of the sensible heat transfer coefficient on the air side, h_a, which depends principally on the mass flow rate of the air stream. For standard air, a staggered arrangement of tubes, h_a is: [8]

$$h_a = 27.42 v_f^{\,0.8} \tag{3.4.12}$$

If the coil is only partially wet then there are two U-values: one using SR_a in equation (3.4.11) and referring to the wetted part of the external surface, and the other using R_a in the equation and referring to the dry part of the surface. The resistance of the fins when dry:

$$R_f = \frac{1-\eta}{\eta} \times R_a \tag{3.4.13}$$

If the fins are wet the SR_a replaces R_a. The term η is the total surface effectiveness and is defined by

$$\eta = (\eta_{fin} \times A_{fin} + A_{tube}) / A_t \tag{3.4.14}$$

where A_{fin} is the surface area of the fins, A_{tube} is the external area of the tubes, A_t is the total external surface area and η_{fin} is the fin efficiency. The latter is an involved function depending on the practical features of coil construction. [1][2][10]

For calculating fin efficiency its recommended to use the Schmidt equation to obtain good accuracy (equation 3.4.15). [9]

$$\eta_{fin} = \frac{\tanh(mr\phi)}{mr\phi} \times \cos(mr\phi) \;\; ; \;\; m = \sqrt{\frac{2h}{\lambda_f \, \delta_f}} \tag{3.4.15}$$

where η_{fin} is the fin efficiency, m and ϕ are fin parameters, r is tubes outer diameter, h is heat transfer coefficient on the air side, λ_f is thermal conductivity of fin, and δ_f is fin thickness.

Parameter ϕ is defined by:

$$\phi = \left(\frac{r_f}{r} - 1\right)\left[1 + \left(0.3 + \left(\frac{m(r_f - r)}{2.5}\right)^{\frac{r_f}{8r}} \left(0.26\left(\frac{r_f}{r}\right)^{0.3} - 0.3\right)\right) \ln\left(\frac{r_f}{r}\right)\right] \tag{3.4.16}$$

where r_f is circular fin radius in meters, solved by equations (3.4.17) (for rectangular) or (3.4.18) (for hexagonal) depending by case seen in:

$$\frac{r_f}{r} = 1.28 \frac{X_t}{r} \sqrt{\frac{X_L}{X_T} - 0.2} \tag{3.4.17}$$

$$\frac{r_f}{r} = 1.27 \frac{X_t}{r} \sqrt{\frac{X_D}{X_T} - 0.3} \quad ; \quad 2X_D = \sqrt{P_l^2 + \frac{P_t^2}{4}} = \sqrt{4X_L^{\,2} + X_T^{\,2}} \tag{3.4.18}$$

where X_T, X_D, X_L, P_l and P_t are coil parameters (see Figure 3.6).

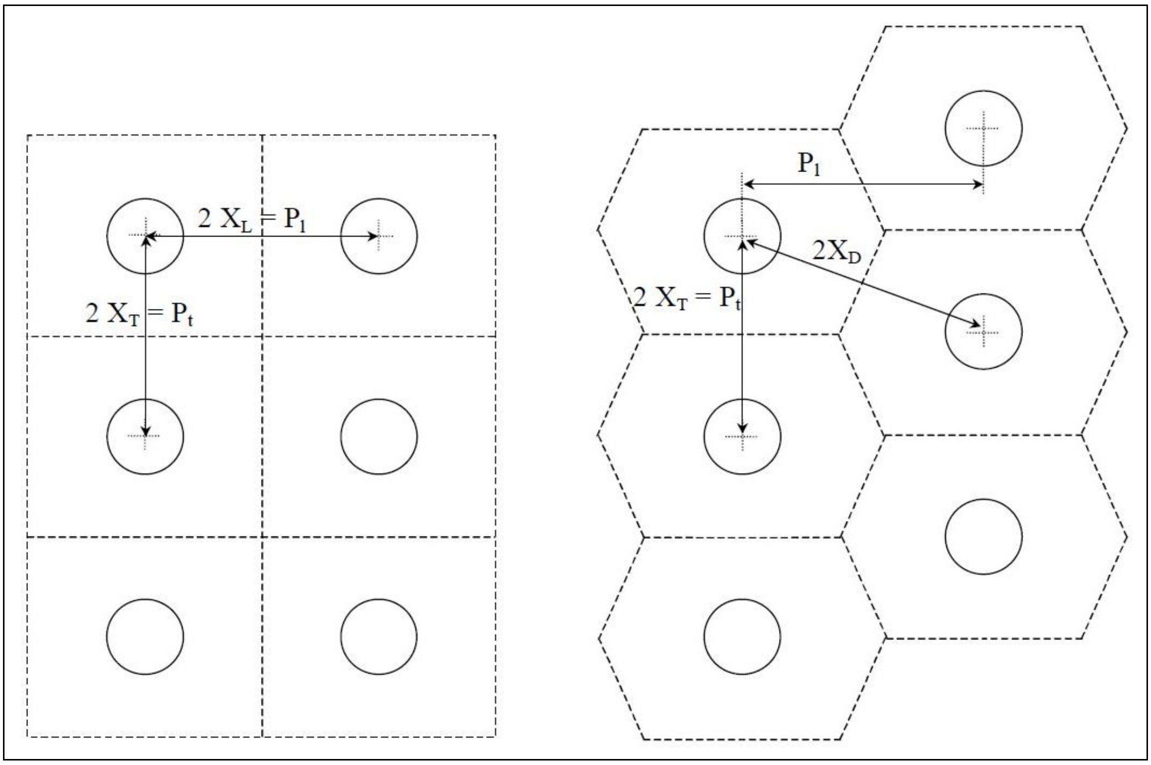


Figure 3.6. *Unit cells for inline and staggered tube layouts with plain plate fins.* [9]

Using copper as a material instead of aluminium improves efficiency but the use of larger diameter tubing reduces it.

The resistance of the metal of the tubes, R_t, referred to the total external surface are, is defined by

$$R_t = \frac{A_t}{A_i} \times \frac{d_o}{2\lambda_t} \times \ln\frac{d_o}{d_i} \tag{3.4.19}$$

wherein A_t/A_i is the ratio of the total external surface to the total internal surface area, λ_t is the thermal conductivity of the metal of the tubes, d_o is the outer tube diameter and d_i its inner diameter, both in meters.

The thermal resistance of the water film inside the tubes, R_w, referred to their inner surface area, is given by the following:

$$R_w = d_i^{\,0.2} / \left[\left(1429 + 20.9 \times t_{wm}\right) \times v^{0.8} \right] \tag{3.4.20}$$

in which t_{wm} is the mean water temperature, v is the mean water velocity and d_i is the inside diameter of the tube in metres. [1][10]

The method of enhancing the U-value of a cooler coil in order to account for the dehumidification occurring, cannot be regarded as anything other than very approximate. Some of the reasons are:

(a) The heat transfer is cross flow for any particular row, even though it is counter flow from row to row. Hence the logarithmic mean temperature difference used in the foregoing should be modified to take account of this.

(b) The airflow distribution over the face of the coil is not uniform, being concentrated in the centre with a tendency to stagnation at the corners. Hence the value of the heat transfer coefficient through the air film, h_a, is not uniform over the coil face.

(c) The use of the sensible-total heat transfer ratio, S, must be regarded as an approximation to account for dehumidification.

(d) It is probable that not all the external surface of the fins and tubes is wet with condensate.

(e) The U-value is not constant throughout the cooler coil.

(f) The psychrometric state of the air leaving the coil is not uniform across the face and hence calculations based on a single state will not be precise.

ε– NTU method

The LMTD -method may be applied to design problems for which the fluid flow rates and inlet temperatures, as well as a desired outlet temperature, are prescribed. For a specified coil type, the required surface area, as well as the other outlet temperature, are readily determined. If the LMTD -method is used in performance calculations for which both outlet temperatures must be determined from knowledge of the inlet temperatures, the solution procedure is iterative. For both, design and performance calculations, ε–NTU (the effectiveness-Number of Transfer Units) -method may be used without iteration. [3][12]

Heat exchanger effectiveness ε, is defined as

$$\varepsilon = \frac{q}{q_{\max}}, \; \left[0 \le \varepsilon \le 1\right] \tag{3.4.21}$$

where q is the total capacity of heat exchanger, and q_{max} is maximum possible heat rate of heat exchanger, defined as

$$q_{\max} = C_{\min} \times \left(T_{hot,in} - T_{cold,in}\right) \tag{3.4.22}$$

where $T_{hot,in}$ is inlet temperature of hot fluid, and $T_{cold,in}$ is inlet temperature of cold fluid. C_{min} is the smaller one of heat capacity rates C_{hot} and C_{cold}.

The effectiveness depends of heat exchanger type, for a counter flow configuration, the effectiveness can be calculated from the following equation,

$$\varepsilon = \frac{\left(1 - \exp^{-NTU(1-C_r)}\right)}{\left(1 - C_r \times \exp^{-NTU(1-C_r)}\right)} \tag{3.4.23}$$

and for parallel flow configuration,

$$\varepsilon = \frac{\left(1 - \exp^{-NTU(1+C_r)}\right)}{\left(1 + C_r\right)} \tag{3.4.24}$$

where NTU, is dimensionless parameter, Number of Transfer Units, and C_r is the critical heat capacity rate of fluids. These both parameters can be calculated from the equations (3.4.25) and (3.4.26).

$$C_r = \frac{C_{\min}}{C_{\max}} \tag{3.4.25}$$

$$NTU = \frac{UA}{C_{\min}} \tag{3.4.26}$$

where U is U-value of coil and A is coil surface area. [3][12]

3.5 Fan performance and airflow in ducts

The most prevalent type of fan used in the HVAC industry, and in Fan Coil Units today is a centrifugal fan. They are usually cheaper than axial fans and simpler in construction. A centrifugal fan is a mechanical device for moving air or other gases with a very low increase in pressure. These fans increase the speed of air stream with the rotating impeller. They use the kinetic energy of the impeller or the rotating blade to increase the pressure of the air stream which in turn moves them against the resistance caused by ducts, dampers and other components. Centrifugal fans accelerate air radially, changing the direction (typically by 90°) of the airflow. They are sturdy, quiet, reliable, and capable of operating over a wide range of conditions. [10]

Centrifugal fan is a constant volume device, meaning that, at a constant fan speed, a centrifugal fan will pump a constant volume of air rather than a constant mass. This means that the air velocity in a system is fixed even though mass flow rate through the fan is not. [14]

When a cylinder of air flows through a duct of circular section its core moves more rapidly than its outer annular shells, these being retarded by the viscous shear stresses set up between them and the rough surface of the duct wall. As flow continues, the energy level of the moving air stream diminishes, the gas expanding as its pressure falls with frictional loss. The energy content of the moving air stream is in the kinetic and potential forms corresponding to the velocity and static pressures. If the section of the duct remains constant then so does the mean velocity and hence, the energy transfer is at the expense of the static pressure of the air. The magnitude of the loss is expressed as a function of the Reynolds number *Re*, which is given by:

$$\mathrm{Re} = V \times d / v \tag{3.5.1}$$

where V is the mean velocity of airflow, d is the duct diameter, and v the kinematic viscosity of the air. Generally, laminar flow can be expected if the Reynolds number is less than about 2300. Fully turbulent flow exists when Reynolds number is more than 10000. Between 2300 and 10000, the flow is in a transition state and predictions are unreliable. [10][12]

When air flows through a system of duct and plant, the prime source of energy to compensate the losses incurred by friction and turbulence is the total pressure of the air stream, defined by a simplified form of Bernoulli's theorem:

$$p_t = p_s + p_{\mathrm{v}} \tag{3.5.2}$$

where p_t is total pressure, p_s is static pressure, and p_v is velocity (or dynamic) pressure. The velocity pressure corresponds to the kinetic energy of the air stream and the static pressure corresponds to its potential energy. [10]

For the air stream to flow through the system of plant and ductwork, in spite of the losses from friction and turbulence, the total energy on its upstream side must exceed the total energy on its downstream side. If a pressure, p, propels a small quantity of air, δq, through a duct in a short time, δt, against a frictional resistance equal and opposite to p, then the rate at which energy must be fed into the system to continue the flow is $p(\delta q/\delta t)$. This is termed the air power, P_{air}, and is delivered to the air stream by the fan impeller.

Air power = force × distance per unit time
= pressure in N / m^2 × area in m^2 × velocity in m / s
= fan total pressure in N / m^2 × volumetric airflow rate in m^3 / s

$$P_{air} = P_{tF} \times \dot{v} \tag{3.5.3}$$

where P_{tF} is fan total pressure in Pa (see equation 3.5.5), and $\dot{v}$ is volumetric airflow rate in m^3/s. [10]

In its passage through a fan the air stream suffers various losses, for example bearing losses. It follows that the power input to the fan shaft must exceed the output from the impeller to the air stream. The ratio of impeller output to shaft input is termed the total fan efficiency, η_t and this provides a definition of fan power:

$$P_{fan} = \frac{P_{air}}{\eta_t} \tag{3.5.4}$$

The air power delivered to the air stream provides for the sum of the following:

The acceleration of outside air from rest to the velocity in the air intake, the frictional resistance of the air inlet louvres, the energy losses incurred by turbulence formed in the vena-contracta at entry, the frictional resistance by each item of plant, frictional resistance in the ducts and duct fittings, the frictional resistance of the index grille, losses incurred by the presence of turbulence anywhere in the system and the kinetic energy loss from the system (represented by the mass of moving air delivered from the index grille). [10]

The energy loss by friction and turbulence would cause a temperature rise in the air stream if it were not exactly offset by the fall in temperature resulting from the adiabatic expansion accompanying the pressure drop. The only temperature rise occurs at the fan, where adiabatic compression takes place. The power absorbed by the electric driving motor must exceed the fan power, during steady-state operation, because of the loss in the drive between the fan and the motor and because the efficiency of the motor is less than 100 %.[10]

The size of the fan depends on the airflow rate and the type of fan depends on the application, but the speed at which the fan must run and the size of the motor needed to drive the fan depend on the total pressure loss in the system of duct and plant. Hence it is necessary to calculate energy losses in the system. The following principles and definitions relate to such calculations:

(a) $P_t = P_s + P_v$ (equation 3.5.2)

This is a simplification of Bernoulli's theorem, stating that, in an air stream, the total energy of the moving air mass is the sum of the potential and kinetic energy. Energy is the product of an applied force and the distance over which it is acting. Hence, since pressure is the intensity of force per unit area, total pressure may be regarded as energy per unit volume of air flowing. (This is seen if the unit for pressure, N/m^2, has its numerator and denominator multiplied by metres, yielding Nm/m^3, which equals J/m^3.) Similarly, static pressure can be considered as potential energy per unit volume and velocity pressure as kinetic energy per unit volume. A conclusion drawn from the above is that energy loss through a system corresponds to fan total pressure.

(b) Energy loss corresponds to a fall of total pressure

It is a corollary of Bernoulli's theorem that a fall in energy should correspond to a fall in total pressure and so, when assessing the energy loss through a system, it is the change in total pressure that must be calculated. On the suction side of a fan the total pressure upstream exceeds that at fan inlet and on the discharge side of the fan the total pressure at fan outlet exceeds that downstream. The fan impeller replaces the energy dissipated, by elevating the total pressure between the fan inlet and the fan outlet.

(c) Velocity pressure is always positive in the direction of airflow

For a given volumetric airflow rate in a duct of constant cross section, the mean velocity of airflow and the corresponding kinetic energy must be constant. If a loss of energy occurs, because of friction or turbulence, the corresponding fall in total pressure can only appear as an equal fall in static pressure. Thus it is the static pressure of an air stream that is the source of energy for compensating losses. If the kinetic energy of the air stream is to be drawn upon then it is first necessary to reduce the velocity by expanding the duct section, in order to convert kinetic energy to potential energy (in the form of static pressure), as described by Bernoulli's theorem.

(d) Fan total pressure, P_{tF}. Defined by

$$P_{tF} = P_{to} - P_{ti} \tag{3.5.5}$$

where P_{to} is the total pressure at fan outlet and P_{ti} the total pressure at fan inlet, both in Pa.

(e) Fan static pressure, P_{sF}. Defined by

$$P_{sF} = P_{so} - P_{ti} \tag{3.5.6}$$

where P_{so} is the static pressure at fan outlet.

(f) Fan static pressure, P_{sF}, by virtue of equation (3.5.2).

Fan static pressure is also defined by

$$P_{sF} = P_{tF} - P_{vo} \tag{3.5.7}$$

where P_{vo} is a velocity pressure at fan outlet based on a notional mean velocity v_{fo} defined by

$$v_{fo} = \frac{\dot{v}}{A_{fo}} \tag{3.5.8}$$

where A_{fo} is the area across the flanges at fan outlet.

The velocity distribution over the outlet area of a fan is very turbulent and not easy to measure with accuracy. Hence a notional mean velocity at fan outlet, defined by equation (3.5.8), is determined and the corresponding velocity pressure is added to fan static pressure to define fan total pressure indirectly, by means of equation (3.5.7). Static pressure may be above or below atmospheric pressure, acting as a bursting or collapsing influence on the system. Hence the zero chosen for the expression of pressure is atmospheric pressure, and static and total pressures are given positive or negative values. The air volume handled determines the size of a fan and the purpose for which it is to be used dictates the type of fan to be chosen. The reason for calculating the fan total pressure is to establish the speed at which the fan should be run, most fans being belt-driven, and the power of the motor that should to be selected to drive it.

When the impeller runs, stresses are developed in the shaft, impeller blades, back plate and other rotating parts. These stresses limit the speed at which the fan can safely operate. The critical speed of the fan shaft depends on the nature of the material, its mass, its diameter and the distance between the bearings. The critical speed must be much greater than the maximum allowable running speed, otherwise the bending and torsional stresses set up in the shaft material will approach dangerous limits. The fan manufacturer sets the maximum allowable safe speed, usually about 2/3 of the critical speed. All fans used should be dynamically balanced by the manufacturer, as well as statically balanced. [10][13]

4 CALCULATION TOOL

This chapter deals with the operation of the calculation tool and has been prepared to serve as a guide for the users. The calculation tool consists of these following tabs: *FCU*, *Fan*, *Coil*, *Prints*, *Main*, and *Charts*. They each have their own rationales, which have been explained in this chapter.

4.1 General information

Some of the general considerations for calculation tool are listed in the following:

- All light-yellow painted cells are convertible by user
- All option buttons, as well as drop-down lists and command buttons, are available for use.
- All red painted cells, which may occur somewhere in the tool, means either, invalid input or any other kind of error situation. Red color in cells is due to the conditional formatting. In all error cases, at least one error message should be shown in the *Fault* –section in *FCU* –tab.

Any of the other cells and contents of the tool, except those mentioned above, are protected to avoid any kind of inadvertent use.

4.1.1 Getting started

First of all, when opening the workbook *FCU calculation tool*, program will ask from user to disable or enable macros, it is recommended to answer affirmatively (enable macros), because some of the parts of the tool need macros for full functionality. After selection, the tool should be open in the main view (*FCU* –tab).

4.1.2 Conditional formatting

Calculation tool contains a lot of the cells that are conditionally shaped. In this tool, conditional formatting has been used to express invalid inputs or any other kind of error situations. Formatting has also been used to change background colour in different cases, for example when cooling mode is changed to the heating mode.

In calculation tool, conditional formatting can be found under the *Format* -command on the menu bar, *Format → Conditional Formatting.* However, this function cannot be used when the active cell(s) is protected. Sheet protection can also be found from menu bar, by choosing *Tools → Protection.*

4.1.3 VBA -macros

This calculation tool also contains several VBA (Visual Basic for Applications) - macros, which have been prepared to speed up the use of the tool. All of the macro - codes are listed in the appendices at the end of the thesis.

In calculation tool, macros are found under the *Tools* -command on the menu bar, *Tools → Macro → Macros*.

4.2 FCU -tab

The first tab (*FCU*) contains main properties of all components of Fan Coil Unit. Layout of FCU –tab can be seen in Figures 4.1 and 4.2. Figure 4.1 has been taken from the left side of page, and Figure 4.2 from right side of page. FCU -tab has been divided into 13 different sections, which are also numbered from 1 to 13 in Figures 4.1 and 4.2. These sections are:

1. Mode selection
2. Supply air state
3. Capacities
4. Air pressures
5. Filter section
6. Defaults
7. Primary air state
8. Circulated air state
9. Listed faults
10. Electric heater section
11. Fan section
12. Water valve
13. Coil Section

4.2.1 Mode selection and supply air state

These sections are numbered 1 and 2 in the Figure 4.1. At first, the user will have to choose a mode of use *(1), cooling* or *heating*. The cooling mode is used in summer time, when cooling is needed against heat gains like solar radiation and heat from the passengers. The heating mode is for winter conditions. Sometimes when the cooling mode is used, both cooling and heating may be needed in order to dehumidify the cabin air. This is possible by heating cooled air with the electric heater.

The circumstances of supply air can be followed in section *Supply air to cabin (2)*. These circumstances are the temperature T_{supply}, the humidity $RH\%_{supply}$ and the flow rate V_{supply}. Supply air is composed of primary and circulated air, which are sometimes called fresh and secondary air.

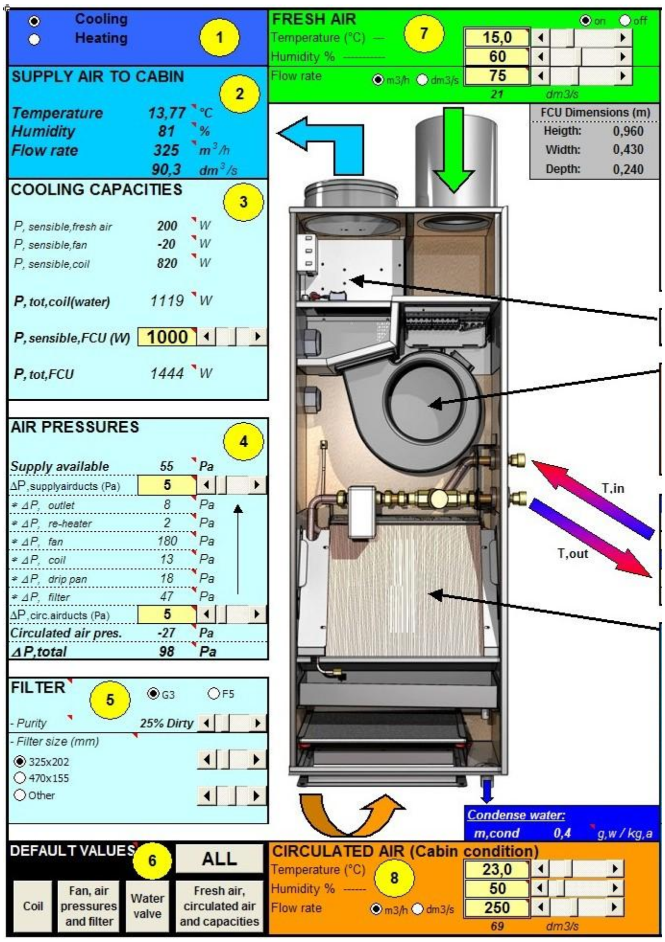


Figure 4.1. *Layout from the left side of FCU-tab.*

4.2.2 Capacities and air pressures

All Fan Coil Unit capacities can be seen in section *Capacities (3)*. The program calculates all the other capacities except total sensible capacity, which has to be adjusted by the user. A tool will give the following information based on total demanded sensible capacity:

- Sensible capacities (primary air, fan, coil)
- Total capacity of refrigerant (water)
- Total capacity of Fan Coil Unit

Air pressures -section *(4)*, contains all details about circulated air pressures in different parts of Fan Coil Unit. Only approximated pressure drops of inlet and outlet ducts in Fan Coil Unit have to be adjusted. The tool calculates all the rest of pressures based on former references. All the calculated pressures are listed in the following:

- *Supply available*

  This pressure equals the pressure between FCU –outlet ducts and cabin free air. Further this means that the pressure should be high enough to distribute treated air uniformly into the cabin.

- Δp_{outlet}

  This parameter describes pressure drop in outlet ducts, located after the electric heater. Circulated air has combined with primary air in this section.

- $\Delta p_{electric\ heater}$

  This is the pressure drop through the electric heater section. Usually this can be noted to be insignificant (less than 5 Pa).

- Δp_{fan}

  This parameter describes a pressure change (rise) in the fan section. The fan must build up the pressure at least an equal amount of the total pressure drop of all components in Fan Coil Unit. Pressure change during fan section is usually 150-250 Pa.

- Δp_{coil}

  Pressure drop through coil section is around 20 Pa in normal use.

- $\Delta p_{drip\ pan}$

  Pressure drop in the drip pan is usually around 20 Pa.

- Δp_{filter}

  Pressure drop in filter -section is highly depended on current filter class, and type (see section 4.2.3). In filter section, the pressure drop with M6 -filter could rise up to 100 Pa.

- *Circulated air pressure*

  This pressure equals the pressure between FCU inlet and cabin free air. This pressure should be less than atmospheric static pressure.

- Δp_{total}

  Total pressure drop through Fan Coil Unit. It usually varies between 150 and 250 Pa.

4.2.3 Filter section and default settings

Filter section *(5)* is for determining filter properties. These properties are *classification, purity and filter size*.

Classification has been divided into three different filters, coarse dust filter G3 and fine dust filters M5 and M6. Loss of pressure is smallest when using coarse dust filter G3. However, it is seldomly used because of increased requirements of cabin air quality. Nowadays, commonly used filters are M5 and even M6. The pressure drops of different filter classes have been studied. It was found that the pressure drops in fine dust filters M5 and M6 are approximately 70 % and 80 % higher than in coarse dust filter G3. Obviously the pressure drop in a filter increases quickly with raised airflow rate, causing higher capacity requirements for fan section.

Selection of *purity* should be done according to the moment when the filter was cleaned or changed. If the filter is brand new, selection should be “Clean”, if the filter has been in use for just a short time, selection could be “25 % dirty”, and so on. This option has been integrated to the tool, because dirtiness affects the pressure drop strongly.

Filter size also affects to the pressure drop, because it will determine flow velocity of air. User could manually set the size of filter (in *millimetres*) by scroll bars.

Default buttons *(6)* are made for avoiding wrong value inputs and other mistakes, and for helping the user generally. There are four different sectional default buttons and one button that resets all inputs.

The first button (Coil) is for resetting all inputs which have been preset to coil section (number 13 in Figure 4.2).

The second button (Fan, Air pressures and Filter) is for resetting all these following sections: Air pressures (number 4 in Figure 4.1), filter (number 5 in Figure 4.1) and fan (number 11 in Figure 4.2).

The third button is for water valve and it will reset all settings in water valve section (number 12 in Figure 4.2).

The fourth button is for primary air, circulated air and demanded sensible capacity and it will reset all settings which have been preset to these mentioned sections (numbers 3, 7 and 8 in Figure 4.1).

The last button collects all of the four buttons mentioned above and resets all inputs. It is recommended to use this button always before using the tool for eliminating all incorrect settings which have possibly been left in the program.

4.2.4 Primary and circulated air states

Primary air -section *(7)* has been made for adjusting the state of primary air. The user can switch primary air on/off by option button located in the right top corner of the box. The colour of the box is white when primary air is turned “off”, and green when turned “on”. Adjustable quantities are temperature in Celsius, *°C*, humidity in per cent, *%*, and flow rate in cubic meters per hour, m^3/h or cubic decimetres per second, dm^3/s. Scroll bar must be used to make these adjustments.

Circulated air control box *(8)* operates exactly as the primary air control box with one exception: circulated air does not have on/off -option button. It’s also good to remember that circulated air state is equivalent with cabin air state.

4.2.5 Listed faults

Listed faults -section *(9)* gives information for the user if settings are indeterminable or if there is any kind of inconsistency between inputs (for example, capacity demand is too high). Program will always print at least one fault message to the screen if any kind of error establishes. Texts of faults are always red bolded as well as all errors in any other part of the tool. All faults and possible reasons of occurring are listed here:

- WATER INLET TEMPERATURE IS TOO HIGH
    - This fault occurs only in cooling mode when inlet water temperature exceeds circulated air temperature after coil, which is determined by demanded capacity of Fan Coil Unit.

- WATER OUTLET TEMPERATURE IS TOO HIGH

- Occurs only in cooling mode when outlet water temperature has been set to be more than circulated air temperature after coil.

- WATER OUTLET TEMPERATURE IS TOO LOW
  - Occurs only in heating mode when outlet water temperature is less than demanded circulated air temperature after coil.

- FLOW RATE OF PRIMARY AIR IS HIGHER THAN RECOMMENDED FOR TWO PERSONS
  - Occurs when primary air flow rate has been set to be more than 150 m^3/h (~42 dm^3/s)

- STATIC PRESSURE OF SUPPLY AIR IS TOO LOW!
  - Occurs when pressure "supply available" does not exceed static cabin air pressure.

- STATIC PRESSURE OF INLET AIR IS TOO MUCH!
  - Occurs when pressure "circulated air pressure" is less than static cabin air pressure.

- AIR FLOW RATE WITH CURRENT AIR PRESSURE DROP IS TOO MUCH FOR CURRENT FAN
  - Occurs when fan performance is not enough for combination of circulated air flow rate and pressure drop in FCU.

- WATER $TEMP_{IN}$ MUST BE LESS THAN WATER $TEMP_{OUT}$ IN COOLING MODE!
  - Occurs in cooling mode when inlet water temperature is higher or equal with outlet water temperature.

- WATER $TEMP_{IN}$ SHOULD BE MORE THAN WATER $TEMP_{OUT}$ IN HEATING MODE!
  - Occurs in heating mode when inlet water temperature is lower or equal with outlet water temperature.

- TABLE OF ACTIVE FAN IN “FAN” -SHEET CONTAINS INVALID INPUTS!
  - Occurs when invalid input(s) have been fed and saved into the fan tables and this same fan has selected to be active fan. In this case, all fan data must to be removed and saved again with correct inputs to remove this incorrect input(s).

- TABLE OF ACTIVE COIL IN “COIL” -SHEET CONTAINS INVALID INPUTS!
  - This fault occurs instantly when invalid input(s) have been fed to the coil tables in "Coil" -sheet. Incorrect cells are shown by red painted colour and these cells are possible to repair one at time.

- CURRENT MAXIMUM SENSIBLE CAPACITY OF FCU HAS EXCEEDED

- Occurs when demanded FCU -capacity exceeds a current available maximum capacity. Available maximum capacity depends highly from circulated air flow rate.

- CIRCULATED AIR TEMPERATURE AFTER COIL PREFER TO BE MORE THAN DEW POINT OF CABIN AIR
  - Occurs when demanded circulated air temperature after coil is less than dew point of cabin. This situation is unfavourable because condensing rate of water starts rising.

- TEMPERATURE DIFFERENCE BETWEEN ROOM AND SUPPLY AIR PREFER TO BE 5-10 DEGREES (°C)
  - This aim is recommendation and occurs only in cooling mode when supply air to cabin is not between 5 and 10 degrees below cabin air.

- FLOW RATE AND PRESSURE DROP OF THE WATER IS TOO HIGH
  - Occurs when water pressure drop exceeds 200 kPa. Pressure drop is highly depended from water flow rate, which in turn is depended of water inlet and outlet temperatures.

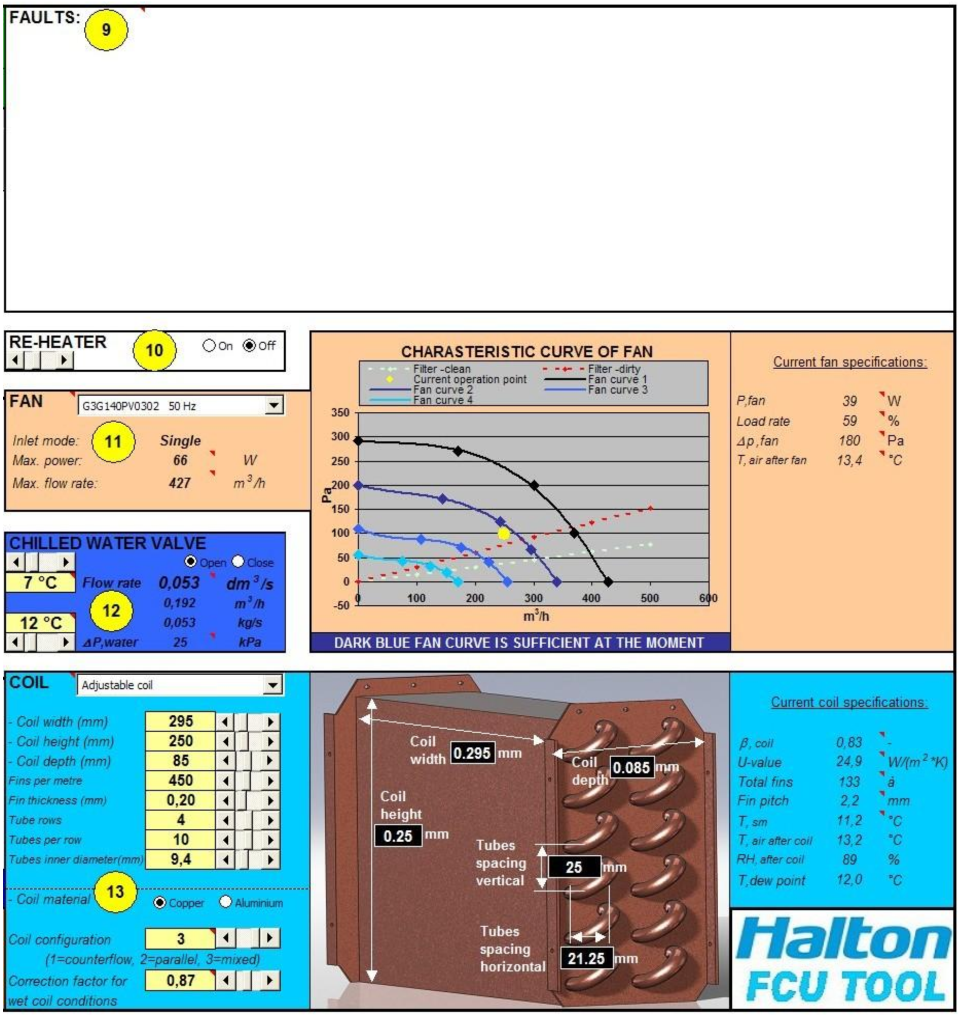


Figure 4.2. *Layout from the right side of FCU-sheet.*

4.2.6 Electric heater and fan -sections

Electric heater -section *(10)* is for using electric heater to warm circulated air. It can be used in heating mode or even cooling mode to dehumidify air when drying cabin air is needed. The scroll bar is not active before user picks option “on” from right top corner of the box. Usually there are 400 W and 800 W electric resistances integrated into Fan Coil Unit, which makes total heating capacity up to 1200 W. The range of heating scroll bar is as well from 0 to 1200 W.

Fan control box *(11)* is for the selection of a suitable fan for the unit. There are preset fans to choose from dropping list. There are nominal values of the active fan below the drop list. Characteristic curve can be seen in the middle and current specifications on the right side of the control box (see Figure 4.2). New fans can be added from Fan –tab if needed (see section 4.3).

4.2.7 Water valve and coil section

Refrigerant details will be determined from water valve -control box *(12)*. An option button can be found to switch valve “On” or “Off” position. Inlet and outlet water temperatures are adjustable by the user. These must be adjusted in a way that ensures the water flow rate and pressure drop of water does not rise too much. The tool calculates the flow rate and pressure drop of refrigerant automatically.

Coil section *(13)* is the widest section of this tool. There is a drop down -list for selecting the coil. All the other coils in the list are preset coils except the first one. While first one (Adjustable coil) is selected, user may adjust all coil dimensions at will and search optimal heat transfer properties. The following dimensions of the coil are adjustable:

- Width, height and depth of coil
- Amount of fins
- Fin thickness
- Amount of tubes per row
- Amount of tube rows
- Tube diameter

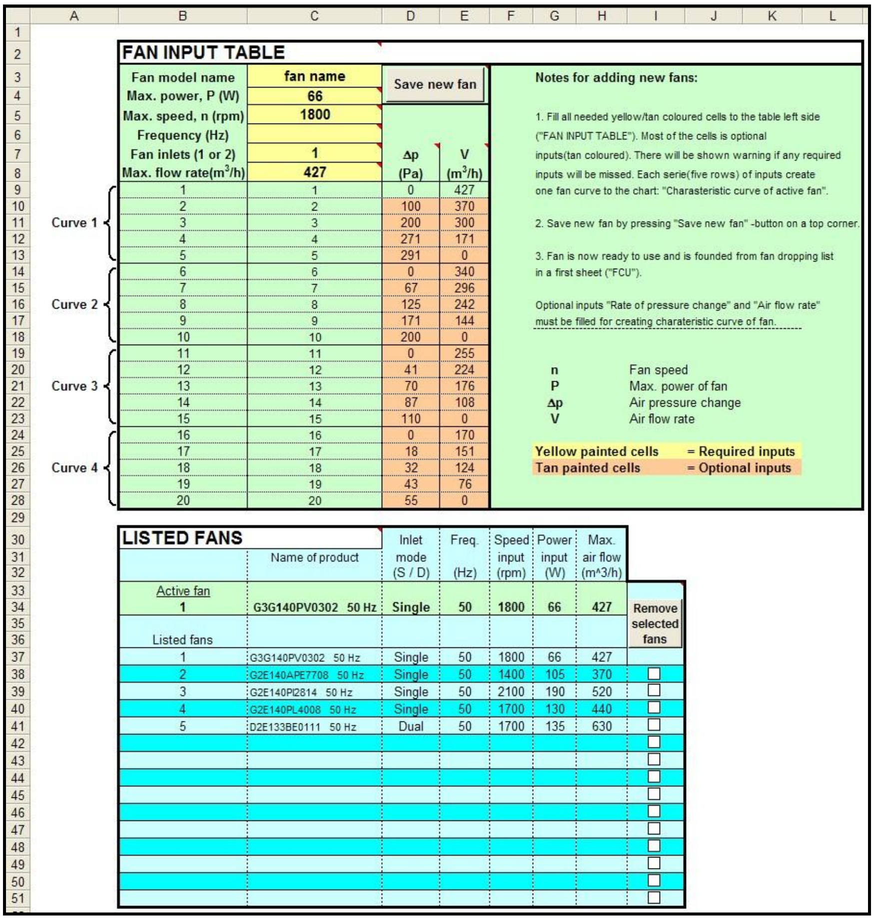
Heat transfer properties of coil are highly depended of width, height and depth of coil. While these dimensions increase, the heat transfer capacity is also increased, because of larger coil surface area (see equation 3.4.10).

The amount of fins also has a large effect on heat transfer capacities. The reason is exactly the same as with increased coil dimensions, more surface area means increased capacity. Increased amount and size of tubes means longer total length of tubes. This makes it possible to remove more heat capacity from water to air with the same flow rate.

These additional features are adjustable: Coil material, coil configuration and correction factor for wet conditions. Two optional coil materials are selectable: copper (commonly used in Fan Coil Unit) and aluminium. Three different coil configurations are selectable: counter flow, parallel flow and a mixture of them. Counter flow is commonly used but the present coil used in Halton is a mixture of counter and parallel flow. Therefore “mixed” configuration is the most suitable one to use in this tool. Correction factor for wet conditions may vary from 0.82 to 0.92. However, the effect to capacity errors is negligible so it is recommended to leave this scroll bar to its preset value 0.87.

4.3 Fan -tab

All fans could be managed from this page. There are three different tables on top of the sheet (see Figures 4.3 and 4.4) All data tables of saved fans can be found below these three fan tables. The first table, “Fan input table” is for adding new fans to the tool, and the second table, “Listed fans“ is for removing old fans from the tool. The third table is named “Active fan table”, where the details of active fan performance can be seen.



FAN INPUT TABLE

Fan model name	fan name	Save new fan
Max. power, P (W)	66	
Max. speed, n (rpm)	1800	
Frequency (Hz)		
Fan inlets (1 or 2)	1	
Max. flow rate(m³/h)	427	

Curve			Δp (Pa)	V (m³/h)
Curve 1	1	1	0	427
	2	2	100	370
	3	3	200	300
	4	4	271	171
	5	5	291	0
Curve 2	6	6	0	340
	7	7	67	296
	8	8	125	242
	9	9	171	144
	10	10	200	0
Curve 3	11	11	0	255
	12	12	41	224
	13	13	87	176
	14	14	87	108
	15	15	110	0
Curve 4	16	16	0	170
	17	17	18	151
	18	18	32	124
	19	19	43	76
	20	20	55	0

Notes for adding new fans:

1. Fill all needed yellow/tan coloured cells to the table left side ("FAN INPUT TABLE"). Most of the cells is optional inputs(tan coloured). There will be shown warning if any required inputs will be missed. Each serie(five rows) of inputs create one fan curve to the chart: "Charasteristic curve of active fan".

2. Save new fan by pressing "Save new fan" -button on a top corner.

3. Fan is now ready to use and is founded from fan dropping list in a first sheet ("FCU").

Optional inputs "Rate of pressure change" and "Air flow rate" must be filled for creating charateristic curve of fan.

n	Fan speed
P	Max. power of fan
Δp	Air pressure change
V	Air flow rate

Yellow painted cells = Required inputs
Tan painted cells = Optional inputs

LISTED FANS

	Name of product	Inlet mode (S / D)	Freq. (Hz)	Speed input (rpm)	Power input (W)	Max. air flow (m^3/h)	Remove selected fans
Active fan 1	G3G140PV0302 50 Hz	Single	50	1800	66	427	
Listed fans							
1	G3G140PV0302 50 Hz	Single	50	1800	66	427	
2	G2E140APE7708 50 Hz	Single	50	1400	105	370	☐
3	G2E140PI2814 50 Hz	Single	50	2100	190	520	☐
4	G2E140PL4008 50 Hz	Single	50	1700	130	440	☐
5	D2E133BE0111 50 Hz	Dual	50	1700	135	630	☐

Figure 4.3. *Tables “Fan input table” and “Listed fans” on the Fan-sheet.*

4.3.1 Adding and removing fans

A new fan can be added by entering all asked mandatory fields (cell range C3:C8 in Figure 4.3), and optional fields (cell range D9:E28 in Figure 4.3) to the table “Fan input table”. Charasteristic curve chart was created by studying manufacturer’s fan performances and entering optional inputs, the air flow rate and corresponding pressures at least from five different operation points, to the table. The program creates the characteristic curves based on third-order functions. Lastly, after all this data has been entered, the fan must be saved by pressing "Save new fan” –button. The program will inform the user that the fan is saved succesfully by a text window. The fan will appear to the table "Listed Fans", located below the fan input table. The program also asks if user wants to clear already stored data from the input table for a new fan. The new fan can be found from the tool and it can be activated from the Fan-section drop-down menu located in the "FCU" –sheet (see Figure 4.2). The program reports errors if the user enters incorrect values into the fan input table.

If the table "Listed Fans” is full, meaning that all 15 different fan tables are in use, it is required to remove some of the older fans from the list before saving new ones. Removal is done by selecting desired fans from check boxes (cell range I38:I51 in Figure 4.3), and removing these fans by a button on top of the boxes.

	P	Q	R	S	T	U	V	W	X	Y	Z	AA
1												
2	ACTIVE FAN TABLE								Equations for charast. curve:			
3									x3	x2	x	b
4									-4,933E-06	0,00069	-0,08147	291
5	Fan model:	G3G140PV0302 50 Hz							-8,403E-06	0,00208	-0,32405	200
6	Inlets:	Single inlet			Max. pressure drop and				-1,841E-05	0,00527	-0,5738	110
7	Frequency:	50	ΔPa	m3/h	sufficiency in current flow rate				-2,782E-05	0,00512	-0,3887	55
8		1	(y)	(x)	(Pa)							
9	101	1	0	427	Max. pressure drop:							
10	102	2	100	370	237							
11	103	3	200	300								
12	104	4	271	171	Current sufficiency:			0	100	200	271	291
13	105	5	291	0	YES			427	370	300	171	0
14	106	6	0	340	Max. pressure drop:							
15	107	7	67	296	117							
16	108	8	125	242								
17	109	9	171	144	Current sufficiency:			0	67	125	171	200
18	110	10	200	0	YES			340	296	242	144	0
19	111	11	0	255	Max. pressure drop:							
20	112	12	41	224	8							
21	113	13	70	176								
22	114	14	87	108	Current sufficiency:			0	41	70	87	110
23	115	15	110	0	NO			255	224	176	108	0
24	116	16	0	170	Max. pressure drop:							
25	117	17	18	151	0							
26	118	18	32	124								
27	119	19	43	76	Current sufficiency:			0	18	32	43	55
28	120	20	55	0	NO			170	151	124	76	0

Figure 4.4. *Active fan table on the Fan-sheet.*

The charasteristic curve chart of the active fan is based on the “Active fan table”. The maximum pressure drop and sufficiency of the fan in current circulated air flow rate are shown, if all rows of optional inputs have been entered (see column “U” in Figure 4.4). The third order functions for creating the characteristic curves can also be seen on this table. It can be concluded that the actual performance curves follow quite closely the calculated curves, which are based on the third order functions.

Note: It is recommended to save the workbook every time before closing it, if new fan(s) has been saved.

4.4 Coil -tab

Most of the calculations, which are associated in any way with the coil section, are located in this page. As told before, table “Coils” (see Figure 4.5) is for managing cooling coils. Adding or removing coils can be done by using this table.

4.4.1 Adding and removing coils

The cooling coil addition is done by entering input data into yellow painted cells in the table (see Figure 4.5). The table contains one row for each cooling coil (Cell range

B20:R30). Twelve coils can be stored at the same time, but the first one of them is protected because it contains an adjustable coil (see section 4.2.7).

Required information for adding a coil are width, height and depth of coil, fins per metre, fin thickness, amount of rows, tubes per row and diameters of the tube. The program calculates automatically all other dimensions for the coil (painted with gray color on the table). The program reports all input errors. If the input feeded by the user is out of range or otherwise invalid, the cell will be painted with red color. Coil data will be saved automatically after the user has entered all input values, and this new coil can be found from the drop-down list on “FCU” –sheet.

COILS

Ekocoil tubes:	9.52 mm tube:	12.7 mm tube:	15 mm tube:
Inner diameter	9,4	12,5	14,8
Outer diameter	10,1	13,3	15,6
Thickness of tube wall	0,35	0,4	0,4

	Coil name	Coil dimensions Width (m)	Height (m)	Depth (m)	Fins Material (Co / Al)	Fins	Fins / m	Heigth (m)	Depth (m)	Thickness (mm)	Tubes Material (Co / Al)	Number of rows	Tubes /row	Length / tube (m)	D,tubes,o (mm)	D,tubes,i (mm)	Tubes spacing horizontal (m)	Tubes spacing vertical (m)
	CURRENT COIL																	
1	Adjustable coil	0,295	0,250	0,085	Co	133	450	0,25	0,085	0,20	Co	4	10	0,295	10,1	9,4	0,0213	0,0250
	LISTED COILS																	
1	Adjustable coil	0,295	0,250	0,085	Co	133	450	0,250	0,085	0,2000	Co	4	10	0,295	10,1	9,4	0,0213	0,0250
2	Coil 2	0,295	0,150	0,085	Co	133	450	0,150	0,085	0,20	Co	4	10	0,295	10,1	9,4	0,0213	0,0150
3	Coil 3	0,295	0,175	0,110	Co	133	450	0,175	0,110	0,22	Co	6	10	0,295	10,1	9,4	0,0183	0,0175
4	Coil 4	0,295	0,200	0,080	Co	133	450	0,200	0,080	0,22	Co	4	10	0,295	10,1	9,4	0,0200	0,0200
5																		
6																		
7																		
8																		
9																		
10																		
11																		
12																		

Cursivated = Protected cells
Normal font = Adjustable cells

Figure 4.5. *Table of the coils on the Coil-sheet.*

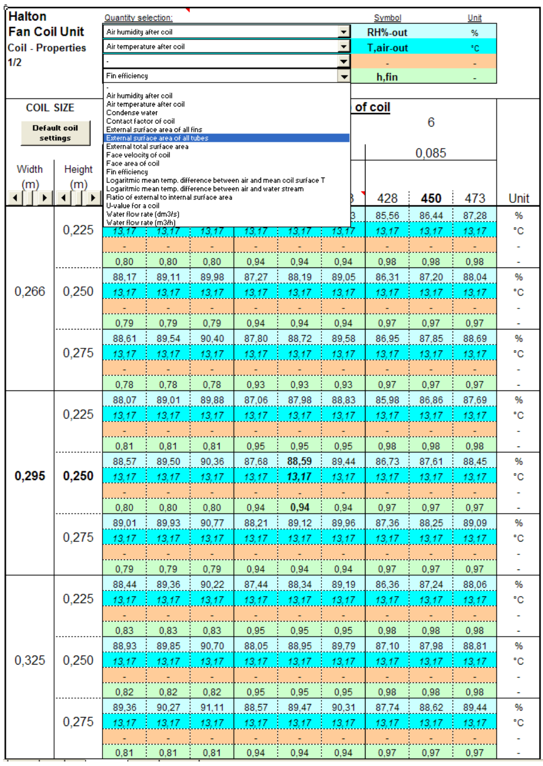
Note: It is recommended to save the workbook every time before closing it, if new coil(s) has been saved.

4.5 Prints -tab

Several prints, which represent most of the cooling coil specifications are located in this tab. There are three different printable pages. The first two pages (shown in Figure 4.6) are valid only when the cooling mode is selected. Values in these prints are automatically painted with red colour if any of errors are found. Program will also inform with red text: "TABLES IS NOT VALID AT THE CURRENT SETTINGS", if the dimensions are adjusted in a way that an error occurs. In this case, the dimensions must be adjusted, until this red text disappears.

The first print shows a table of technical specifications of cooling coil in different range of coil sizes. The second print shows maximum capacities of cooling coil in these same coil sizes, and the third print shows technical information of the whole Fan Coil Unit.

Prints of this tab are easily printable for example customers who may need specific information about coil properties and performances.



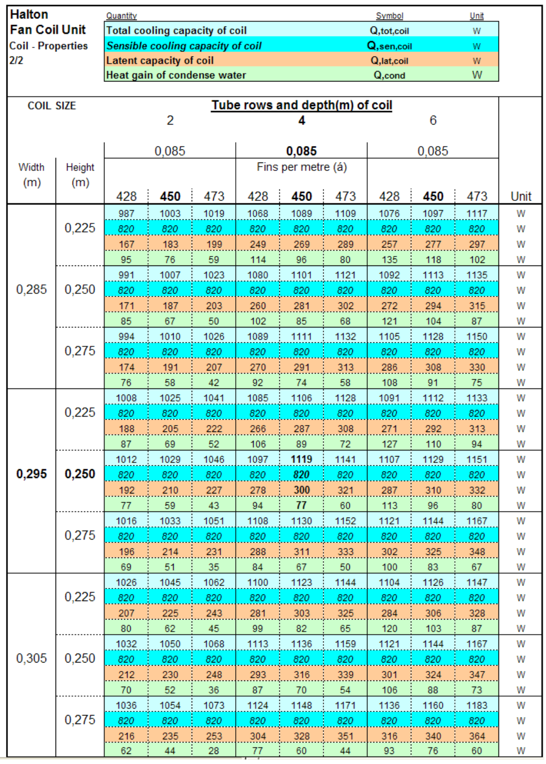


Figure 4.6. *Current coil properties.*

4.5.1 Coil properties

The first print contains all the main parameters of the cooling coil. It is possible to select four different parameters at the same time from drop-down menus. The table shows the changes in the results of selected parameters in different coil size options. Bolded values in the middle of the table represent results of parameters in current settings. Results of these same parameters are also shown in the table with the following changes in columns:

- Current, 5 % reduced and 5 % increased amount of fins.
- All above mentioned amounts of fins with 2, 4, and 6 tube rows.

And in rows:

- Current, 10 % reduced and 10 % increased widths of coil.
- All above mentioned widths with current, 10 % reduced and 10 % increased heights of coil.

The user can easily study how the cooling coil characteristics and capacities behave with all of these dimension changes mentioned above. Dimensions are easily changeable with scroll bars in the first print, and therefore there is no need to browse different tabs for dimension changes. These dimensions will be automatically changed in FCU –tab also. The next different quantities and properties are selectable from the drop-down list:

- Air humidity after coil
- Air temperature after coil

- Condense water
- Contact factor of coil
- External surface area of all fins
- External surface area of all tubes
- External total surface area
- Face velocity of coil
- Face area of coil
- Fin efficiency
- Logarithmic mean temperature difference between air and mean coil surface
- Logaritmic mean temperature difference between air and water stream
- Ratio of external to internal surface area
- U-value of the coil
- Water flow rate m^3/h or dm^3/s

All needed cooling coil capacities are shown in the second print (right side in Figure 4.6). All capacities in the second print will follow dimension changes as well.

4.5.2 Fan Coil Unit specifications

All the most important parameters and specifications of Fan Coil Unit are collected to the last print and are shown on their own separated sections in the table (see appendix B.3).

4.6 Main -tab

Most of the general calculations and functions of the program are located in the table on this tab (table "Fan Coil Unit functions"). Roughly speaking, the structure of the table is divided so that all different water and air states are located to the rows, and the corresponding quantities of them to the columns. The only inputs that can be managed from this sheet are "Total fan efficiency" (located in cell C24), and total outer dimensions of Fan Coil Unit (cells E35:I35). Anyway, it must be observed that changing the total fan efficiency has negligible effect on the Fan Coil Unit properties.

Tables of water and dry air properties are also located on this tab. The source for these tables is the *Basic heat and mass transfer* by Mills. [3]

4.7 Charts -tab

All graphs of the calculation tool are located on this tab, as well as some of the sources for them. In the following are listed charts of this sheet:

- Cooling capacities in different cabin temperatures
- Contact factor in different rows in coil
- Circulated air humidity after coil in different cabin humidities
- Function of water flow speed and pressure drop of water
- Pressure drop charts in different Fan Coil Unit components

5 CONCLUSIONS

A few difficult calculation problems were revealed during this work, which gave their own challenges for planning the calculation tool. The biggest problems of all were in model humidity changes, as well as the amount of condensate water in the coil section. Especially the latter of those was very hard to model accurately, because there were not enough reference cases from earlier tests to support these theoretical calculations. However, this error is not significant when considering the operation of the calculation tool.

There were also a few problems with linking functions and cells to each other. In the middle of the project was needed to clarify the calculation tool because it had become too complex and difficult for further design. The final phase of the project was laborious and the most time consuming part, because the corrections of the tool operation took more time than was originally planned.

There are also unwanted features occurring when references are used in calculations to correct the results. One of those is that when using references for correcting results, these corrections are not working anymore when significant changes are made to Fan Coil Unit, for example when changing a totally different type of coil section to Unit.

Overall, the work was a success, although some of the originally planned features are still missing because of limited time. One of the missing features is the calculation of costs. On the other hand, the calculation tool includes some extra features that were not originally planned for the tool. The possibilities provided by the calculation tool will bring a lot of savings in time and in resources.

The calculation tool has been tested many times by comparing its results with earlier reference results obtained from previous measurements, and it can be concluded that the calculated results have been very close to the reference results.

In future, the calculation tool can be easily extended in many directions. For example, it might be useful property if the tool would consider all outer dimension changes of Fan Coil Unit and give an alert to the user if the outer shell of Unit is too small for fitting the needed Unit components. It would also be useful to integrate the calculation tool with cabin drawings and cabins heat load calculations, and of course, the tool should warn the user as well if Unit is too small for needed cabin size and cabins heat loads.

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APPENDICES

APPENDIX A: EXPLODED VIEW OF FAN COIL UNIT

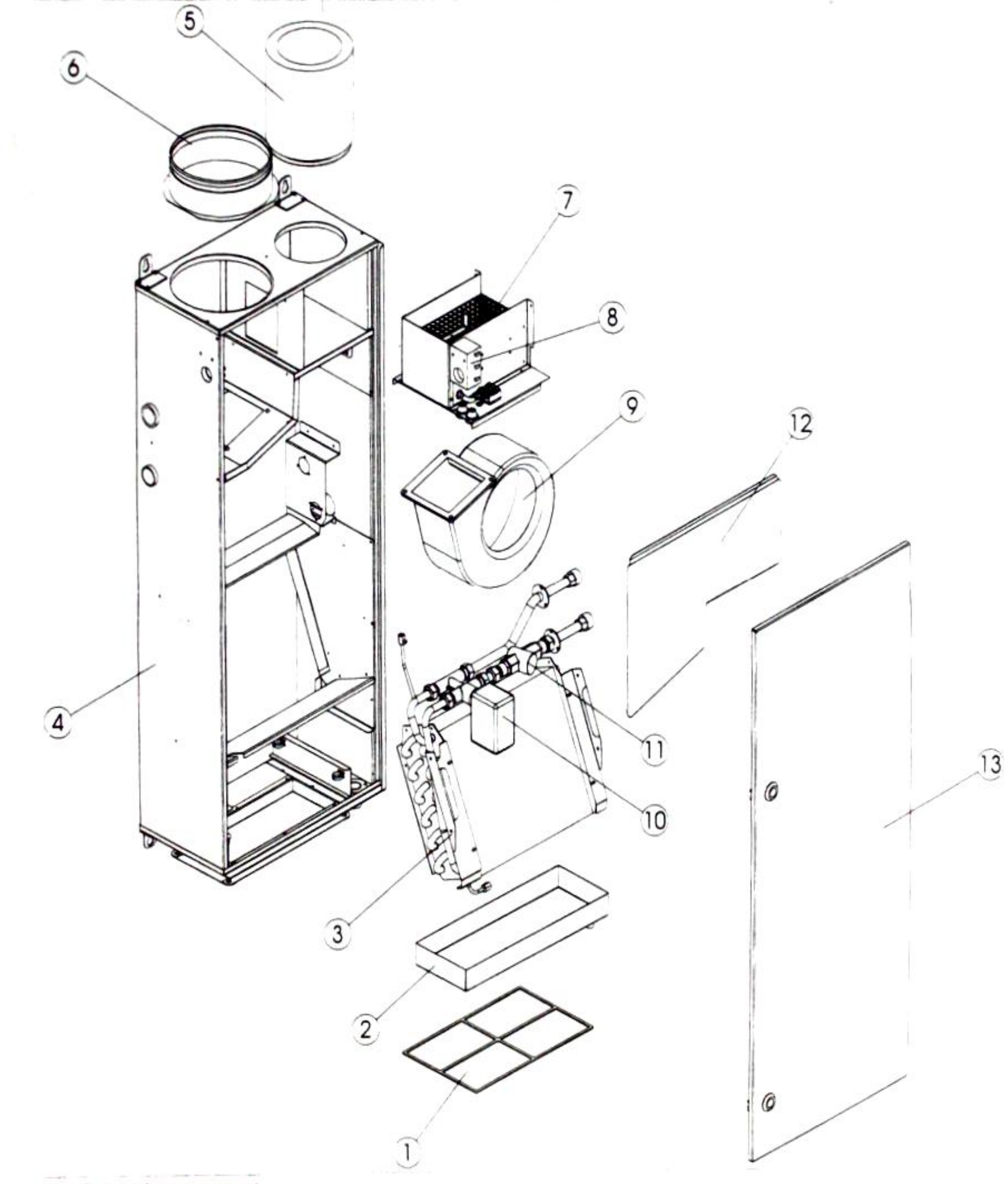
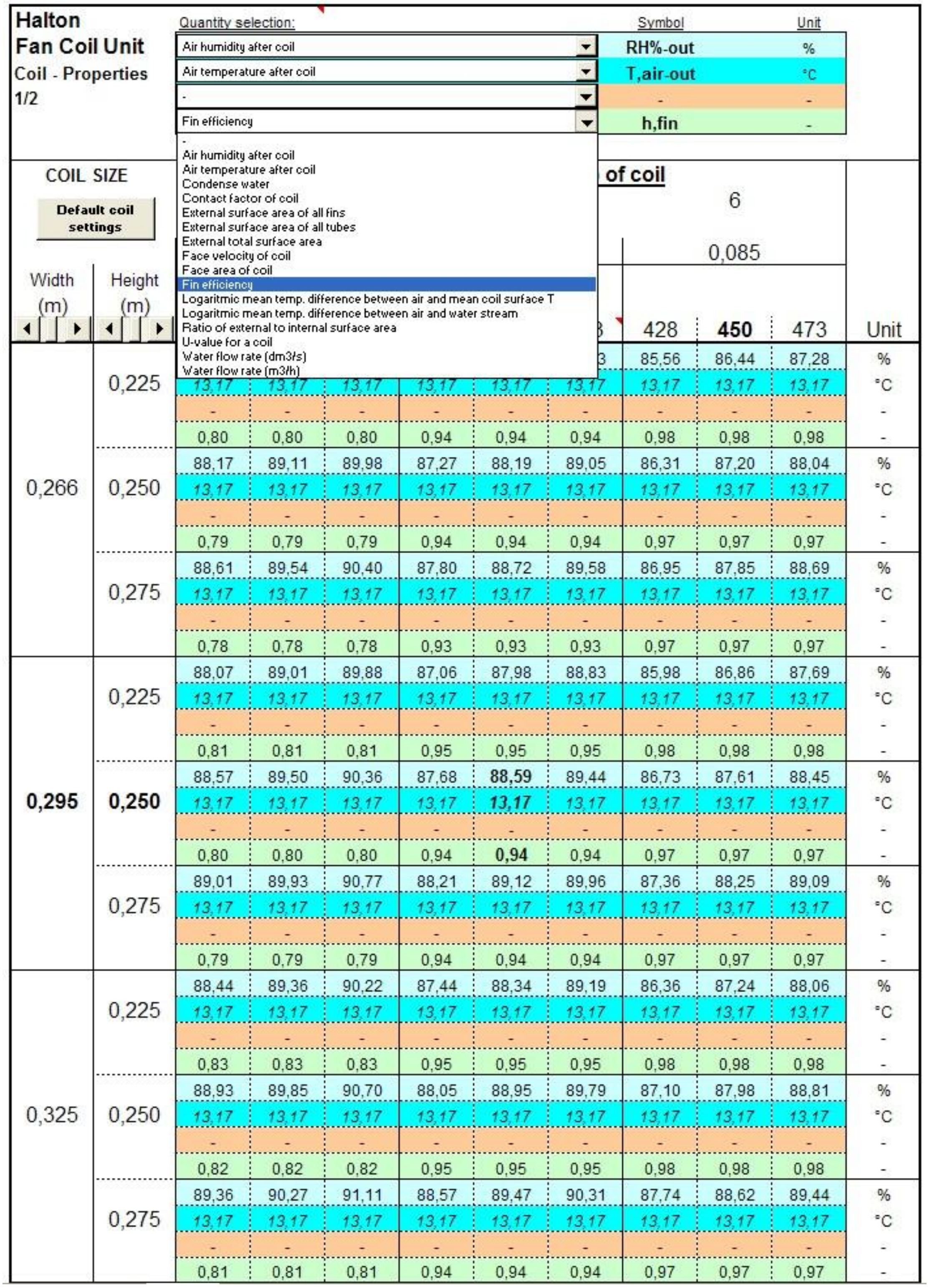


Fig. 2-1: Unit components

Pos. 1: Filter
Pos. 2: Condensate tray
Pos. 3: Heat exchanger
Pos. 4: Basic casing
Pos. 5: Primary air connector
Pos. 6: Supply air connector
Pos. 7: Electric heating
Pos. 8: Safety temperature limiter
Pos. 9: Fan
Pos. 10: Volume control valve
Pos. 11: Control valve
Pos. 12: Cover
Pos. 13: Inspection cover

***Figure A.1.** FCU –components in exploded view.*

APPENDIX B: PRINTS OF THE TOOL



Halton Fan Coil Unit
Coil - Properties
1/2

Quantity selection:

Quantity	Symbol	Unit
Air humidity after coil	RH%-out	%
Air temperature after coil	T,air-out	°C
-	-	-
Fin efficiency	h,fin	-

-
Air humidity after coil
Air temperature after coil
Condense water
Contact factor of coil
External surface area of all fins
External surface area of all tubes
External total surface area
Face velocity of coil
Face area of coil
Fin efficiency
Logaritmic mean temp. difference between air and mean coil surface T
Logaritmic mean temp. difference between air and water stream
Ratio of external to internal surface area
U-value for a coil
Water flow rate (dm3/s)
Water flow rate (m3/h)

COIL SIZE

Default coil settings

of coil

6

0,085

Width (m)	Height (m)						3	428	450	473	Unit
	0,225						3	85,56	86,44	87,28	%
		13,17	13,17	13,17	13,17	13,17	13,17	13,17	13,17	13,17	°C
		-	-	-	-	-	-	-	-	-	-
		0,80	0,80	0,80	0,94	0,94	0,94	0,98	0,98	0,98	-
0,266	0,250	88,17	89,11	89,98	87,27	88,19	89,05	86,31	87,20	88,04	%
		13,17	13,17	13,17	13,17	13,17	13,17	13,17	13,17	13,17	°C
		-	-	-	-	-	-	-	-	-	-
		0,79	0,79	0,79	0,94	0,94	0,94	0,97	0,97	0,97	-
	0,275	88,61	89,54	90,40	87,80	88,72	89,58	86,95	87,85	88,69	%
		13,17	13,17	13,17	13,17	13,17	13,17	13,17	13,17	13,17	°C
		-	-	-	-	-	-	-	-	-	-
		0,78	0,78	0,78	0,93	0,93	0,93	0,97	0,97	0,97	-
	0,225	88,07	89,01	89,88	87,06	87,98	88,83	85,98	86,86	87,69	%
		13,17	13,17	13,17	13,17	13,17	13,17	13,17	13,17	13,17	°C
		-	-	-	-	-	-	-	-	-	-
		0,81	0,81	0,81	0,95	0,95	0,95	0,98	0,98	0,98	-
0,295	**0,250**	88,57	89,50	90,36	87,68	**88,59**	89,44	86,73	87,61	88,45	%
		13,17	13,17	13,17	13,17	**13,17**	13,17	13,17	13,17	13,17	°C
		-	-	-	-	-	-	-	-	-	-
		0,80	0,80	0,80	0,94	**0,94**	0,94	0,97	0,97	0,97	-
	0,275	89,01	89,93	90,77	88,21	89,12	89,96	87,36	88,25	89,09	%
		13,17	13,17	13,17	13,17	13,17	13,17	13,17	13,17	13,17	°C
		-	-	-	-	-	-	-	-	-	-
		0,79	0,79	0,79	0,94	0,94	0,94	0,97	0,97	0,97	-
	0,225	88,44	89,36	90,22	87,44	88,34	89,19	86,36	87,24	88,06	%
		13,17	13,17	13,17	13,17	13,17	13,17	13,17	13,17	13,17	°C
		-	-	-	-	-	-	-	-	-	-
		0,83	0,83	0,83	0,95	0,95	0,95	0,98	0,98	0,98	-
0,325	0,250	88,93	89,85	90,70	88,05	88,95	89,79	87,10	87,98	88,81	%
		13,17	13,17	13,17	13,17	13,17	13,17	13,17	13,17	13,17	°C
		-	-	-	-	-	-	-	-	-	-
		0,82	0,82	0,82	0,95	0,95	0,95	0,98	0,98	0,98	-
	0,275	89,36	90,27	91,11	88,57	89,47	90,31	87,74	88,62	89,44	%
		13,17	13,17	13,17	13,17	13,17	13,17	13,17	13,17	13,17	°C
		-	-	-	-	-	-	-	-	-	-
		0,81	0,81	0,81	0,94	0,94	0,94	0,97	0,97	0,97	-

Figure B.1. *Print, coil properties 1/2.*

Halton Fan Coil Unit Coil - Properties 2/2	Quantity	Symbol	Unit
	Total cooling capacity of coil	Q,tot,coil	W
	Sensible cooling capacity of coil	Q,sen,coil	W
	Latent capacity of coil	Q,lat,coil	W
	Heat gain of condense water	Q,cond	W

COIL SIZE		Tube rows and depth(m) of coil									
		2			**4**			6			
		0,085			**0,085**			0,085			
					Fins per metre (á)						
Width (m)	Height (m)	428	**450**	473	428	**450**	473	428	**450**	473	Unit
0,285	0,225	987	1003	1019	1068	1089	1109	1076	1097	1117	W
		820	*820*	*820*	*820*	*820*	*820*	*820*	*820*	*820*	W
		167	183	199	249	269	289	257	277	297	W
		95	76	59	114	96	80	135	118	102	W
	0,250	991	1007	1023	1080	1101	1121	1092	1113	1135	W
		820	*820*	*820*	*820*	*820*	*820*	*820*	*820*	*820*	W
		171	187	203	260	281	302	272	294	315	W
		85	67	50	102	85	68	121	104	87	W
	0,275	994	1010	1026	1089	1111	1132	1105	1128	1150	W
		820	*820*	*820*	*820*	*820*	*820*	*820*	*820*	*820*	W
		174	191	207	270	291	313	286	308	330	W
		76	58	42	92	74	58	108	91	75	W
0,295	0,225	1008	1025	1041	1085	1106	1128	1091	1112	1133	W
		820	*820*	*820*	*820*	*820*	*820*	*820*	*820*	*820*	W
		188	205	222	266	287	308	271	292	313	W
		87	69	52	106	89	72	127	110	94	W
	0,250	1012	1029	1046	1097	**1119**	1141	1107	1129	1151	W
		820	*820*	*820*	*820*	***820***	*820*	*820*	*820*	*820*	W
		192	210	227	278	**300**	321	287	310	332	W
		77	59	43	94	***77***	60	113	96	80	W
	0,275	1016	1033	1051	1108	1130	1152	1121	1144	1167	W
		820	*820*	*820*	*820*	*820*	*820*	*820*	*820*	*820*	W
		196	214	231	288	311	333	302	325	348	W
		69	51	35	84	67	50	100	83	67	W
0,305	0,225	1026	1045	1062	1100	1123	1144	1104	1126	1147	W
		820	*820*	*820*	*820*	*820*	*820*	*820*	*820*	*820*	W
		207	225	243	281	303	325	284	306	328	W
		80	62	45	99	82	65	120	103	87	W
	0,250	1032	1050	1068	1113	1136	1159	1121	1144	1167	W
		820	*820*	*820*	*820*	*820*	*820*	*820*	*820*	*820*	W
		212	230	248	293	316	339	301	324	347	W
		70	52	36	87	70	54	106	88	73	W
	0,275	1036	1054	1073	1124	1148	1171	1136	1160	1183	W
		820	*820*	*820*	*820*	*820*	*820*	*820*	*820*	*820*	W
		216	235	253	304	328	351	316	340	364	W
		62	44	28	77	60	44	93	76	60	W

Figure B.2. *Print, coil properties 2/2.*

CURRENT FCU -SPECIFICATIONS

FCU CAPACITIES AND HEAT GAINS			
- Sensible cooling capacity of coil	P,sen,coil	820	W
- Sensible cooling capacity from fresh air	P,sen,f-a	200	W
- Heating of fan	P,fan	20	W
- Heating from electric resistor	P,res	0	W
- Latent gains of condense water	P,lat-g	77	W
- TOTAL SENSIBLE CAPACITY	***P,sen***	***1000***	***W***
- TOTAL LATENT CAPACITY	***P,lat***	***444***	***W***
- TOTAL CAPACITY	***P,tot***	***1444***	***W***
- TOTAL CAPACITY OF WATER	***P,water***	***1119***	***W***

COIL PROPERTIES			
- Contact factor	β	0,83	-
- Mean surface temperature of coil	Tsm	11,18	°C
- U-value of coil	U,coil	24,91	W/(m2*K)
- Piping rows of coil	Rows	4	à
- Fins in coil	Fins	133	à
- Condense water	m,cond	0,38	g,w/kg,a
- Coil dimensions	Width	0,295	m
	Height	0,250	m
	Depth	0,085	m

FAN PROPERTIES			
- Power of fan	P,fan	39	W
- Current load rate	Load rate	59	%
- Pressure change in fan	Dp,fan	180	Pa
- Total fan efficiency	γ,fan	0,55	-

AIR AND WATER PROPERTIES		Flow rate		Temp.	Humidity
		m³/h	dm³/s	°C	%
- Inlet circulated airflow	V,circ. air	250	69,4	23,0	50
- Inlet fresh airflow	V,f-a	75	20,8	15,0	60
- Outlet total airflow	V,total	325	90,3	13,8	81
- Inlet water	V,water-in	0,192	0,053	7,0	-
- Outlet water	V,water-out	0,192	0,053	12,0	-

PRESSURES OF FCU			
- Supply air pressure above static pressure	p,supply air	55	Pa
- Inlet air pressure below static pressure	p,circ. air	-26,5	Pa
- Total pressure drop in Fan Coil Unit	Dp,FCU	98	Pa

Figure B.3. *Print, current FCU –specifications.*

APPENDIX C: VBA-MACRO CODES

```
' Fan_add Macro
' Macro recorded 8.11.2011 by Hakala

Sub Fan_add()
Name = Range("C3")
Power = Range("C4")
Speed = Range("C5")
Freq = Range("C6")
Inlet = Range("C7")
Flow = Range("C8")
   If IsEmpty(Name) Then
      MsgBox "Fan model name required"
      Range("C3").Select
      Exit Sub
   ElseIf IsEmpty(Power) Then
     MsgBox "Fan maximum power is
     required"
      Range("C4").Select
      Exit Sub
   ElseIf Power < 1 Or Power > 300 Then
      MsgBox "Invalid maximum fan power"
      Range("C4").Select
      Exit Sub
   ElseIf IsEmpty(Speed) Then
     MsgBox "Maximum fan speed is
     required"
      Range("C5").Select
      Exit Sub
   ElseIf Speed < 0 Or Speed > 5000 Then
      MsgBox "Invalid maximum fan speed"
      Range("C5").Select
      Exit Sub
   ElseIf Freq <> 50 And Freq <> 60 Then
     MsgBox "Invalid Frequency, valid
     frequencys: 50 or 60"
      Range("C6").Select
      Exit Sub
   ElseIf Inlet <> 1 And Inlet <> 2 Then
     MsgBox "Invalid amount of inlets, valid
     inlets: 1 or 2"
      Range("C7").Select
      Exit Sub
   ElseIf Flow <= 0 Or Flow > 999 Then
     MsgBox "Invalid maximum flow rate, set
     value between 1 and 999"
      Range("C8").Select
      Exit Sub
   Else
      Dim Answer As Integer
     Answer = MsgBox(prompt:="Are you
     sure you want to save a new fan?", _
      Buttons:=vbYesNo + vbQuestion,
Title:="Fan saving")
      If Answer = vbYes Then
         Range("D9:E28").Select
         Selection.Copy
ActiveSheet.Range("C63").End(xlDown).Offs
et(6, 1).Select
         Selection.PasteSpecial
Paste:=xlValues, Operation:=xlNone,
SkipBlanks:= _
         False, Transpose:=False
         Range("C3:C7").Select
         Selection.Copy
ActiveSheet.Range("C63").End(xlDown).Offs
et(1, 0).Select
         Selection.PasteSpecial
Paste:=xlValues, Operation:=xlNone,
SkipBlanks:= _
         False, Transpose:=False
         Range("A1").Select
         Selection.ClearContents
         Dim Answer2 As Integer
           Answer2 = MsgBox(prompt:="Fan is
           now added and will be found from fan
           dropping list.
           Do you want also clear input data
           table for a next fan? ", _
         Buttons:=vbYesNo + vbQuestion,
Title:="Note")
         If Answer2 = vbYes Then
            Range("C3:C8").Select
            Selection.ClearContents
            Range("D10:E28").Select
            Selection.ClearContents
            Range("C3").Select
            ActiveCell.FormulaR1C1 = "Fan
            name"
            Range("A1").Select
            Selection.ClearContents
         Else
         Range("A1").Select
         Selection.ClearContents
         End If
      Else
      End If
   End If
End Sub
```

```
' Fan_remove Macro
' Macro recorded 10.11.2011 by Hakala

Sub Fan_remove()
If Range("I38:I51").Text = False Then
   MsgBox "You must select at least one fan"
   Exit Sub
   End If
Dim Answer As Integer
Answer = MsgBox(prompt:="Are you sure
you want remove selected fans?", _
Buttons:=vbYesNo + vbQuestion,
Title:="Note")
If Answer = vbYes Then
fan2 = Range("I38")
fan3 = Range("I39")
fan4 = Range("I40")
fan5 = Range("I41")
fan6 = Range("I42")
fan7 = Range("I43")
fan8 = Range("I44")
fan9 = Range("I45")
fan10 = Range("I46")
fan11 = Range("I47")
fan12 = Range("I48")
fan13 = Range("I49")
fan14 = Range("I50")
fan15 = Range("I51")
If fan2 = True Then
      Range("C88:C92").Select
      Selection.ClearContents
      Range("D93:E112").Select
      Selection.ClearContents
      End If
If fan3 = True Then
      Range("C113:C117").Select
      Selection.ClearContents
      Range("D118:E137").Select
      Selection.ClearContents
      End If
If fan4 = True Then
      Range("C138:C142").Select
      Selection.ClearContents
      Range("D143:E162").Select
      Selection.ClearContents
      End If
If fan5 = True Then
      Range("C163:C167").Select
      Selection.ClearContents
      Range("D168:E187").Select
      Selection.ClearContents
      End If
If fan6 = True Then
      Range("C188:C192").Select
      Selection.ClearContents
      Range("D193:E212").Select
      Selection.ClearContents
      End If
If fan7 = True Then
      Range("C213:C217").Select
      Selection.ClearContents
      Range("D218:E237").Select
      Selection.ClearContents
      End If
If fan8 = True Then
      Range("C238:C242").Select
      Selection.ClearContents
      Range("D243:E262").Select
      Selection.ClearContents
      End If
If fan9 = True Then
      Range("C263:C267").Select
      Selection.ClearContents
      Range("D268:E287").Select
      Selection.ClearContents
      End If
If fan10 = True Then
      Range("C288:C292").Select
      Selection.ClearContents
      Range("D293:E312").Select
      Selection.ClearContents
      End If
If fan11 = True Then
      Range("C313:C317").Select
      Selection.ClearContents
      Range("D318:E337").Select
      Selection.ClearContents
      End If
If fan12 = True Then
      Range("C338:C342").Select
      Selection.ClearContents
      Range("D343:E362").Select
      Selection.ClearContents
      End If
If fan13 = True Then
      Range("C363:C367").Select
      Selection.ClearContents
      Range("D368:E387").Select
      Selection.ClearContents
      End If
If fan14 = True Then
      Range("C388:C392").Select
      Selection.ClearContents
      Range("D393:E412").Select
      Selection.ClearContents
      End If
If fan15 = True Then
      Range("C413:C417").Select
      Selection.ClearContents
      Range("D418:E437").Select
      Selection.ClearContents
      End If
Range("I38:I51").Value = "FALSE"
Range("A1").Select
MsgBox "Selected fans are succesfully
removed"
End If
End Sub
```

```
' CroppingSheets Macro
' Macro recorded 10.11.2011 by Hakala

Sub Auto_open()
    Application.ScreenUpdating = False
    Sheets("FCU").Activate
    Sheets("FCU").ScrollArea = "A1:AB100"
    Sheets("Fan").Activate
    Sheets("Fan").ScrollArea = "A1:AB440"
    Sheets("Coil").Activate
    Sheets("Coil").ScrollArea = "A1:AA240"
    Sheets("Prints").Activate
    Sheets("Prints").ScrollArea = "A1:AP360"
    Sheets("Main").Activate
    Sheets("Main").ScrollArea = "A1:Z160"
    Sheets("Charts").Activate
    Sheets("Charts").ScrollArea = "A1:AC170"
    Sheets("FCU").Activate
End Sub
```

```
' Prim_air_Cir_air_capacities Macro
' Macro recorded 9.10.2011 by Toni Hakala

Sub Prim_air_Cir_air_capacities()
Dim Answer As Integer
Answer = MsgBox(prompt:="Are you sure
you want to reset air and capacity settings?", _
Buttons:=vbYesNo + vbQuestion, Title:="Air
flow rate and capacity settings")
If Answer = vbYes Then
    Application.ScreenUpdating = False
    Sheets("Main").Select
    Range("I10").Select
    ActiveCell.FormulaR1C1 = "60"
    Range("I12").Select
    ActiveCell.FormulaR1C1 = "50"
    Range("K10").Select
    ActiveCell.FormulaR1C1 = "150"
    Range("K12").Select
    ActiveCell.FormulaR1C1 = "230"
    Range("M10").Select
    ActiveCell.FormulaR1C1 = "75"
    Range("M12").Select
    ActiveCell.FormulaR1C1 = "250"
    Range("E21").Select
    ActiveCell.FormulaR1C1 = "1000"
    Range("F68").Select
    ActiveCell.FormulaR1C1 = "1"
    Range("F73").Select
    ActiveCell.FormulaR1C1 = "1"
    Range("F75").Select
    ActiveCell.FormulaR1C1 = "1"
    Range("F95").Select
    ActiveCell.FormulaR1C1 = "1"
    Range("F93").Select
    ActiveCell.FormulaR1C1 = "2"
    Sheets("FCU").Select
Else
End If
End Sub
```

```
' Fan_air_pressures_filter Macro
' Macro recorded 9.10.2011 by Toni Hakala

Sub Fan_air_pressures_filter()
Dim Answer As Integer
Answer = MsgBox(prompt:="Are you sure
you want to reset fan, air pressure and filter
settings?", _
Buttons:=vbYesNo + vbQuestion,
Title:="Settings")
If Answer = vbYes Then
    Application.ScreenUpdating = False
    Sheets("Main").Select
    Range("T19").Select
    ActiveCell.FormulaR1C1 = "5"
    Range("T30").Select
    ActiveCell.FormulaR1C1 = "5"
    Range("F67").Select
    ActiveCell.FormulaR1C1 = "2"
    Range("F84").Select
    ActiveCell.FormulaR1C1 = "1"
    Range("F85").Select
    ActiveCell.FormulaR1C1 = "1"
    Range("U38").Select
    ActiveCell.FormulaR1C1 = "300"
    Range("V38").Select
    ActiveCell.FormulaR1C1 = "200"
    Sheets("FCU").Select
Else
End If
End Sub
```

```
' Water_valve Macro
' Macro recorded 9.10.2011 by Toni Hakala

Sub Water_valve()
Dim Answer As Integer
Answer = MsgBox(prompt:="Are you sure
you want to reset water valve settings?", _
Buttons:=vbYesNo + vbQuestion,
Title:="Water valve settings")
If Answer = vbYes Then
    Application.ScreenUpdating = False
    Sheets("Main").Select
    Range("K7").Select
    ActiveCell.FormulaR1C1 = "70"
    Range("K8").Select
    ActiveCell.FormulaR1C1 = "120"
    Range("F89").Select
    ActiveCell.FormulaR1C1 = "1"
    Sheets("FCU").Select
Else
End If
End Sub
```

```
' Coil Macro
' Macro recorded 9.10.2011 by Toni Hakala

Sub Coil()
Dim Answer As Integer
Answer = MsgBox(prompt:="Are you sure
you want to reset coil settings?", _
Buttons:=vbYesNo + vbQuestion, Title:="Coil
settings")
If Answer = vbYes Then
Application.ScreenUpdating = False
    Sheets("Main").Select
    Range("E34").Select
    ActiveCell.FormulaR1C1 = "85"
    Range("G34").Select
    ActiveCell.FormulaR1C1 = "250"
    Range("I34").Select
    ActiveCell.FormulaR1C1 = "295"
    Range("L34").Select
    ActiveCell.FormulaR1C1 = "4"
    Range("N34").Select
    ActiveCell.FormulaR1C1 = "10"
    Range("P34").Select
    ActiveCell.FormulaR1C1 = "450"
    Range("S34").Select
    ActiveCell.FormulaR1C1 = "20"
    Sheets("Coil").Select
    Range("A13").Select
    ActiveCell.FormulaR1C1 = "1"
    Sheets("FCU").Select
Else
End If
End Sub
```

```
' All Macro
' Macro recorded 2.12.2011 by Toni Hakala

Sub All()
Dim Answer As Integer
Answer = MsgBox(prompt:="Are you sure
you want to reset all FCU settings?", _
Buttons:=vbYesNo + vbQuestion, Title:="FCU
settings")
If Answer = vbYes Then
    Application.ScreenUpdating = False
    Sheets("Main").Select
    Range("K7").Select
    ActiveCell.FormulaR1C1 = "70"
    Range("K8").Select
    ActiveCell.FormulaR1C1 = "120"
    Range("I10").Select
    ActiveCell.FormulaR1C1 = "60"
    Range("I12").Select
    ActiveCell.FormulaR1C1 = "50"
    Range("K10").Select
    ActiveCell.FormulaR1C1 = "150"
    Range("K12").Select
    ActiveCell.FormulaR1C1 = "230"
    Range("M10").Select
    ActiveCell.FormulaR1C1 = "75"
    Range("M12").Select
    ActiveCell.FormulaR1C1 = "250"
    Range("T19").Select
    ActiveCell.FormulaR1C1 = "5"
    Range("T30").Select
    ActiveCell.FormulaR1C1 = "5"
    Range("E20").Select
    ActiveCell.FormulaR1C1 = "400"
    Range("E21").Select
    ActiveCell.FormulaR1C1 = "1000"
    Range("E34").Select
    ActiveCell.FormulaR1C1 = "85"
    Range("G34").Select
    ActiveCell.FormulaR1C1 = "250"
    Range("I34").Select
    ActiveCell.FormulaR1C1 = "295"
    Range("L34").Select
    ActiveCell.FormulaR1C1 = "4"
    Range("N34").Select
    ActiveCell.FormulaR1C1 = "10"
    Range("P34").Select
    ActiveCell.FormulaR1C1 = "450"
    Range("S34").Select
    ActiveCell.FormulaR1C1 = "20"
    Range("X34").Select
    ActiveCell.FormulaR1C1 = "94"
    Range("D36").Select
    ActiveCell.FormulaR1C1 = "87"
    Range("C37").Select
    ActiveCell.FormulaR1C1 = "3"
   Range("U38").Select
    ActiveCell.FormulaR1C1 = "300"
    Range("V38").Select
    ActiveCell.FormulaR1C1 = "200"
    Range("F67").Select
    ActiveCell.FormulaR1C1 = "2"
    Range("F68").Select
    ActiveCell.FormulaR1C1 = "1"
    Range("F73").Select
    ActiveCell.FormulaR1C1 = "1"
    Range("F75").Select
    ActiveCell.FormulaR1C1 = "1"
    Range("F84").Select
    ActiveCell.FormulaR1C1 = "1"
    Range("F85").Select
    ActiveCell.FormulaR1C1 = "1"
    Range("F89").Select
    ActiveCell.FormulaR1C1 = "1"
    Range("F90").Select
    ActiveCell.FormulaR1C1 = "1"
    Range("F92").Select
    ActiveCell.FormulaR1C1 = "1"
    Range("F93").Select
    ActiveCell.FormulaR1C1 = "2"
    Range("F95").Select
    ActiveCell.FormulaR1C1 = "1"
    Sheets("FCU").Select
    Range("A1").Select
Else
End If
End Sub
```