CZECH TECHNICAL UNIVERSITY IN PRAGUE
FACULTY OF MECHANICAL ENGINEERING
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APPLICABILITY OF CHILLED BEAM-SYSTEM IN THE MIDDLE EAST

BACHELOR THESIS

JACQUES MATTA 2 – EE – 2017

I declare that this bachelor thesis entitled "APPLICABILITY OF CHILLED-BEAM SYSTEM IN THE MIDDLE EAST" is my own work performed under the supervision of Vladimír Zmrhal, with the use of the literature presented at the end of my bachelor thesis in the list of references.

In Prague 27.05.2017

Jacques Matta
signature

ACKNOWLEDGMENTS

I would like to express my special thanks to my advisor Vladimir Zmrhal, for his continuous devotion, invaluable support and intellectual guidance though the course of my work.

ABSTRACT

Nowadays, everyone wants to live in buildings with comfort indoor environment. Chilled beam is a type of convection HVAC system designed to heat or cool large buildings that makes that comfort possible. As a developer, and for the comfort of the people in the building, chilled beams are quieter than VAV system and easy to maintain, space can be maximized because beam don’t need equipment/mechanical rooms or large ductwork, fan use is minimized, which saves energy, and a potential for better thermal comfort because these systems have a better air-distribution pattern. These are the advantages of the chilled beam system, but for them to work efficiently depends on many factors that are based on the weather conditions. The fact that is it energy effective pushed me to choose the chilled beam cooling as my topic, coming from Lebanon a country where energy is a real problem with lack of electricity production, high rate of pollution and with the climate change problems worldwide, it was an interesting topic to pick.

This research focuses on comparing weather conditions in Tel Aviv and Prague and whether the chilled beam system is applicable in the Middle East.

Symbols Used

t_{cc}	Surface temperature of cooling coil	[°C]
t_o	Outdoor air temperature	[°C]
t_s	Supply air temperature	[°C]
t_i	Indoor air temperature	[°C]
x_{cc}	Humidity ratio on cooling coil surface	[g/kg]
x_s	Humidity ratio of supply air	[g/kg]
x_o	Humidity ratio of outdoor air	[g/kg]
x_i	Humidity ratio of indoor air	[g/kg]
Δx_{cc}	Humidity ratio difference of cooling coil	[g/kg]
Δx_s	Humidity ratio difference of supply air	[g/kg]
Δt_{cc}	Temperature ratio difference of cooling coil	[°C]
Δt_s	Temperature ratio difference of supply air	[°C]
h_o	Enthalpy of outdoor air	[J/kg]
h_s	Enthalpy of supply air	[J/kg]
φ	Relative humidity	[%]
φ_i	Relative humidity of indoor air	[%]
p_v''	Saturation pressure	[Pa]
p_v	Partial Pressure of water vapour	[Pa]
p_{vi}	Partial Pressure of indoor air water vapour	[Pa]
ρ	Density of indoor air	[kg/m^3]
P	Atmospheric Pressure	[Pa]
t_{dp}	Dew point temperature	[°C]
θ	Angle of incidence	[°]
H	Hour angle	[°]
ϕ	Solar Azimuth	[°]
δ	Solar Declination	[°]

β	Solar Altitude	[°]
$E_{t,b}$	Beam component	[W/m^2]
$E_{t,d}$	Diffuse component	[W/m^2]
$E_{t,r}$	Ground reflected component	[W/m^2]
E_t	Total irradiance	[W/m^2]
q_{rad}	Heat flux by solar radiation through real glazing	[W/m^2]
Q_{rad}	Heat gain by solar radiation	[W]
Q_{tr}	Heat transfer through glazing by transmission	[W]
Q_{tsen}	Total sensible heat gains	[W]
V	Volume flow rate	[m^3/h]
M_v	Production of water vapour per person	[g/h]
U_w	Thermal transmittance coefficient of the window	[W/m^2.K]
ASL	Sunlit area of window	[m^2]
A_w	Total area of window	[m^2]
ASH	Shaded area of window	[m^2]
ψ	Surface solar azimuth	[°]
γ	Surface solar azimuth angle	[°]
Σ	Slope	[°]
L	Local latitude	[°]

Table of Contents

1. Introduction

Heating Ventilation and Air conditioning (HVAC) systems are designed to control the indoor environment including indoor air quality and thermal comfort for occupants. The chilled beam is a type of convection HVAC system [7]

Chilled beams units are becoming increasingly popular in building design. It is an energy efficient system that can provide with radiant cooling, heating and ventilation, it is a quiet system that decreases energy consumption in buildings and makes the area comfortable and healthy for the people. Chilled beams are often used in Europe for offices, libraries, schools and hospitals.

The thesis starts with a general review of the system, basic theoretical understanding of the technology, the calculation of weather conditions in Tel Aviv and Prague (fig. 1) during complete months through different years including outdoor, supply and indoor air conditions, solar geometry, irradiance and cooling load with respect to the room F407 that will be specified later.

After studying this work, the reader will have a clear picture of the steps taken to suggest whether chilled beam system can be used in the Middle East.

An excel sheet with the detailed calculations and a pdf file of the work plan will be attached to the work on a CD.

1.1 Location on the map [4]

The thesis is done in two different locations, calculation are done in Prague (Czech Republic) and Tel Aviv located near Lebanon.

Figure 1: Map with three locations: Czech Republic, Lebanon and Tel Aviv

1.2 Data for Prague and Tel Aviv

In figure 2 a temperature difference (between 10 to 20 °C) is noticeable between Prague and Tel Aviv. With a high of 37°C in Tel Aviv and 30 °C in Prague, as well as a low of 0 °C and -13 °C respectively.

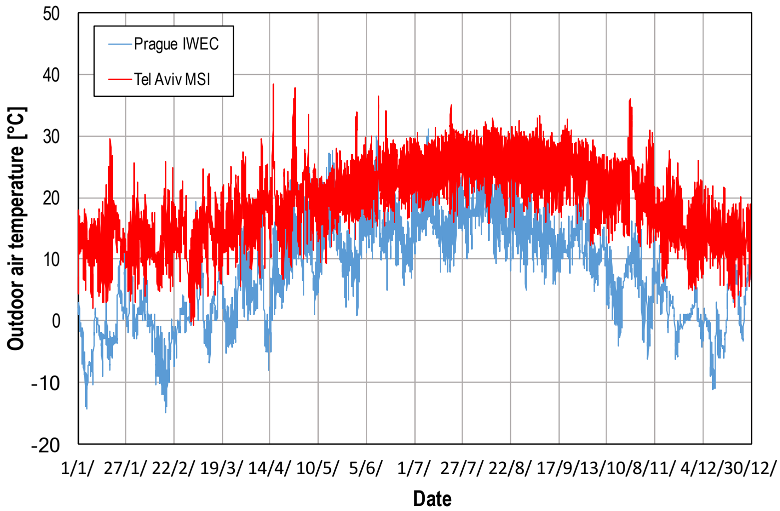

Figure 2: Outdoor temperature in Prague (blue) and Tel Aviv (red) in function of time

In figure 3 an enthalpy difference (up to 30 kJ/kg) is noticeable between Prague and Tel Aviv. With a high of 83 kJ/kg in Tel Aviv and 58 kJ/kg in Prague, as well as a low 9 kJ/kg and -10 kJ/kg respectively.

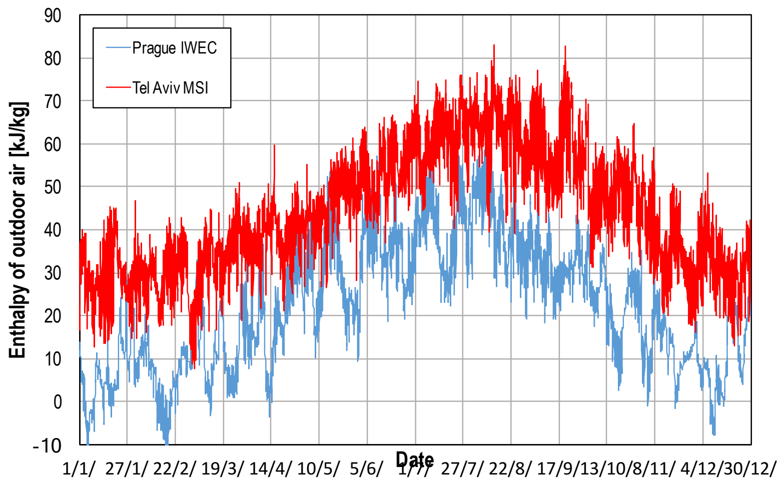

Figure 3: Enthalpy of outdoor air in Prague (blue) and Tel Aviv (red) in function of time

In figure 4 a small relative humidity difference is noticeable depending on the months. For example during the months of January and December the relative humidity in Tel Aviv is much lower than in Prague

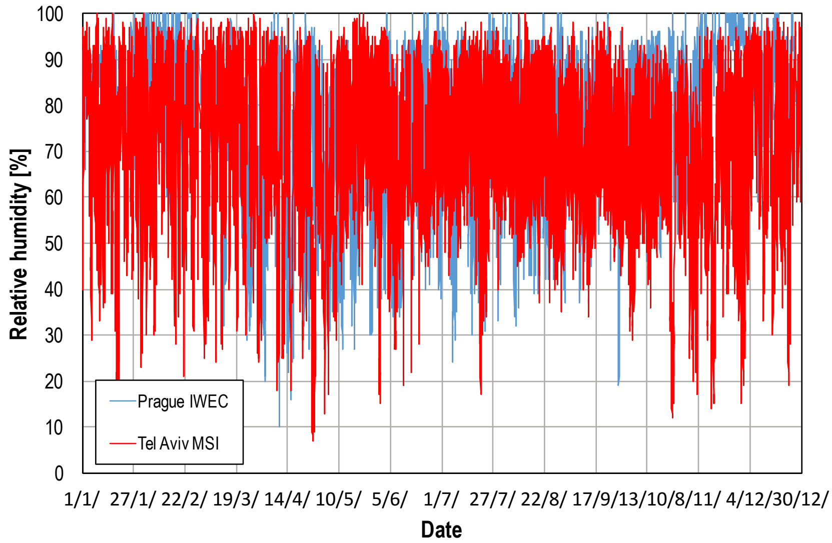

Figure 4: Relative humidity in Prague (blue) and Tel Aviv (red) in function of time

In figure 5 the global horizontal radiation is much higher in Tel Aviv than in Prague which is logical when we compare the temperature in both areas.

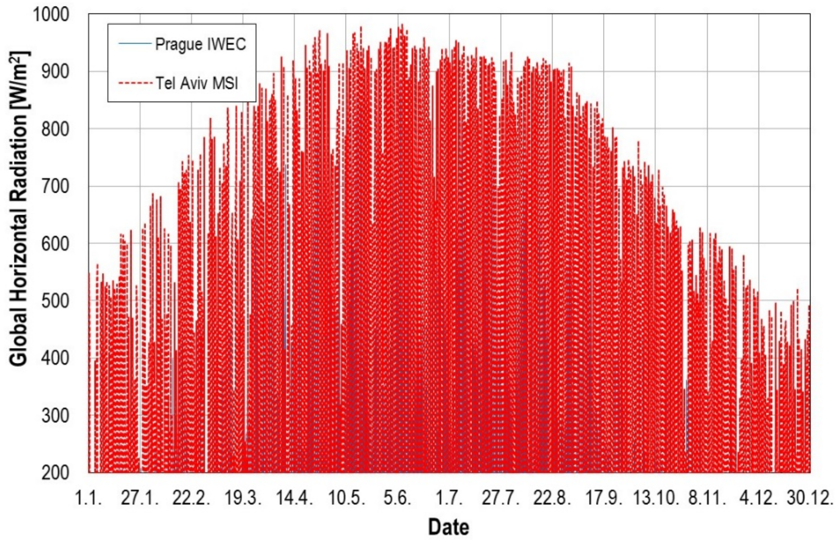

Figure 5: Global Horizontal Radiation in Prague (blue) and Tel Aviv (red) in function of time

In figure 6 the humidity ratio difference is noticeable, a high of 20 g/kg in Tel Aviv compared to 13 g/kg in Prague, with approx. difference of 8 g/kg.

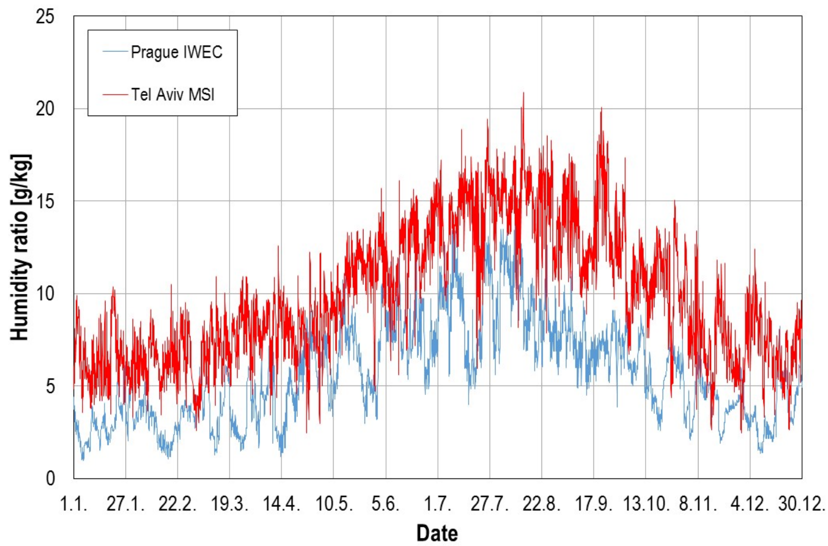

Figure 6: Humidity ratio in Prague (blue) and Tel Aviv (Orange) in function of time

For an office space, the occupant density is equal to 5 per 100 m^2 so equal to 20 per 1m^2 . By Multiplying the density by the area outdoor air rate 0,3 L/s.m^2 we get 6 L/s. By adding it to the People outdoor air rate (5 L/s) we finaly get 8,5 L/s which is the combine outdoor air rate for offices. Moreover 8,5 L/s is approx. equal to 30 m^3/h

TABLE 6-1 MINIMUM VENTILATION RATES IN BREATHING ZONE
(This table is not valid in isolation; it must be used in conjunction with the accompanying notes.)

Occupancy Category	People Outdoor Air Rate R_p		Area Outdoor Air Rate R_a		Notes	Default Values: Occupant Density (see Note 4)	Default Values: Combined Outdoor Air Rate (see Note 5)		Air Class
	cfm/person	L/s·person	cfm/ft²	L/s·m²		#/1000 ft² or #/100 m²	cfm/person	L/s·person	
Hotels, Motels, Resorts, Dormitories									
Bedroom/living Room	5	2.5	0.06	0.3		10	11	5.5	1
Barracks sleeping areas	5	2.5	0.06	0.3		20	8	4.0	1
Lobbies/prefunction	7.5	3.8	0.06	0.3		30	10	4.8	1
Multi-purpose assembly	5	2.5	0.06	0.3		120	6	2.8	1
Office Buildings									
Office space	5	2.5	0.06	0.3		5	17	8.5	1
Reception areas	5	2.5	0.06	0.3		30	7	3.5	1
Telephone/data entry	5	2.5	0.06	0.3		60	6	3.0	1
Main entry lobbies	5	2.5	0.06	0.3		10	11	5.5	1

GENERAL NOTES FOR TABLE 6–1

1 **Related Requirements:** The rates in this table are based on all other applicable requirements of this standard being met.
2 **Smoking:** This table applies to no-smoking areas. Rates for smoking-permitted spaces must be determined using other methods. See Section 6.2.9 for ventilation requirements in smoking areas.
3 **Air Density:** Volumetric airflow rates are based on an air density of 0.075 lb_{da}/ft^3 (1.2 kg_{da}/m^3), which corresponds to dry air at a barometric pressure of 1 atm (101.3 kPa) and an air temperature of 70°F (21°C). Rates may be adjusted for actual density but such adjustment is not required for compliance with this standard.
4 **Default Occupant Density:** The default occupant density shall be used when actual occupant density is not known.
5 **Default Combined Outdoor Air Rate (per person):** This rate is based on the default occupant density.
6 **Unlisted Occupancies:** If the occupancy category for a proposed space or zone is not listed, the requirements for the listed occupancy category that is most similar in terms of occupant density, activities and building construction shall be used.
7 **Residential facilities, Healthcare facilities and Vehicles:** Rates shall be determined in accordance with Appendix E.

Figure 4.5 Parts of *Table 6-1, ASHRAE Standard 62.1-2004*

Figure 7: Minimum ventilation rates in breathing zones in Tel Aviv [2]

1.3 Office used for calculations (4F07)

The room used for calculation is an office room on the 4th floor of an office building situated in Lebanon. It is an office with an area of 13,92 m^2 for two person, with an LCD screen, two computers and a printer.

It has a façade facing south, two windows with double glazing with an area of 1,42 m^2 each.

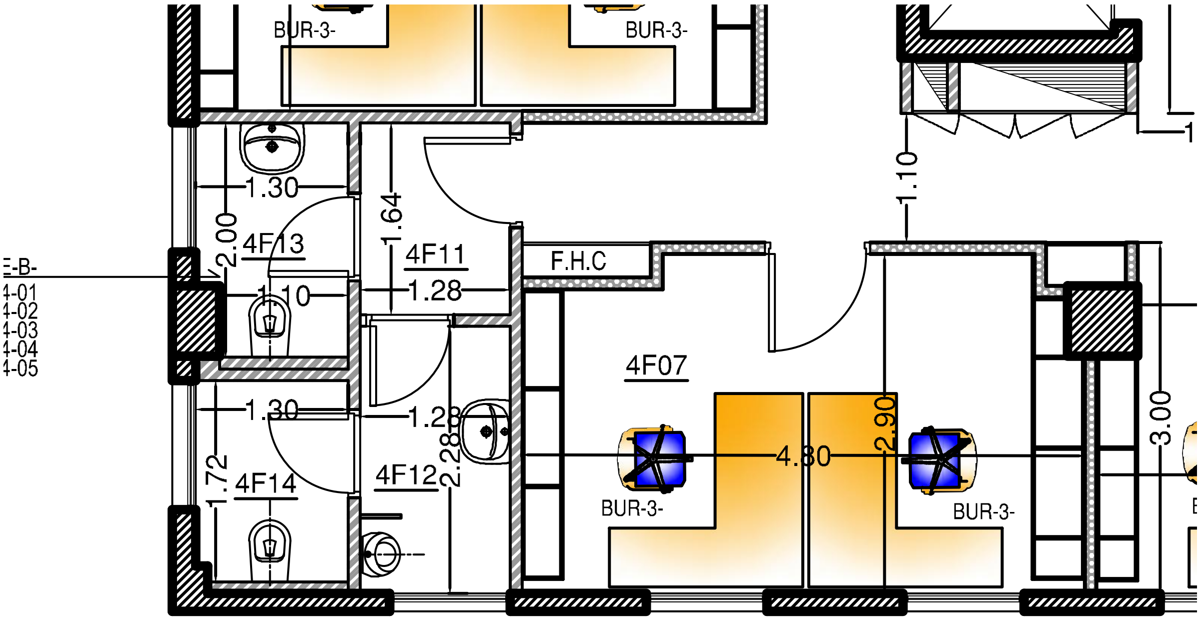

Figure 8: Office used for calculation (Room 4F07)

2. Description of the system

Chilled beam is a modern technology used to provide cooling, heating or ventilation to the internal spaces of buildings.

2.1 Definition of chilled beams

Chilled beams where first used in the Shell headquarters in London in 1960's, it became popular recently due to the energy consumption efficiency and its importance nowadays in buildings. Generally, chilled beams are distributed across the ceiling of a space. It has chilled water pipes to it usually made of copper coils bonded to aluminium fins that cool the air by convection. [5]

2.2 Types of chilled beams

There are two types of chilled beams: active chilled beams and passive chilled beams.

Active chilled beams

Active chilled beams uses the ventilation system to increase the output of the coil and to handle both latent and sensible loads of a space. It is a convector with integrated air supply where air passes through the cooling coils. The cooling medium in the coil is water. The beam is normally under the ceiling. [3]

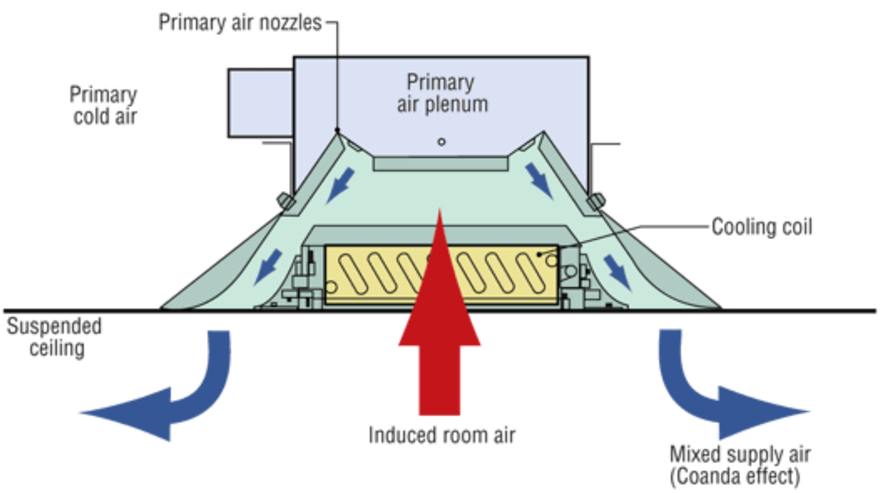

Figure 9: Picture showing an active chilled beam system [3]

Passive chilled beams

A passive chilled uses a heat exchanger usually a coil to change the temperature of the transferring heat and create a difference in density with ambient air. The cooled element is fixed in, above or under a ceiling fitted with a cooling coil mainly convectively using natural airflows. [3]

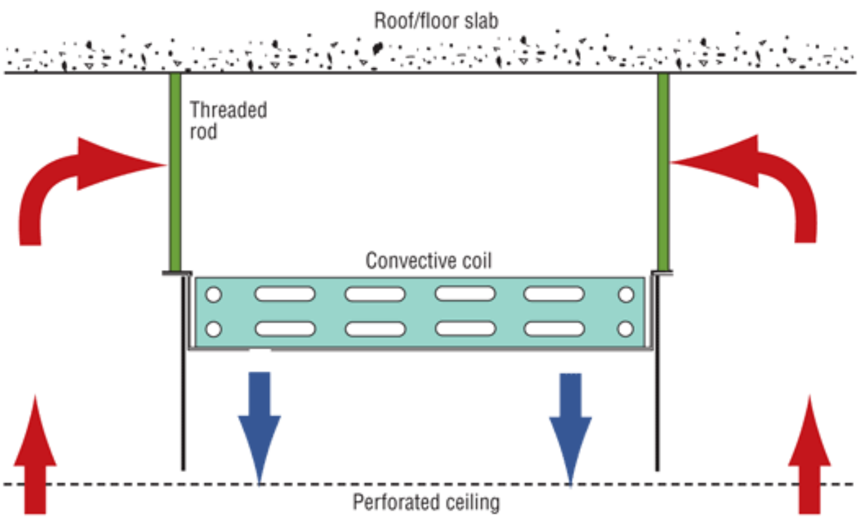

Figure 10: Picture showing a passive chilled beam system [3]

2.3 Principle of operation

As the beam chills the air around it, the air becomes denser, moves down to the floor and takes the place of the hot air that moves up to get cooled down. The system provides excellent thermal comfort with good energy efficiency. [6]

2.4 Operation range

The recommended water intlet temperature for cooling is between 14 °C and 16 °C. Condensation must be avoided and will happen if the dew point of the indoor air temperature is higher than the water inlet temperature

2.5 Heat Transfer

Convection is the main transfer method in chilled beam, the transfer of heat from one place to another by the movement of fluids. The layer with higher density, lower temperature flows down to the room and the layer with lower density goes up.

2.6 Advantages and disadvantages of the chilled beam

The chilled beam gives excellent thermal conditions, the maintenance is easy as there is no moving part like in a fan coil unit, there are no drains, fans or filters that needs change. It is a quiet operation system. [5]

On the other hand, chilled beam are less suited for places where the heat loads are very heavy and when condensation might happen. This study will give out more information and details about that.

3. Risk of condensation analysis

Calculations are going to be shown first in Prague, the beam cooling system will be working during the work hours only, so between 7:00 am and 19:00 pm

The calculation shown on this paper are done on the 11/09/1988 at 13:00 (Filled in yellow on the excel sheet)

The goal of these calculation is to calculate the risk of condensation in both Prague and Tel Aviv, and conclude if the application of chilled beam is possible.

Later on, capacity required will also be calculated and compared.

3.1 Calculation in Prague

Calculation of Psychometrics of Outdoor air

Given:

Indoor air temperature t_i = 26 °C

Supply air temperature t_s = 16 °C

Temp. of cooling coil t_{cc} = constant = 5 °C

Production of water vapour M_v = 116 g/h per person

Example: outdoor conditions on the 11/09/1988 at 13:00

Outdoor air temperature t_o = 17,3 ^{o}C

Relative humidity of outdoor air φ = 64 %

3.1.2 Calculation of the saturation pressure [1]

It is the state of neutral equilibrium between moist air and the condensate water phase

The maximum vapour pressure possible before the vapour start to condense at an actual temperature is called saturation pressure.

For

$t_o > 0\ ^{o}C$

$$p''_v = exp(23.58 - \frac{4044,2}{235,6 + t_o})\ [Pa]$$

Example: For $t_o = 17,3$ °C

$$p''_v = exp(23,58 - \frac{4044.2}{235,6 + 17,3}) = 1976\ Pa$$

3.1.3 Calculation of the partial pressure of water vapour [1]

The partial pressure is calculated through this formula:

$$p_v = \frac{p''_v . \varphi}{100}$$

For example:

$$p_v = \frac{1976\,.\,64}{100} = 1264\ Pa$$

3.1.4 Calculation of humidity ratio of outdoor air [1]

It is the mass of water vapour per unit mass of dry air.

$$x_o = 0,622\ \frac{p_v}{p - p_v}$$

For example:

$$x_o = 0,622\left(\frac{1264.5}{1000\,.\,97,42 - 1264,5}\right)1000 = 8,2\ g/kg$$

3.1.5 Calculation of enthalpy of outdoor air [1]

The enthalpy of outdoor air is the total heat content in the outdoor air

Calculated using:

$$h_o = c_v \, . t_o + x_o * (L + \; c_a . t_o)$$

For example:

$$h_o = \; 1{,}01 \, . \, 17{,}3 + \left(\frac{8{,}2}{1000}\right)(2500 + 1{,}84 \, . \, 17{,}3) = 38{,}2 \, J/kg$$

3.2 Calculation of psychometrics of supply air [1]

Humidity ratio of cooling coil is calculated as

$$x_{cc} \; = 0{,}622 * \frac{p_v''}{p - p_v''}$$

For example:

$$x_{cc} = \; 0{,}622 \; \frac{1976}{97{,}42 \, . \, 1000 - 1976} = 5{,}6 \; g/kg$$

3.2.1 Calculation of the humidity ratio difference on the cooling coil surface [1]

If, $x_o < x_{cc}$ then $\Delta x_{cc} = 0 \; g/kg$

Otherwise,

$$\Delta x_{cc} = x_o - x_{cc}$$

In our example we have $x_o > x_{cc}$ so we get:

$$\Delta x_{cc} = x_o - x_{cc} = 8{,}2 - 5{,}6 = 2{,}6 \; g/kg$$

3.2.2 Calculation of the temperature difference on the cooling coil surface [1]

Calculated only if $t_o > t_s$ by

$$\Delta t_{cc} = t_o - t_{cc}$$

Here, t_o =17,3 °C , t_s = 16 °C and t_{cc} = 5 °C

So for example we get,

$$\Delta t_{cc} = t_o - t_{cc} = 17{,}3 - 5 = \ 12{,}3\, J/kg$$

3.2.3 Calculation of the temperature difference between outdoor and supply air [1]

As said above, the temperature difference between outdoor and supply air is calculated only if $t_o > t_s$ by

$$\Delta t_s = t_o - \ t_s$$

For example:

$$\Delta t_s = t_o - \ t_s = 17{,}3 - 16 = \ 1{,}3\, ^oC$$

3.2.4 Calculation of the humidity ratio between outdoor and supply air [1]

As said above the humidity ratio between outdoor and supply air is calculated only if, $t_o > t_s$ by

$$\Delta x_s = \frac{\Delta t_s * \Delta x_{cc}}{\Delta t_{cc}}$$

For Δt_s, Δx_{cc}, Δt_{cc} calculated above, we get for example:

$$\Delta x_s = \frac{\Delta t_s * \Delta x_{cc}}{\Delta t_{cc}} = \frac{1{,}3 * 2{,}6}{12{,}3} = 0{,}3\ g/kg$$

3.2.5 Calculation of the humidity ratio of the supply air [1]

As said abone the humidity ratio of the supply air is calculated only if, $t_o > t_s$

And if, $x_o \le x_{cc}$,

⇨ $x_s = x_{cc}$

If not

⇨ $x_s = x_o - \Delta x_s$

In this example, x_o = 8,2 g/kg and x_{cc} = 5,6 g/kg so using the formula for $x_o > x_{cc}$

We get for example:

$$x_s = x_o - \Delta x_s = 8{,}2 - 0{,}3 = 7{,}9\ g/kg$$

3.2.6 Calculation of the enthalpy of supply air [1]

As said above the enthalpy of supply air is calculated only if $t_o > t_s$ by,

$$h_s = c_v.t_s + x_s * (L + c_a.t_s)$$

For example:

$$h_s = 1{,}01.16 + \left(\frac{7{,}9}{1000}(2500 + 1{,}84.16)\right) = 36{,}2J/kg$$

3.3 Calculation of indoor air psychometrics [1]

All the calculation can be done only only if, $t_o > t_s$, below on our examples, $t_o > t_s$

3.3.1 Calculation of density of indoor air [1]

$$\rho = \frac{1}{r_a t_i}(p - p_v.0{,}378)$$

For example

$$\rho = \frac{1}{287{,}1 * (26 + 273)} * (97{,}42.1000 - 0{,}378.1264{,}5) = 1{,}129\ kg/m^3$$

3.3.2 Calculation of humidity ratio of indoor air [1]

$$x_i = \frac{M_w}{V * \rho} + x_s$$

For Example

$$x_i = \frac{M_w}{V * \rho} + x_s = \frac{116}{50 * 1{,}129} + 7{,}9 = 9{,}96\ g/kg$$

3.3.3 Calculation of the indoor air relative humidity [1]

The amount of water vapour present in air expressed as a percentage of the amount needed for saturation at the same temperature

$$\varphi_i = \frac{x_i}{1000.p\ \ .1000} \frac{1}{p''_{vi}.(0{,}622 + x_i/1000)}$$

For example:

$$\varphi_i = \frac{9{,}96}{1000.97{,}42.1000} \frac{1}{3363.(0{,}622 + 9{,}96/1000)} = 46\%$$

3.3.4 Calculation of Partial pressure of water vapour [1]

The partial pressure of water vapour is calculated by,

$$p_{vi} = p''_{vi}\varphi_i$$

In our case for example we get:

$$p_{vi} = p''_{vi}\varphi_i = \frac{46}{100}3363 = 1536\ Pa$$

3.3.5 Calculation of the dew point temperature [1]

The dew point temperature is the atmospheric temperature below which water droplets begin to condense and dew point form

Our goal is to get the dew point temperature as condensation risk is the most important factor.

$$t_{dp} = \frac{-4044{,}2}{lnp_{vi} - 23{,}58} - 235{,}6$$

For example

$$t_{dp} = \frac{-4044{,}2}{lnp_{vi} - 23{,}58} - 235{,}6 = \frac{-4044{,}2}{\ln\ 15360 - 23{,}58} - \ 235{,}6 = 13{,}4\,℃$$

3.3.6 Finding the dew point on the Psychometric Chart

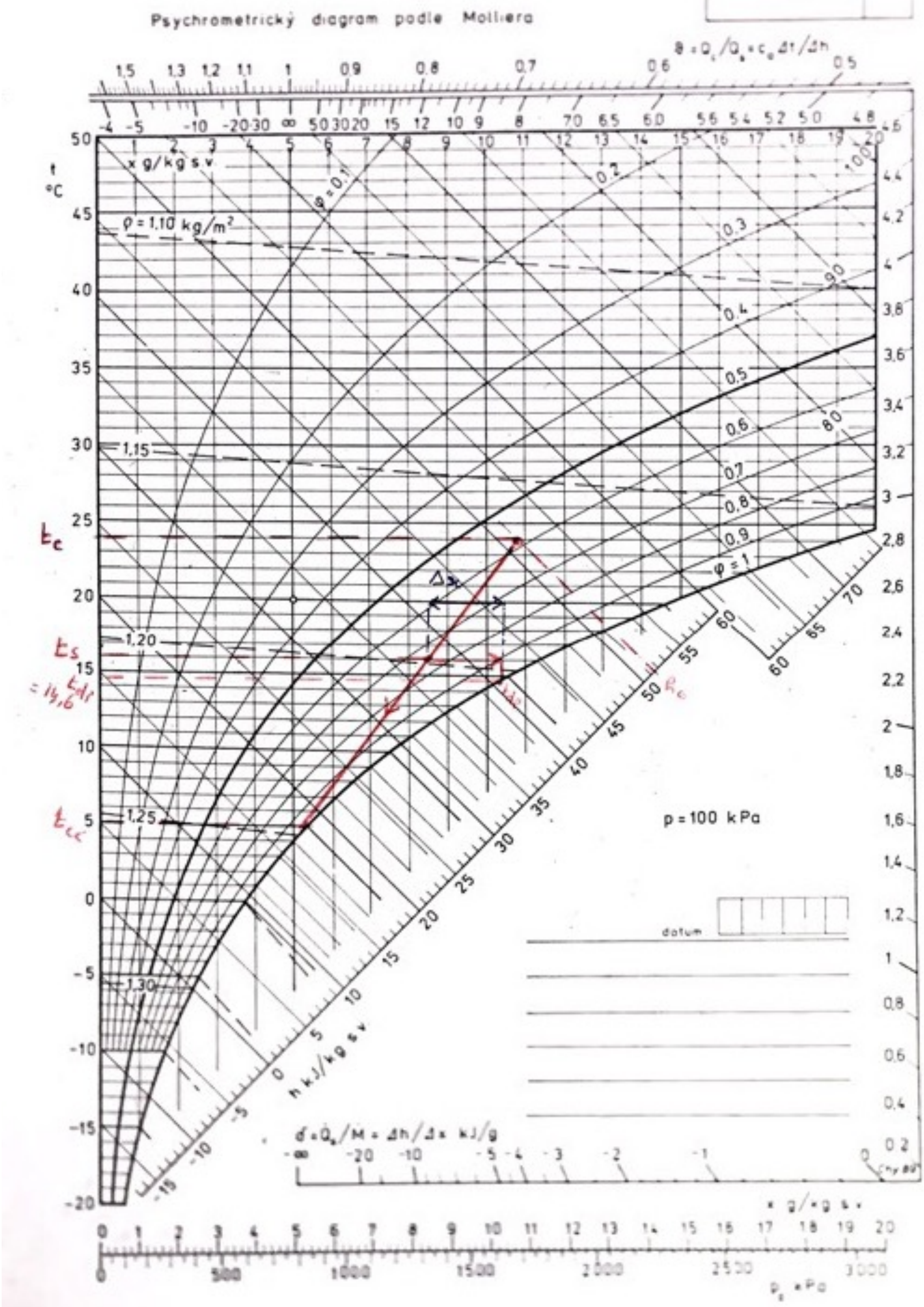

Figure 11:Psychometric chart, method to find the dew point

3.3.7 Graphs comparison between Prague and Tel Aviv

The following graphs shows the risk of condensation in Prague (blue) and Tel Aviv (Orange) for different volume flow rate used. It is obvious that with a higher volume flow rate the risk of condensation is smaller.

Nevertheless, by using a volume flow rate of 25 to 35 m^3/h, the risk of condensation at 16 °C is between 50-70 % in Tel Aviv and 20-40% in Prague. These values are very big and very negative for the applicability of a chilled beam.

By supplying a flow of 50 m^3/h, the risk of condensation in Prague lowers to 10 % and in Tel Aviv to 20 %. Again the risk of condensation is still high especially in Tel Aviv.

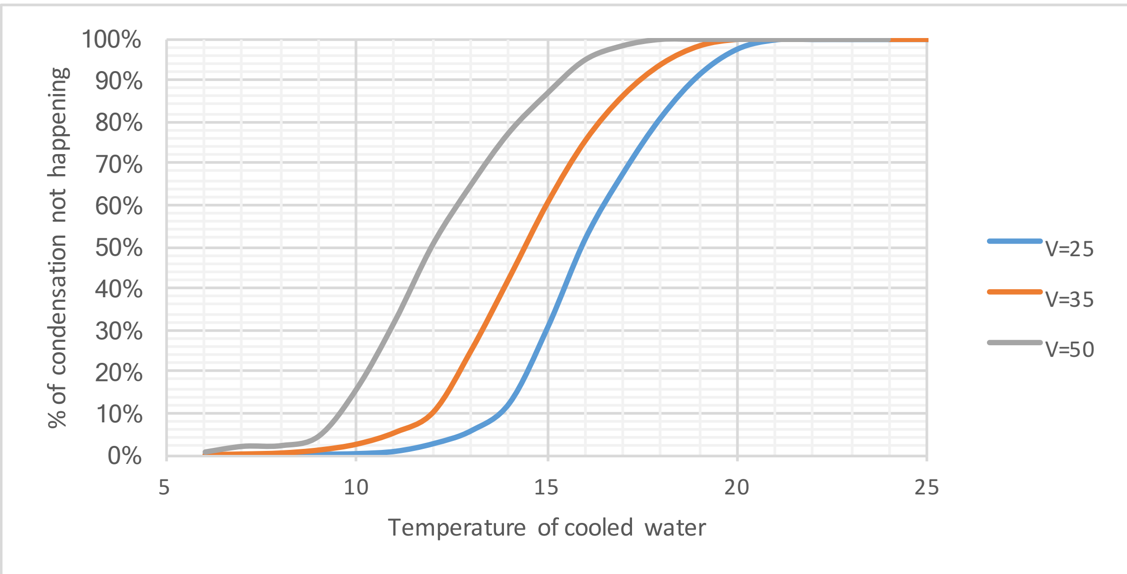

Figure 12:Graph showing the risk of condensation in Prague for different water temperature and different flow rate

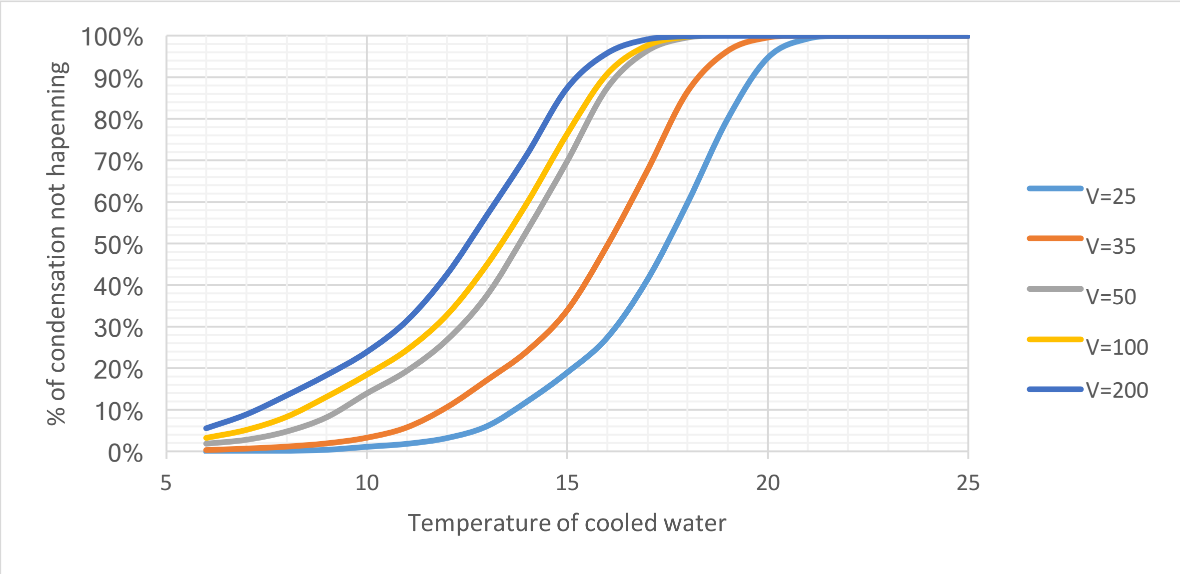

Figure 13: Graph showing the risk of condensation in Tel Aviv for different water temperature and different flow rate

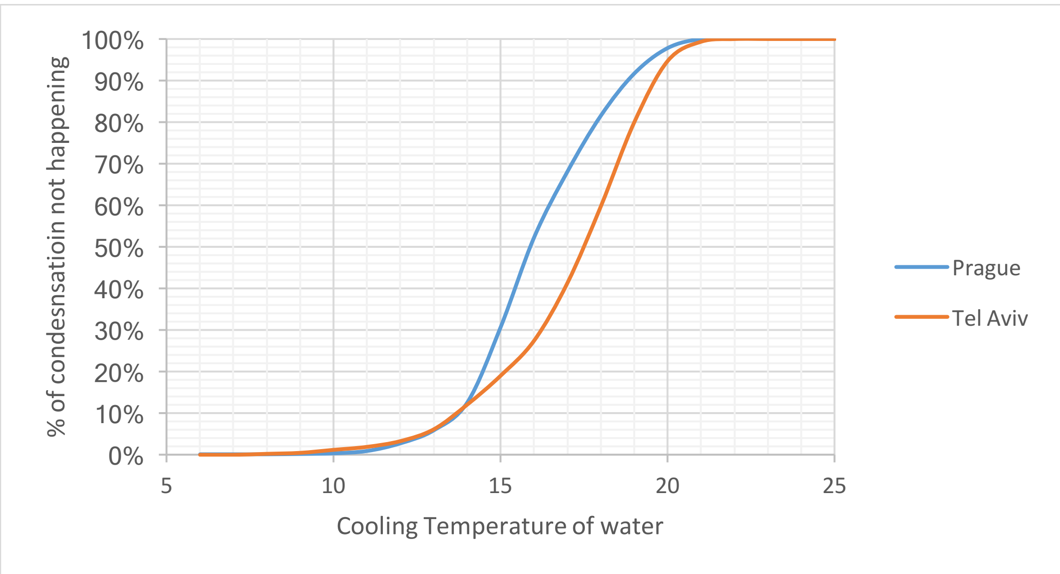

Figure 14: Graph comparing the risk of condensation in Prague and Tel Aviv for a volume flow rate of 25 m^3/h per person

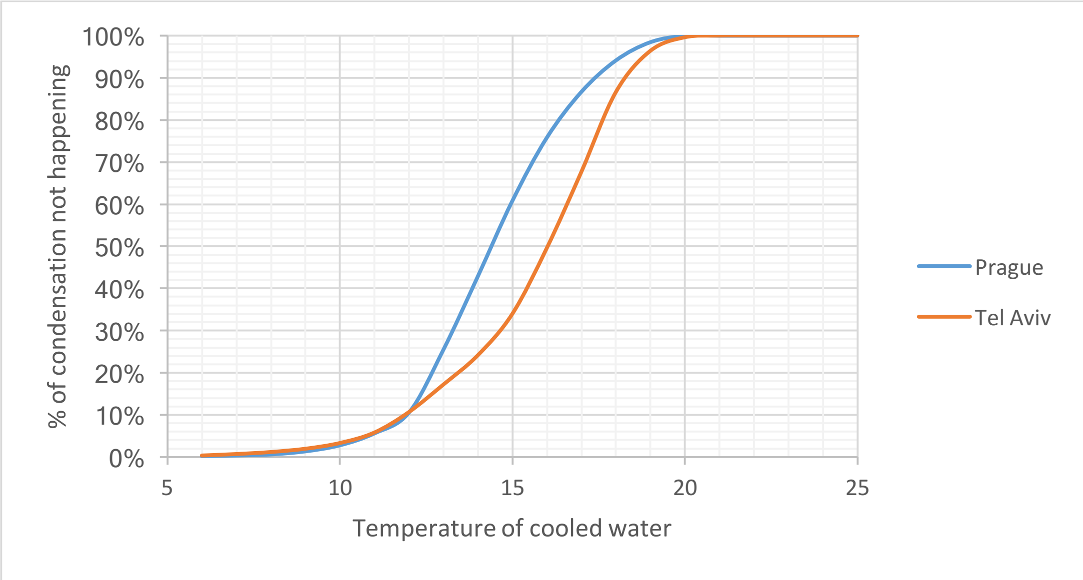

Figure 15: Graph comparing the risk of condensation in Prague and Tel Aviv for a volume flow rate of 35 m³/h per person

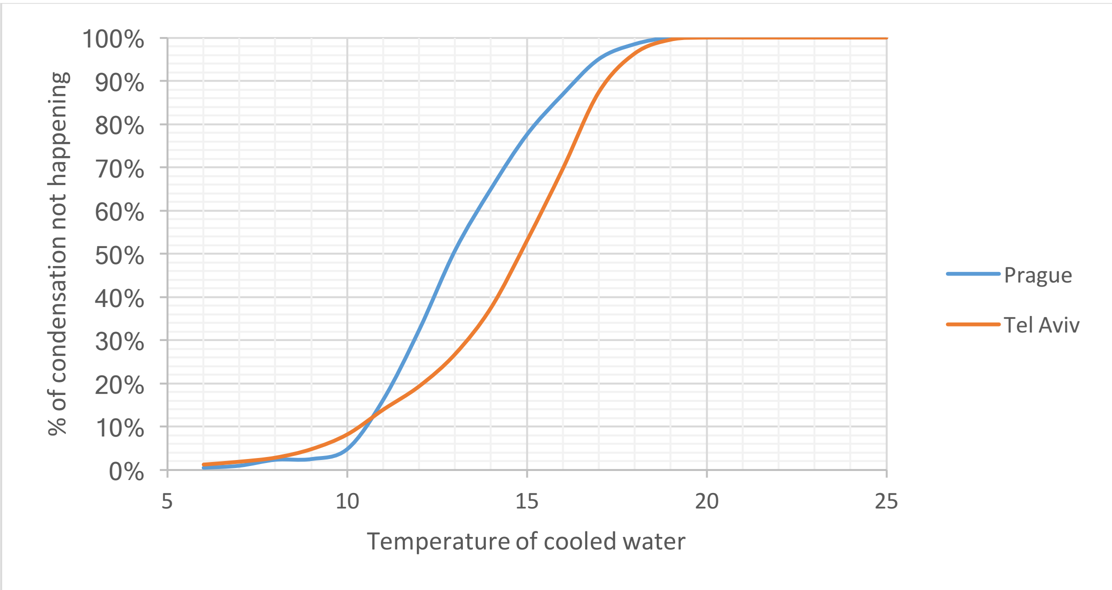

Figure 16: : Graph comparing the risk of condensation in Prague and Tel Aviv for a volume flow rate of 50 m³/h per person

3.4 Calculation of Cooling load

Solar Geometry

Calculation of solar geometry helps calculate the radiation heat loads.

With a local latitude, L = 50,1°

Table: Surface Orientations and Azimuths, Measured from South [1]

Orientation	N	NE	E	SE	S	SW	W	NW
Surface Azimuth Ψ	180°	-135°	-90°	-45°	0	45°	90°	135°

The façade chosen is South, the surface azimuth is $\Psi = 0^o$

The figure 17 shows the solar angles for Vertical and Horizontal Surfaces

The important angles are shown in that figure; as the Solar Altitude, Solar Azimuth etc.

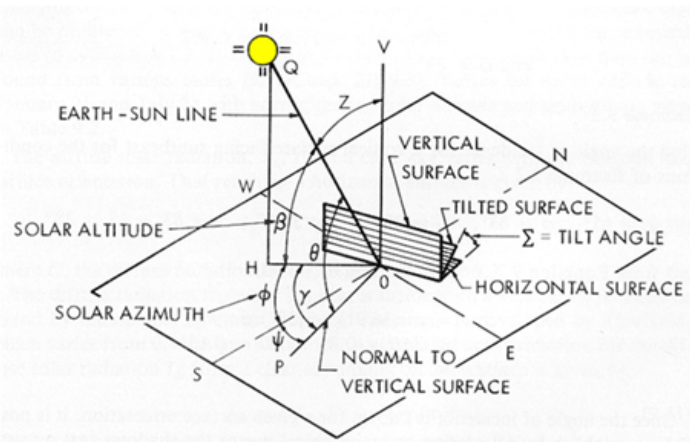

Figure 17: Solar angles for Vertical and Horizontal Surfaces

3.4.1 Calculation of Solar Declination [1]

The angle between the earth sun line and the equatorial plane

$$\delta = 23{,}45.\,sin\left(\frac{360.\,(Year\ day + 284)}{365}\right)$$

For example, the calculation being done on the 254th day of the year,

$$\delta = 23{,}45.\sin\left(\frac{360.\,(254 + 284)}{365}\right) = \ 3{,}82\,^{\circ}$$

3.4.2 Calculation of Solar Altitude [1]

The angle between the horizontal plane and a line emanating from the sun

$\beta = \operatorname{asin}(sinL.\,sin\delta + cosL.\,cos\delta.\,cosH)$

With Hour Angle H,

$H = 15.\,(hour - 12) = 15 * (\,9 - 12) = \ -45°$

For Example:

$\beta = \operatorname{asin}(sin50{,}1.\,sin3{,}82 + cos50{,}1.\,cos3{,}82.\cos(-45)) = 30{,}24\,^{\circ}$

If, $\beta < 0°$, we get $\beta = 0°$

3.4.3 Calculation of Solar Azimuth [1]

The angular displacement from south of the projection is,

$\phi = 0° \; if, \beta = 0°$

If not, we calculate the solar azimuth using

$$\phi = asin\left(sinH \frac{cosH}{cos\beta}\right)$$

For, $\beta = 30{,}24°$ we get for example:

$$\phi = asin\left(sinH \frac{cosH}{cos\beta}\right) = asin(sin(-45)\frac{cos\,(-45)}{cos\,(30{,}24)} = -54{,}75°$$

3.4.4 Calculation of the surface solar azimuth angle [1]

The surface solar azimuth angle is,

$\gamma = 0°$ if, $\beta = 0°$

If not, we calculate the surface solar azimuth angle using,

$\gamma = \phi - \psi$

The Facade taken is south so, the surface azimuth is $\psi = 0°$

For example:

$\gamma = \phi - \psi = -54{,}75 - 0 = -54{,}45°$

3.4.5 Calculation of the Angle of incidence [1]

It is the angle between the line normal to the irradiated surface and the earth-sun line.

$\theta = 0°$ *if,* $\beta = 0°$

if not we calculate the angle of incidence by,

$\theta = (acos(cos\,\beta)).\,cos\,\gamma$

For example, for $\beta = 30{,}24°$ we get,

$\theta = (acos(cos\,\beta)).\,cos\,\gamma = acos(30{,}24)\,.\,cos(-54{,}45) = 60{,}10°$

3.5 Calculation of Irradiance [1]

It is the radiant flux (power) received by a surface per unit area

3.5.1 Calculation of Beam Component [1]

The equation for the calculation of the beam component is

$E_{t,b} = 0{,}18.cos\theta$

Example for $\theta = 60{,}1°$:

$E_{t,b} = 0{,}18.cos\theta = 0{,}18.\mathring{u}\,cos(60{,}1) = 148\ \ W/m^2$

3.5.2 Calculation of Diffuse component [1]

The equation for the diffuse component is:

$$E_{t,d} = G_{dh}\frac{(1 + cos(\Sigma))}{2}$$

Example for $\Sigma\ = 90°\ and\ G_{dh} = 314$

$E_{t,d} = G_{dh}\frac{(1+cos(\Sigma))}{2} = 314\frac{1+\cos(90)}{2} =\ 89\ W/m^2$

3.5.3 Calculation of the ground reflected component [1]

The equation for the ground reflected component is:

$E_{t,r} = (G_{bn} * cos\beta + G_{dh}).\rho_g.(\frac{1-COS\Sigma}{2})$

Example for $\Sigma\ = 90°,\ \ G_{dh} = 314\ and\ \ G_{bn} = 296$

$E_{t,r} = (G_{bn} * cos\beta + G_{dh}).\rho_g.(\frac{1-COS\Sigma}{2}) = 32\ W/m^2$

3.5.4 Calculation of the total irradiance [1]

The total irradiance is the sum of the beam component, the diffuse component and the ground reflected component

$E_t = E_{t,b} + E_{t,d} + E_{t,r}$

For example:

$E_t = E_{t,b} + E_{t,d} + E_{t,r} = 148 + 89 + 32 = 269\ W/m^2$

3.6 Calculation of Cooling Load – simplified method

Optical properties of real glazing : Double glass + internal blinds

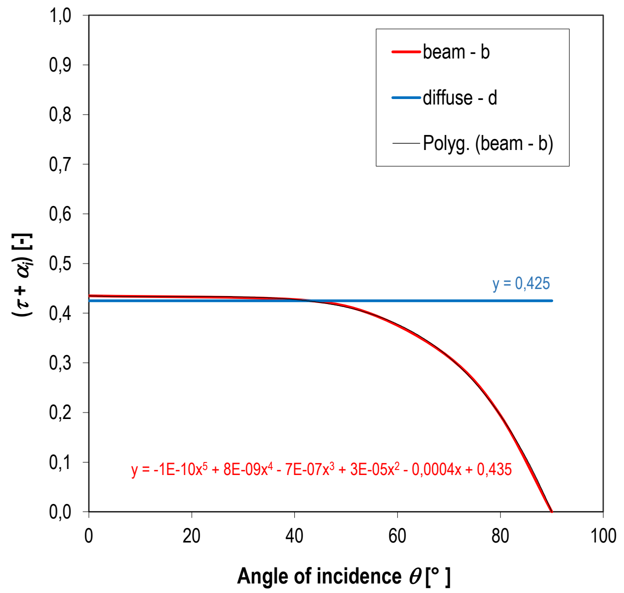

Figure 18*: Chart showing the optical properties of real glazing*

$(\tau + \alpha)_d = 0{,}425n$ [-]

$(\tau + \alpha)_b =$

$= -1E - 10.\theta^5 + 8E - 09.\theta^4 - 7E - 07.\theta^3 + 3E - 05.\theta^2 - 0{,}0004.\theta + 0{,}435$

$= 0{,}39$ [-]

3.6.2 Calculation of the heat flux by solar radiation through real glazing [1]

$q_{rad} = E_{t,b}.(\tau+\alpha)_b + (E_{t,d}+E_{t,r})(\tau+\alpha)_d$

Example for, $(\tau+\alpha)_d = 0{,}425$ and $(\tau+\alpha)_b = 0{,}39$

$q_{rad} = E_{t,b}(\tau+\alpha)_b + (E_{t,d}+E_{t,r})(\tau+\alpha)_d$

$= 148\,.\,0{,}39 + (89+32)\,.\,0{,}425 = 109\ W/m^2$

3.6.3 Calculation of the heat gain by solar radiation [1]

$Q_{rad} = (q_{rad}.ASL)$

ASL being 1,42m^2 per window,

Example for the room with two windows, ASL = 2,84 m^2

The heat gain by solar radiation is thus equal to:

$Q_{rad} = (q_{rad} * ASL) = (109.2{,}84) = 311\ W$

3.6.4 Calculation of the heat transfer through glazing by transmission [1]

$Q_{tr} = U_w.ASL.(t_o - t_i)$

Example with the thermal transmittance coefficient of the window taken as U_w = 1,4

$Q_{tr} = U_w.ASL.(t_o - t_i) = 1{,}4.2{,}84.(17{,}3 - 26) = -35\ W$

3.6.5 Calculation of internal heat gains [1]

The office is a work place for two people, with two personal computers, an LCD screen and a printer

$$Q_i = 2 * heat\ gain\ of\ person + 2 * heat\ production\ of\ PC + heat\ gain\ monitor + heat\ gain\ printer$$

$= 2.62 + 2.100 + 50 + 70 = 444\ W$

3.6.5 Calculation of the total sensible heat gains [1]

$Q_{t,sen} = Q_{rad} + Q_i + +Q_{tr}$

The total sensible heat gains is the sum of the heat gain by radiation, the heat transfer through glazing by transmission and the internal heat gains.

$Q_{t,sen} = Q_{rad} + Q_i + +Q_{tr} = 311 + 444 - 35 = 720\ W$

3.6.6 Cooling load comparison

The capacity needed in Tel Aviv is higher than in Prague, reaching a high of 1610 W and 1300 W respectively. The chart in Tel Aviv is denser with a regularity of capacity between 1000 W and 1600 W. On the other hand, the capacity in Prague is not constant, it varies from 400 W to 1200 W.

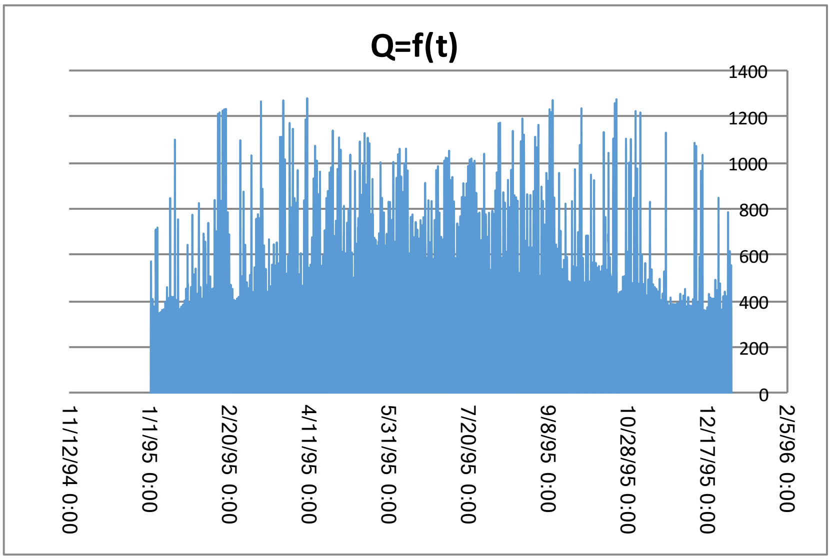

Figure 19: Graph showing the capacity needed from 11/12/1994 till 02/05/1996 in Prague

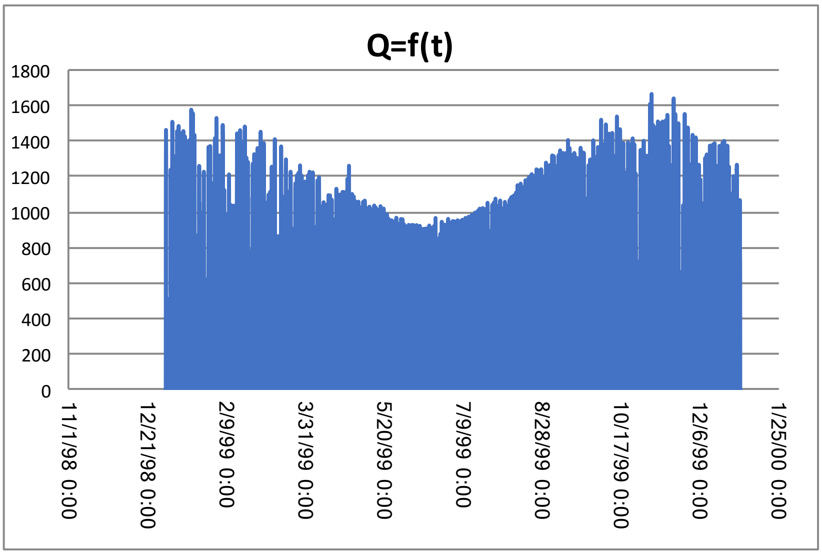

Figure 20: Graph showing the capacity needed from 11/01/1998 till 01/25/2000 in Tel Aviv

By comparing both graphs, with notice that the capacity demanded in winter is much bigger than in summer. We notice a big difference in capacity between Tel Aviv and Prague, a difference of approx. 900W. We find a difference in capacity in summer too, but a difference of approx. 200 W.

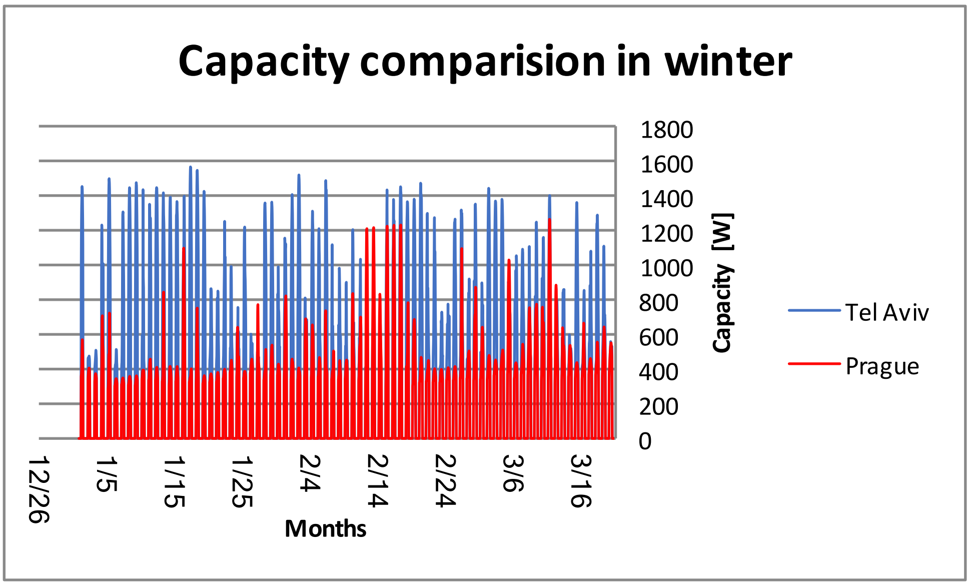

Figure 21:Graph showing the difference in capacity during winter in Prague and Tel Aviv

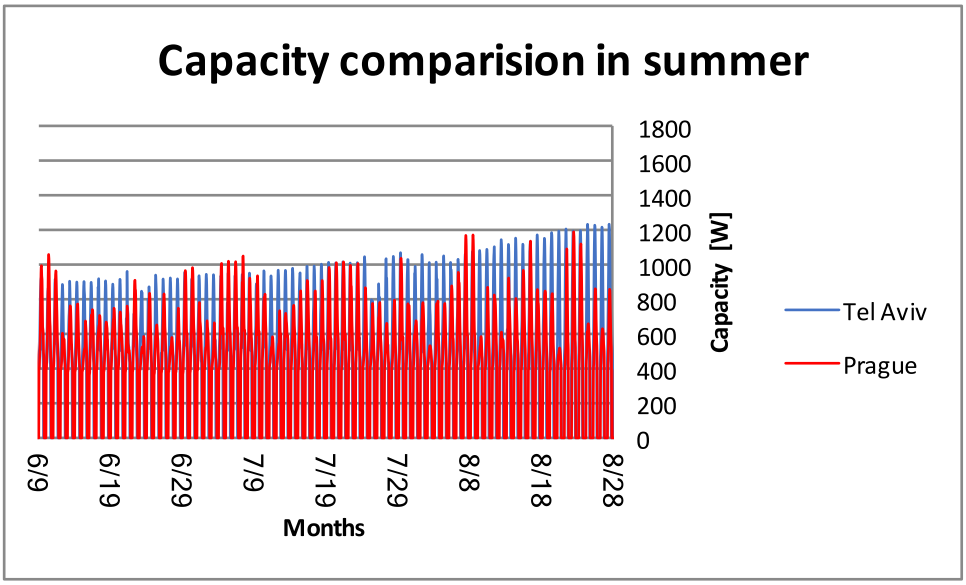

Figure 22: Graph showing the difference in capacity during summer in Prague and Tel Aviv

4. Prevention of Condensation

The most important characteristic of the chilled beam is the condensation. The surface temperature of the heat exchanger should be higher than the dew point temperature. Condensation sensors is a good way of controlling the process, and if moisture is found then the cooling water circulation pump should be stopped or the control valve shut off.

To prevent the condensation of happening, the cooling water temperature should change to a higher one, or the volume flow rate should be higher.

4.1 System Control

Our goal is to get the people to feel comfortable, to be able to do that, the control of the temperature is essential, as said above the issue is to check whether condensation will happen. Indeed, condensation will happen if the dew point of the room temperature is higher than the inlet water temperature. If the dew point temperature is lower than the inlet water temperature, then chilled beam could be used without additional use of power. If condensation does happen, the temperature of the water inlet must be changed to a higher temperature thus by using more power.

5. Conclusion

In conclusion, due to high condensation risks, the chilled beam is not the first option that could come in mind for an HVAC system, especially in Tel Aviv.

If used, the volume flow rate must be high which requires higher energy consumption, or the water of the chilled beam must be regulated.

6. Bibliography

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7. Annex

Find in the annex, a CD with the excel calculation and the thesis in PDF.