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THEME

THE STUDY OF A HEATING, VENTILATION AND AIR CONDITIONING SYSTEM FOR A HOTEL IN MEDEA

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We reach with our thanks to ALLAH all mighty for blessing us with the courage, will and motivation to successfully complete this work.

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DEDICATION

Thank you ALLAH !!!!!!!

At first I dedicate this work to my parents who sacrificed their life for me and who have never stopped encourage and support me.

To my brothers (aymen & walid) who helped me a lot with this work.

To my partner my colleague my brother

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Who without this work and this year would have never been possible.

To all my friends and family whom have made my life pleasant and full of good memories.

-Imad

DEDICATION

‘’I humbly dedicate this modest work to my precious mother and father, my younger brother Zoheir, the MESSADI and the HADJADJ family as well as all of friends and loved ones whom I cherish and appreciate to the fullest, and not to mention Imad Guedmani who has been more than a partner in this journey, but a faithful *brother and an amazing friend’’*

‘’Well mom, you finally got yourself an engineer’’.

-Mounir

ملخص

يعالج هذا المشروع المطروح دراسة الاحتياجات الحرارية و كذا التجهيزات اللازمة الخاصة بالتدفئة، التهوية و التكييف من أجل بلوغ الراحة الحرارية المثلى لفندق تحت أسم فندق لزهر و المتواجد بلدية المدية بولاية المدية، و الذي يعتمد فيه هذا التجهيز على نظام ذو أساس هوائي مدعم بجهاز تعديل الهواء بنسب حجم هواء متغير.

الكلمات المحورية: تدفئة، تكييف، التهوية، الراحة الحرارية، الانتقال الحراري، الاكتساب الحراري، الفقدان الحراري، وحدة مناولة الهواء، حجم هواء متغير.

SYNOPSIS

This project is the study of the calorific needs as well as the appropriate heating, ventilating and air conditioning system required to provide thermal comfort for a hotel establishment under the name of HOTEL LAZHAR in the chief town of Medea using an aero-based system with variable air handling to achieve maximal comfort.

Keywords: HVAC, thermal transfer, thermal comfort, heat loss, heat gain, air handling unit, variable air volume, pressure drop, air flow.

RESUME

Ce projet de fin d’étude prend on compte l’étude des besoins calorifique ainsi que l’équipement de chauffage, ventilation et conditionnement d’air nécessaire a fin de fournir un confort thermique optimal a un hôtel sous le nom de HOTEL LAZHAR situé au chef lieu de la Wilaya de Médéa, en basant sure un system aéraulique qui permet le control variable de l’air conditionné.

Les mots clés: HVAC, transfert thermique, confort thermique, déperdition de chaleur, apport de chaleur, central de traitement d’air, perte de charge, débit d’air, volume d’air variable.

Table of content

Synopsis

General Introduction **1**

Chapter I: Project presentation and description .....

I.1 Introduction 2

I.2 The importance of air conditioning 3

I.3 Thermal Comfort 4

I.4 HVAC systems classifications 5

I.5 Project presentation 5

I.5.1 Project goal 5

I.5.2 Geographical localization 5

I.5.2.1 External climate conditions 6

I.5.2.2 Internal climate conditions 6

I.5.3 Architectural plan 7

I.5.4 Project specifications 13

I.5.4.1 Ground floor specifications 13

I.5.4.2 First floor specifications 14

I.5.4.3 Second and third floor specifications 15

I.5.4.4 Fourth floor specifications 17

Chapter II: Heat loss and thermal calculation .....

II.1 Introduction 19

II.2 Heat transfer modes 19

II.2.1 Conduction heat transfer 19

II.2.2 Convection heat transfer 20

II.2.3 Radiation heat transfer 21

II.3 Interior/exterior surface heat transfer coefficients 22

II.4 Calculating overall heat transfer coefficients K 22

II.4.1 Calculating overall heat transfer coefficients of the different components of the project 23

II.4.2 Building components 24

II.4.3 Coefficient K calculation example 28

II.4.4 Final results of the calculated coefficient K 29

II.5 Calculating heat balance in winter season 29

II.5.1 Presentation of the Regulatory Technical Document 29
II.5.1.1 Objective of the RTD C3-2 29
II.5.1.2 Areas of usage 30
II.5.1.3 Types of heat loss 30
II.5.1.4 Calculations 30
II.6 Calculating winter season heat balance 35
II.6.1 Calculating the unheated volume temperature tn 35
II.6.2 Example of calculation of a room and final calculation results 37
Chapter III: Heat gain calculation .....
III.1 Introduction 50
III.2 Presentation of the Regulatory Technical Document 50
III.2.1 Objective of the RTD C3-4 51
III.2.2 Calculations 51
III.2.2.1 Calculating Sensitive heat gain 52
III.3 Example of calculation of a room 62
III.4 Heat gains calculation results 69
Chapter IV: HVAC system selection .....
IV.1 Introduction 75
IV.2 HVAC system selection 75
IV.2.1 Introduction 75
IV.2.2 HVAC system types 75
IV.3 HVAC system selection criteria 78
IV.4 Project HVAC system comparison 79
IV.4.1 Comparison between types 79
IV.5 Packaged heating and air cooling unit selection 82
IV.5.1 Features and model selection 83
IV.5.2 Options and accessories 83
IV.5.3 Nominal capacity and technical data 84
IV.5.4 Dimensions and hydraulic connections 85
IV.5.5 Model selection 87
IV.6 Diffuser units and variable air volume 87
IV.6.1 Definition of diffuser in HVAC 87
IV.6.2 Definition of Variable air volume 88
IV.6.3 Variable air volume model selection 88
IV.6.4 Application, function and design 89

Chapter V: Ductwork sizing and efficiency calculation.....
V.1 Ductwork sizing, calculation and design for efficiency 94
V.2 Air distribution.....94
V.2.1 Design consideration 95
V.3 Air distribution ducts.....96
V.4 Duct sizing.....100
V.5 Duct fittings sizing 101
V.6 Pressure drop calculation and duct sizing results 103
V.6.1 Ground floor 103
V.6.2 First floor 105
V.6.3 Second floor 106
V.6.4 Third floor.....108
V.6.5 Fourth floor 109
V.6.6 Vertical duct.....111
V.7 Blower fan's Pressure drop 112
V.8 Blower fan's Pressure drop 113
V.8.1 Fan selection 113
V.8.2 Pulley selection 113
General Conclusion
Bibliographical References
Annex

List of abbreviations

Sign	Nominal sign meaning	Unit
Ti°	Temperature	C°
Φi	Relative Humidity	%
Qcond	Heat transfer flow by conduction	W
λ	Coefficient of thermal conductivity	W/m^2. C°
e	Wall thickness	m
S	Thermal exchange surface	m^2
T_1	Temperature of the initial body	C°
T_2	Temperature of the terminal body	C°
Qconv	heat transfer flow by convection	W
hc	Convection coefficient	W/m2. C°
Qray	Heat transfer flow by radiation	W
hr	radiation coefficient	W/m2. C°
R_{th}	Thermal resistance of the wall	m^2.C°/W
h_i	Internal surface heat transfer flow coefficient	W/m^2.C°
h_e	External surface heat transfer flow coefficient	W/m^2.C°
e_i	Thickness of the first layer of the wall	m
λ_i	thermal conductivity of the first layer of the wall	W/m.C°
K	overall heat transfer coefficients of the different components	W/ m^2C°
Rtot	Total thermal resistance	m^2C°/W
D_T	Heat loss by transmission	W
D_s	Surface losses through the common parts of the walls in contact with the exterior	W
D_{li}	Heat leakage through wall links	W
D_{sol}	Heat loss through the walls in contact with the soil	W
D_{lnc}	Heat loss through walls in contact with unheated rooms	W
A	Surface of the calculated panel	m²
T_i	Internal Temperature	C°
T_e	External Temperature	C°
Tau	temperature reduction coefficient	N/A

t_i	Internal temperature	C°
tn	Unheated volume temperature	C°
te	External Temperature	C°
de	Heat loss from the unheated room to the exterior	W/C°
ac	Heat gain from the various heated rooms to the unheated room	W/C°
Qv	The Fan volume flow	m³/h
Vh	The living volume	m³
Qvref	The reference extract of the fan volume flow	m³/h
Qs	Additional flow by fan infiltration	m³/h
Ev	wind exposure coefficient against the wall	N/A
P_P	Air infiltration from the wall	N/A
P_0	Surface air infiltration from the opening	N/A
Dref	Heat loss reference	W
APO	Opaque walls heat gains	W
AV	Glass wall heat gains	W
Al_s	The sensitive parts of the internal gains	W
Al_l	The latent parts of the internal gains	W
$AINF_s$	The sensitive parts of heat gain due to air infiltration	W
$AINF_l$	The latent parts of heat gain due to air infiltration	W
$APO_{ext}(t)$	Heat gain via external panels	W
$APO_{int}(t)$	Heat gain via internal panels	W
$APO_{soil}(t)$	Heat gain via panels in contact with the soil	W
$K_{été}$	The transmission coefficient in the straight part of the wall considered for summer	W/m².°C
S_{int}	The total interior surface of the wall considered for roofs in slope, whilst taking the horizontal projection of the surface	m²
$\Delta te(t)$	The equivalent difference of temperature at a given time t	C°
$\Delta tes(t)$	the temperature difference equivalent to time t considering that the wall is shadowed	C°
$C\Delta te$	a correction factor	C°
α	The wall absorption factor	N/A
$\Delta tem(t)$	the equivalent temperature difference at time t for the orientation of the panel in question	C°

$\mathbf{I_{t,b}}$	the total base radiation for the month with the latitude and orientation taken into consideration	W/m²
$\mathbf{I_{t,b(40)}}$	The total base radiation for the month of July with a latitude of 40° North and orientation considered	W/m²
$\mathbf{TS_e(t)}$	The dry exterior temperature at a given time t	C°
$\mathbf{C_{inc}}$	Correcting coefficient	N/A
$\mathbf{TS_{b,i}}$	The basic interior temperature	C°
$\mathbf{S_{gain}}$	The Surface of the heat gain area	m^2
$\mathbf{TS_m}$	Temperature of the air conditioned room	C°
AVT (t)	Represents the heat gains that are due to temperature gradients penetrating the glass panels	W
AVE (t)	Represents the heat gains that are due to solar rays penetrating the glass panels.	W
$\mathbf{S_{ouv}}$	The surface of the opening in the opaque wall in question	m^2
$\mathbf{SV_{ens}}$	The sun exposed glass surface at 15h (3P.M)	m^2
$\mathbf{I_t}$	The real total maximum radiation	W/m^2
$\mathbf{SV_{omb}}$	The shadowed glass surface	m^2
$\mathbf{I_d}$	The real diffused maximum radiation	W/m^2
FS	The Glazing solar factor	N/A
$\mathbf{N_{pvi}}$	The amortization coefficient relating to heat gains via sunshine penetrating glass panels at a time t	N/A
A_s	The sensitive part of the internal gains	W
A_l	The latent part of the internal gains	W
CS	The relative coefficient of simultaneity	N/A
N	The relative coefficient of amortization	N/A
$\mathbf{W_n}$	The nominal power of the fluorescent tube or bulb	N/A
$\mathbf{C_{me}}$	The increase coefficient for lighting gain	N/A
$\mathbf{C_{cr}}$	The residual heat percentage	%
$\mathbf{Qv_{inf}}$	Flow due to infiltration	m^3/h
$\mathbf{Qv_{fa}}$	Flow due to fresh air	m^3/h
$\mathbf{HS_{b,e}}$	The basic specific humidity of the exterior	$g_v/kg_{humid\ air}$
$\mathbf{HS_{b,i}}$	The basic specific humidity of the interior	$g_{vap}/kg_{humid\ air}$
$\mathbf{Qv0_{inf,i}}$	The infiltration flow corresponding to the considered opening i	m^3/h
$\mathbf{A_{réf,ph}}$	The reference heat gains via horizontal opaque panels	W

$\mathbf{A_{réf,pv}}$	The reference heat gains via vertical opaque panels	W
$\mathbf{A_{réf,pvi}}$	The reference heat gains via glass panels	W
a	Coefficient related to the nature and utility of the building in question	N/A
$\mathbf{\Delta TS_{réf,ph}}$	The temperature difference of the reference for the horizontal opaque panel	C°
c	Coefficient associated with the nature and functions of the climatic area	N/A
$\mathbf{\Delta TS_{réf,pv}}$	The temperature difference of the reference for the vertical opaque panel	C°
$\mathbf{AVE_{réf}}$	Reference gains via sunshine penetrating the glass panels	W
$\mathbf{AVT_{réf}}$	Reference gains via temperature gradient	W
SV	The total Glass surface	m^3
It	The total maximal radiation for the considered latitude and orientation	N/A
Id	The diffused maximal radiation	N/A
$\mathbf{FS_{réf}}$	The solar factor of the reference	N/A
$\mathbf{N_{PVI,\ réf}}$	The amortization coefficient relative to heat gains of the glass panels in the direction of the orientation in question	N/A
e'	A relative coefficient to the nature of the room	$W/m^2.°C$
$\dot{\mathbf{m}}$	mass flow rate	kg/s
Q	cooling load of the room	kW
Cp	specific heat capacity of the air	kJ/kg.K
$\dot{\mathbf{V}}$	volume flow rate	m^3/s
V	specific volume	m^3/kg
$\mathbf{C_o}$	Loss coefficient	N/A
Rho	The density of the air	kg/m3
ΔP_{coil}	Coil Pressure drop of the air handler unit	pa
$\mathbf{\Delta P_{AF}}$	Air filter pressure drop	pa
η	Efficiency	%

List of Tables

Table 1.1	HVAC Systems Classifications…….	4
Table 1.2	External climate condition of the city of Medea……...	6
Table 1.3	External climate condition of the hotel……..	6
Table 1.4	Ground floor specifications……...	13
Table 1.5	First floor specifications……..	14
Table 1.6	Second and third floor specifications (from F2-01 toF2-15)……..	15
Table 1.7	second and third floor specifications (from F2-16 toF2-29)……...	16
Table 1.8	Fourth floor specifications (from F4-01 toF4-15)……	17
Table 1.9	Fourth floor specifications (from F4-15 to F4-32)……..	18
Table 2.1	Interior/exterior surface heat transfer coefficients……	22
Table 2.2	Thermal resistance of each layer of the external wall……..	24
Table 2.3	Thermal resistance of each layer of the internal wall between rooms……	25
Table 2.4	Thermal resistance of the layers of the internal wall contacted with the air……	25
Table 2.5	Thermal resistance of the layers of the ceiling separating floors…….	26
Table 2.6	Thermal resistance of the layers of the Lower floor……	27
Table 2.7	Thermal resistance of the layers of the roof…….	27
Table 2.8	Coefficient K for variety of doors and windows…….	28
Table 2.9	Coefficient K of each building surface type…….	29
Table 2.10	Area B Dref coefficients (a, b, c, d and e)…….	34
Table 2.11	Unheated room's temperature on the ground floor……..	35
Table 2.12	Unheated room's temperature on the first floor……..	35
Table 2.13	Unheated room's temperature on the second floor…….	36
Table 2.14	Unheated room's temperature on the third floor…….	36
Table 2.15	Unheated room's temperature on the fourth floor……	36
Table 2.16	Calculation results of heat losses for the ground floor…….	40
Table 2.17	Calculation results of heat losses for the First floor……..	41
Table 2.18	Calculation results of heat losses for the second floor…….	42
Table 2.19	Calculation results of heat losses for the second floor…….	43
Table 2.20	Calculation results of heat losses for the third floor…….	44
Table 2.21	Calculation results of heat losses for the third floor…….	45
Table 2.22	Calculation results of heat losses for the fourth floor……..	46

Table 2.23	Calculation results of heat losses for the fourth floor…….	47
Table 2.24	Total thermal losses for the hotel…….	49
Table 3.1	Total base radiation $I_{t,b}$and $I_{d,b}$…….	53
Table 3.2	Lighting internal heat gains……...	57
Table 3.3	The heat gain of some electrical devices (1)……...	57
Table 3.4	The heat gain of some electrical devices (2)……	58
Table 3.5	Value of ΔTem (t) and ΔTes (t)……	64
Table 3.6	Total heat gains of the room(F2-20)……	68
Table 3.7	Calculation results of heat gains for the ground floor……	69
Table 3.8	Calculation results of heat gains for the first floor…….	70
Table 3.9	Calculation results of heat gains for the second floor…….	71
Table 3.10	Calculation results of heat gains for the third floor…….	72
Table 3.11	Calculation results of heat gains for the Fourth floor…….	73
Table 3.12	Total heat gains……	74
Table 4.1	Comparison table for the different types of HVAC systems……...	81
Table 4.2	Nominal Capacity of BLN and SLN versions of YLRA…….....	84
Table 4.3	Technical Data of BLN and SLN versions of YLRA…….....	84
Table 4.4	Technical specification, Belimo VAV controller…….....	90
Table 4.5	Quick selection PEGASUS LØV-S-K/L measurements…….....	90
Table 4.6	Adjustment and regulation range for VAV, air flow rate in m^3/h…….....	90
Table 4.7	Dimensions and weight for PEGASUS…….....	90
Table 5.1	Loss coefficient for the 90° elbow bend…….....	101
Table 5.2	loss coefficient for the Tee fitting……...	102
Table 5.3	Ground floor pressure drop results……..	103
Table 5.4	First floor pressure drop results…….....	105
Table 5.5	Second floor pressure drop results……...	107
Table 5.6	Third floor pressure drop results……..	108
Table 5.7	Fourth floor pressure drop results…….....	110
Table 5.8	Vertical duct pressure drop ……	111
Table 5.9	Total pressure drop ……...	111
Table 5.10	Air filters pressure drop…….....	112
Table 5.11	Pulley diameter and RPM……	114

List of figures

Figure 1.1	Basement plan……..	8
Figure 1.2	Ground floor plan…….	9
Figure 1.3	First floor plan……..	10
Figure 1.4	Second and third floor plan…….	11
Figure 1.5	Fourth floor plan……..	12
Figure 2.1	One-dimensional conduction heat transfer…….	20
Figure 2.2	One-dimensional heat transfer by conduction……...	21
Figure 2.3	Heat transfer via radiation……..	21
Figure 2.4	Sectional view of an exterior double partition wall……	24
Figure 2.5	Sectional view of an interior wall between rooms……...	24
Figure 2.6	Sectional view of an interior wall in contact with traffic……...	25
Figure 2.7	Sectional view of a floor between floors…….	26
Figure 2.8	Sectional view of a Floor on ground…….....	26
Figure 2.9	Sectional view of a roof (roof)……	27
Figure 2.10	Example of calculating a room (heat loss)……	37
Figure 3.1	Example of calculating a room (heat gain)…….....	63
Figure 4.1	Split system air conditioning system……...	76
Figure 4.2	Heating unit system…….	76
Figure 4.3	Hybrid split system infused with solar energy…….....	77
Figure 4.4	Mitsubishi model duct free mini split system……	77
Figure 4.5	Packaged heating and cooling unit system with return duct…….....	78
Figure 4.6	Example of a Partial plaster fake ceiling for duct use…….....	82
Figure 4.7	The YORK YLRA air cooled heat pump scroll compressor…….....	82
Figure 4.8	Schematics of the YLRA 0200 to 0280 models…….....	85
Figure 4.9	Schematics of the YLRA 0300 and 0330 models…….....	86
Figure 4.10	Typical square wall air diffuser…….....	87
Figure 4.11	Typical variable air volume system box…….....	88
Figure 4.12	The PEGASUS-S (left) and the PEGASUS-B (right) VAV diffuser…….....	89
Figure 4.13	Dimensioned sketch, of the PEGASUS-S…….....	91
Figure 4.14	Example of the flow pattern of the PEGASUS…….....	91
Figure 4.15	Diagram of the throw length of the PEGASUS…….....	92

Figure 4.16	Example of velocity distribution……..……..	92
Figure 4.17	Mounting of the PEGASUS-S……..……	93
Figure 4.18	Example of duct installation of the PEGASUS-S…….…….	93
Figure 5.1	Ground floor ducts location……	96
Figure 5.2	First floor ducts location…….	97
Figure 5.3	Second and third floor ducts location……	98
Figure 5.4	Fourth floor ducts location…….	99
Figure 5.5	Friction Chart for Round Duct (ρ = 1.20 kg/m3 and ε = 0.09 mm)……..	100
Figure 5.6	Duct tee pressure loss…….……..	102
Figure 5.7	Ground floor duct and fitting labels…….....	103
Figure 5.8	First floor duct and fitting labels……	105
Figure 5.9	Second floor duct and fitting labels……	106
Figure 5.10	Third floor duct and fitting labels……..	108
Figure 5.11	Fourth floor duct and fitting labels……	109
Figure 5.12	Wheel Type H – YFFCDH……	114

General Introduction

Introduction

Never has there been a time where the frail human being found themselves independent from the need of thermal comfort, even before their birth, this intelligent living creature finds themselves covered, enveloped and coated in their mother's womb, once born embedded in the warmth of a blanket, then tucked in the smoothening warmth of clothes for as long as they walk on earth.

Yet, climate is seldom in favor of the individual, however, as humans have learned ever since the beginning of time, the need for thermal comfort is as crucial as the need for food and water, pivotal to insure productivity, well being and ensured health.

According to **Marcus Vitruvius Pollio** in his book entitled "**de architectura**" written in 15 BC. "A building architecture must be supported by three basic qualities: **quality firmitas**, **quality utiiitas** and **venusitas quality**'', which in turn describe strength, functionality and beauty respectively, and as thermal comfort or temperature comfort falls into the category of functionality its consideration is of key importance to not only the quality of the building or establishment in question, but to the environmental comfort of the occupants of the aforementioned edifice.

This masters dissertation project will discuss the study of the Heating, ventilation and air conditioning preparation for a hotel in the province of Medea, consisting of a variety of rooms stretching from offices, conference rooms and meeting rooms to hotel rooms, cafeteria, restaurant space and more, The HVAC system of choice according to this thesis will be decided depending on the results that will be casted along the study.

This academic work will be divided into five chapters which each tackling:

- **Chapter I** being a general introduction of thermal comfort as well as the knowledge necessary for the work plan.
- **Chapter II** tackling the calculation of heat loss in the establishment in question.
- **Chapter III** addresses the calculations of heat gain in the building at hand.
- **Chapter IV** dedicates itself to the choice of the HVAC system that will be used for the heating, ventilating and air conditioning task.
- **Chapter V** sets about calculating the duct work as well as the flow and pressure drop of the applied system.

Chapter I

Presentation and Project description

1.1 Introduction:

Ever since the dawn of man, the need for thermal comfort has always been as crucial as the need for food and water, and as time passes, exponential advancement of technology knows neither boundaries nor limits, including the endless research for optimal thermal comfort in buildings.

Thermal comfort is a state of mind where the body expresses satisfaction with the thermal environment, in the individual's presence, they would feel neither too cold or two hot, an equilibrium of temperature allowing them to work in a comfortable state as it is important to health and well being as well as productivity and performance, where too much exposure to heat causes tiredness while too much absence of heat leads to restlessness distraction and in extreme cases can even have serious or mild health complications.

1.2 The Importance of Air Conditioning:

As a person spends most of his time indoors, the main purpose of an HVAC (Heating, ventilating and air conditioning) system is to insure thermal control in buildings, depending on the season in question, a heating system insures the limitation of temperature to a certain value in winter, minimizing heat loss through transmission and air renewal.

In summer, air condition systems set to not only refresh and renew air but also –through chilling air- insure a decrease in residual heat, countering heat gain, as well as providing a more comfortable environment.

The utility of HVAC systems is bound to a vast majority of factors, as it is in charge of controlling a multitude of factors concerning the air residing in a building or an individual room stretching from temperature and humidity to air purity, circulation and filtration.

The previously mentioned characteristics concerning air vary depending on the type and usage of the building, whether it is meant for domestic or industrial purposes.

1.3 Thermal Comfort:

Heating, Ventilation and Air Conditioning system's main goal is to insure thermal comfort as well as the well being of individual in a given building or habitat, based on basic as well as advanced knowledge of the physiological needs of human beings in an artificial thermal environment, therefore, knowledge about physiology balance, energetic and thermal needs, human metabolism as well as heat exchange of living and artificial organisms is a must to reach the purposed goal. [1]

For a thermal environment to be deemed comfortable, a set of conditions must be withheld in order for a human organism to find a mental state of satisfaction, also defined as the thermal comfort zone; where an individual finds thermal balance whatever the atmospheric state.

The thermal comfort zone is defined by a particular set of conditions and parameters, putting aside clothing and wear. [1]

We distinguish 4 main parameters:

- The air temperature.
- The average temperature of the concealments (walls, ceiling)
- The air circulation
- The air humidity

1.4 HVAC Systems Classifications:

HVAC systems are classified in function of thermodynamic number of functions.

Setup designation	Non-thermodynamic functions	Thermodynamic functions	
		Number of functions	Type of functions
Ventilation setup	CI and F CI and F CI and F CI and F CI and F	None or one at most	CH R H d
Cooling or partial air conditioning setup	CI and F CI and F CI and F CI and F CI and F CI and F	Two	CH and R CH and H CH and D R and H R and D H and D
	CI and F CI and F CI and F CI and F	Three	CH, R and H CH, R and D R, H and D CH, H and D
Cooling or total air conditioning setup	CI and F	Four	CH, R, H and D
CH: Heating; **R**: Cooling; **H**: Humidification; **D**: Dehumidification. **F**: Filtration ; **Cl**: Air circulation			

Table 1.1: HVAC Systems Classifications.

1.5 Project Presentation:

The theme of our thesis is the study of a heating, ventilation and air conditioning system provided for a new hotel consisting of a basement parking lot, ground floor reception, and offices in the second floor, three residual floors, and an accessible rooftop for open space leisure, whereas the parking lot and the accessible rooftop are unheated.

This project is situated in the city of Medea, chief town of the state of Medea, Algeria.

1.5.1 Project goal:

The goal of our thesis is to define the calorific needs that needs compensating during winter period, as well as the thermal energy that needs extraction during summer season for each room of the hotel in addition to choosing the adequate equipments and tools to insure the previously mentioned needs to be met, through precise regulation of system setups as well as pinpoint dimensioning of heat transfer liquid conducts in order to insure a thermal comfort zone accurately regarded from an acoustic and thermal perspective.

1.5.2 Geographical localization:

As previously mentioned, this project is situated in the city of Medea, situated in the climatic Area B, where climatic conditions are as presented as follows:

1.5.2.1 External Climate conditions:

				WINTER SEASON		SUMMER SEASON	
Area	Altitude (m)	Latitude (N°)	Longitude (E°)	Average Temperature (C°)	Relative Humidity (φ %)	Average Temperature (C°)	Relative Humidity (φ %)
B'	1036	36.2637	2.7588	1	90	32	53

Table 1.2: External climate condition of the city of Medea. [D1]

1.5.2.2 Internal Climate Conditions:

Designation	WINTER SEASON		SUMMER SEASON	
	Ti°(C°)	Φi (%)	Ti°(C°)	Φi (%)
Reception	20	50	25	50
Diner (ground floor)	20	50	25	50
Cafeteria (ground floor)	20	50	25	50
Hall	20	50	25	50
Conference room	20	50	25	50
Offices	20	50	25	50
Hotel rooms	20	50	25	50
Suits	20	50	25	50

Table 1.3: External climate condition of the hotel.

1.5.3 Architectural Plan:

Next, we present our plans and schematics for our project's building being the Hotel consisting of a basement parking lot, ground floor reception, and offices in the second floor, three residual floors, and an accessible rooftop for open space leisure, review figures (1-1, 1-2, 1-3, 1-4 and 1-5).

It is worth mentioning that the parking lot as well as the accessible rooftop will be of little if not no intervention in our study as they are both unheated floors, however, centralization of our HVAC systems will be maintained and placed in the basement.

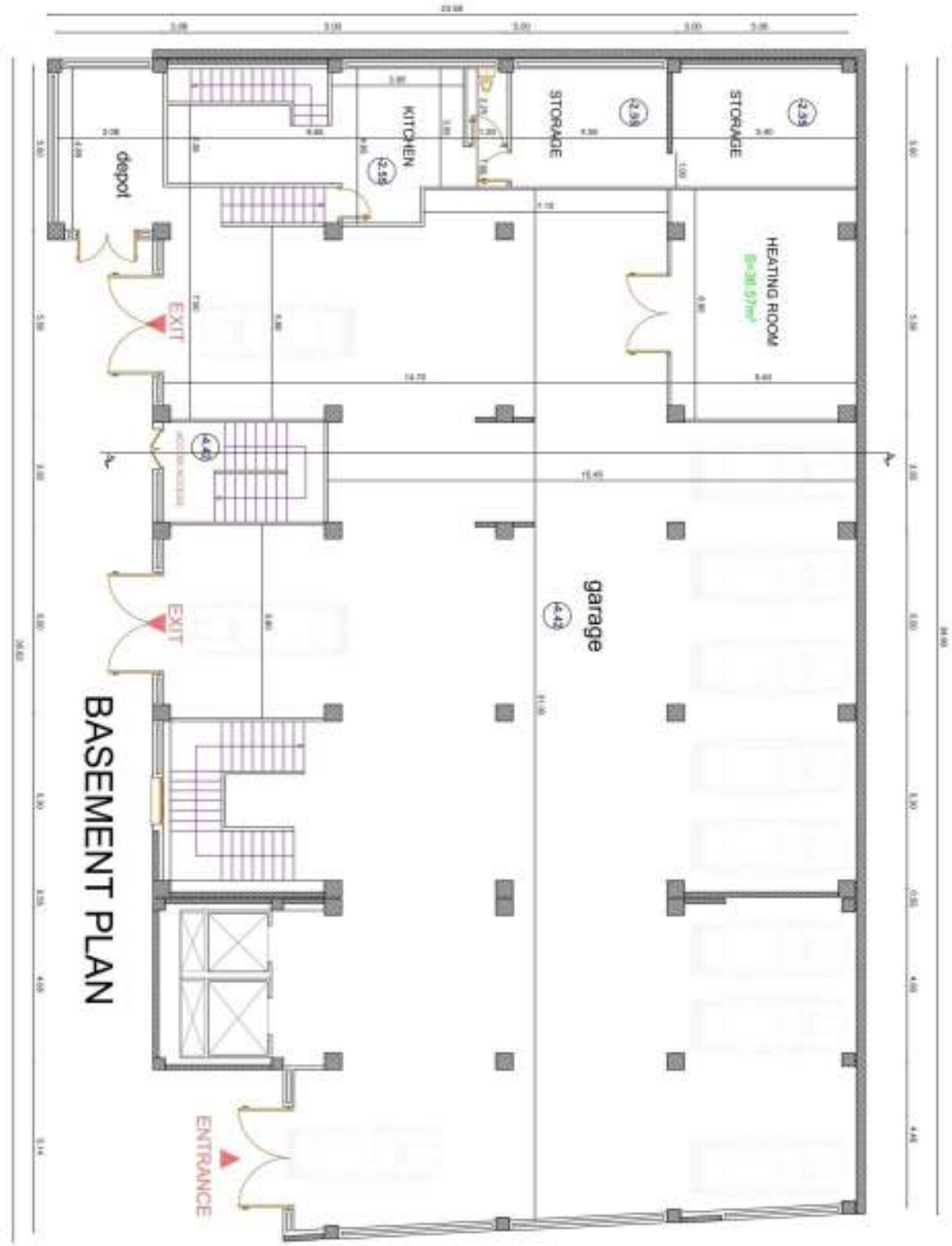


Figure 1.1: Basement plan.

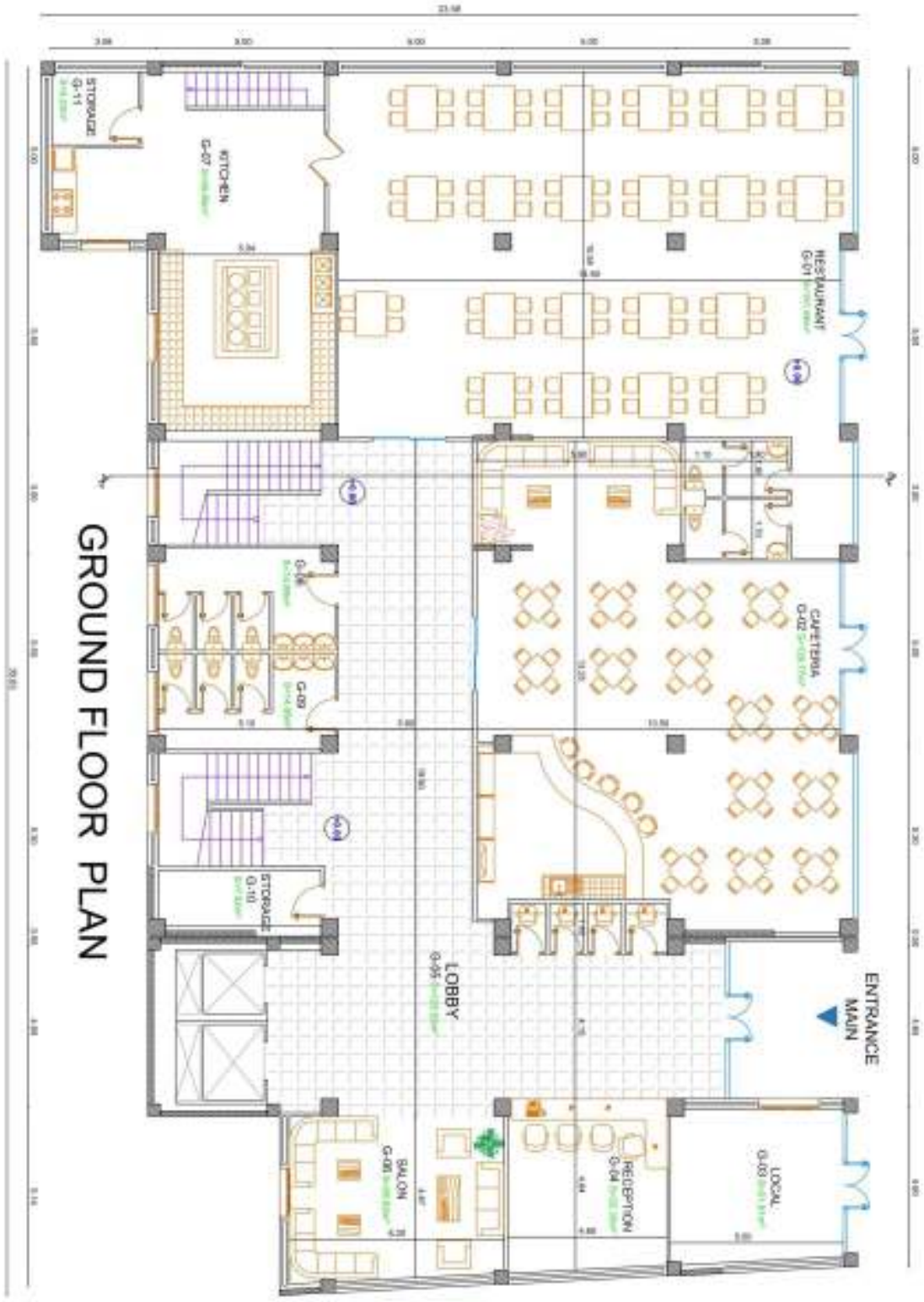


Figure 1.2 : Ground floor plan.

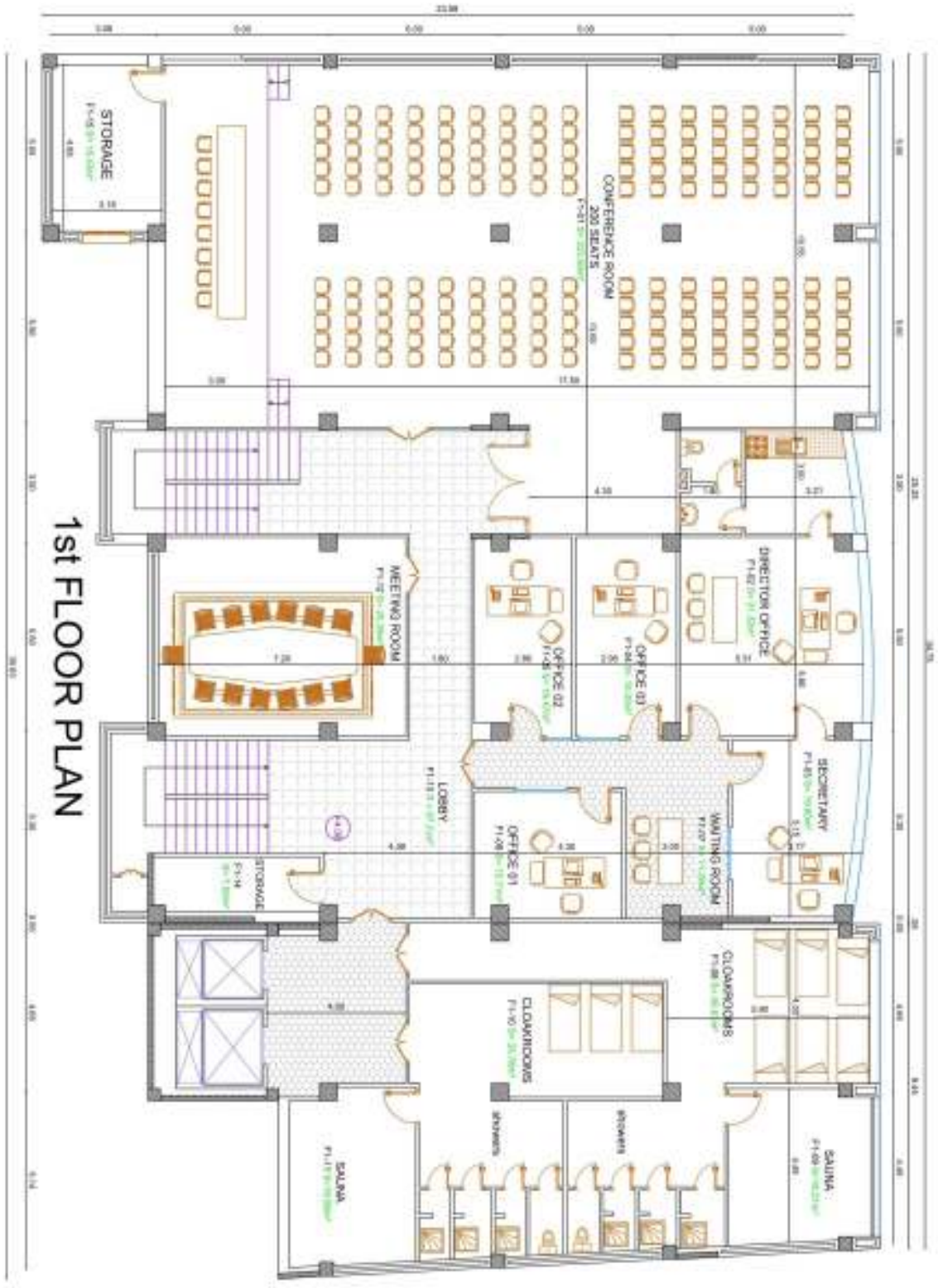


Figure 1.3: First floor plan.



Figure 1.4: Second and third floor plan.



Figure 1.5: Fourth floor plan.

1.5.4 Project specifications:

According to the detailed plans that have been presented in the previous figures (1-1, 1-2, 1-3, 1-4 and 1-5), we suggest the following tables of specifications for each floor of our project's building:

1.5.4.1 Ground floor specifications:

Room Designation	Sign	Number of occupants	Height (m)	Surface (m^2)	Temperature (Win/Sum)	Φi (%)
Restaurant	G-01	60	4.08	161.66	20/25	50
Cafeteria	G-02	45	4.08	129.77	20/25	50
Local	G-03	5	4.08	21.81	20/25	50
Reception	G-04	5	4.08	22.33	20/25	50
Lobby	G-05	0	4.08	122.52	20/25	50
Salon	G-06	8	4.08	28.82	20/25	50
Kitchen	G-07	5	4.08	55.85	20/25	50
Bathroom 1	G-08	2	4.08	14.89	20/25	50
Bathroom 2	G-09	2	4.08	14.89	20/25	50
Main duct opening	G-10	0	4.08	7.52	20/25	50
Storage 2	G-11	0	4.08	5.23	20/25	50

Table 1.4: Ground floor specifications.

1.5.4.2 First floor specifications:

Room Designation	Sign	Number of occupants	Height (m)	Surface (m^2)	Temperature (Win/Sum)	Φi (%)
Conference room	F1-01	200	7.82	223.80	20/25	50
Director office	F1-02	5	7.82	31.32	20/25	50
Secretary	F1-03	2	7.82	19.80	20/25	50
Office 1	F1-04	3	7.82	15.20	20/25	50
Office 2	F1-05	3	7.82	16.47	20/25	50
Office 3	F1-06	3	7.82	15.71	20/25	50
Waiting room	F1-07	3	7.82	11.20	20/25	50
Cloakrooms	F1-08	6	7.82	45.67	20/25	50
Sauna	F1-09	4	7.82	18.21	40/40	90
Cloakrooms	F1-10	3	7.82	33.78	20/25	50
Sauna	F1-11	4	7.82	18.56	40/40	90
Meeting room	F1-12	15	7.82	25.39	20/25	50
Lobby	F1-13	0	7.82	57.20	20/25	50
Main duct opening	F1-14	0	7.82	7.52	20/25	50
Storage	F1-15	0	7.82	15.43	20/25	50

Table 1.5: First floor specifications.

1.5.4.3 Second and third floor specifications:

Room Designation	Sign	Number of occupants	Height (m)	Surface (m^2)	Temperature (Win/Sum)	Φi (%)
Balcony	F2-01	0	11.22	9.21	20/25	50
Balcony	F2-02	0	11.22	9.50	20/25	50
Balcony	F2-03	0	11.22	8.26	20/25	50
Balcony	F2-04	0	11.22	7.87	20/25	50
Room 08	F2-05	1	11.22	28.15	20/25	50
Room 07	F2-06	1	11.22	30.50	20/25	50
Room 06	F2-07	1	11.22	18.70	20/25	50
Room 05	F2-08	1	11.22	32.00	20/25	50
Room 04	F2-09	1	11.22	31.14	20/25	50
Room 03	F2-10	2	11.22	25.52	20/25	50
Room 02	F2-11	2	11.22	26.00	20/25	50
Bathroom	F2-12	1	11.22	5.31	20/25	50
Bathroom	F2-13	1	11.22	6.52	20/25	50
Bathroom	F2-14	1	11.22	4.37	20/25	50
Bathroom	F2-15	1	11.22	6.53	20/25	50

Table 1.6: Second and third floor specifications (from F2-01 toF2-15).

Room Designation	Sign	Number of occupants	Height (m)	Surface (m^2)	Temperature (Win/Sum)	Φi (%)
Bathroom	F2-16	1	11.22	4.43	20/25	50
Bathroom	F2-17	1	11.22	6.90	20/25	50
Bathroom	F2-18	1	11.22	6.46	20/25	50
Bathroom	F2-19	3	11.22	9.10	20/25	50
Suite	F2-20	5	11.22	91.11	20/25	50
Bathroom	F2-21	1	11.22	6.90	20/25	50
Room 01	F2-22	2	11.22	19.96	20/25	50
Lobby	F2-23	0	11.22	170.33	20/25	50
Bathroom	F2-24	1	11.22	7.52	20/25	50
Room 09	F2-25	3	11.22	29.33	20/25	50
Main duct opening	F2-26	0	11.22	7.52	20/25	50
Balcony	F2-27	0	11.22	13.50	20/25	50
Balcony	F2-28	0	11.22	8.56	20/25	50
Balcony	F2-29	0	11.22	5.32	20/25	50

Table 1.7: second and third floor specifications (from F2-16 toF2-29).

1.5.4.4 Fourth floor specifications:

Room Designation	Sign	Number of occupants	Height (m)	Surface (m^2)	Temperature (Win/Sum)	Φi (%)
Balcony	F4-01	0	14.62	9.21	20/25	50
Balcony	F4-02	0	14.62	9.50	20/25	50
Balcony	F4-03	0	14.62	8.26	20/25	50
Balcony	F4-04	0	14.62	7.87	20/25	50
Room 08	F4-05	1	14.62	28.15	20/25	50
Room 07	F4-06	1	14.62	30.50	20/25	50
Room 06	F4-07	1	14.62	18.70	20/25	50
Room 05	F4-08	1	14.62	32.00	20/25	50
Room 04	F4-09	1	14.62	31.14	20/25	50
Room 03	F4-10	2	14.62	25.52	20/25	50
Room 02	F4-11	2	14.62	26.00	20/25	50
Bathroom	F4-12	1	14.62	5.31	20/25	50
Bathroom	F4-13	1	14.62	6.52	20/25	50
Bathroom	F4-14	1	14.62	4.37	20/25	50
Bathroom	F4-15	1	14.62	6.53	20/25	50

Table 1.8: Fourth floor specifications (from F4-01 toF4-15).

Room Designation	Sign	Number of occupants	Height (m)	Surface (m^2)	Temperature (Win/Sum)	Φi (%)
Bathroom	F4-16	1	14.62	4.43	20/25	50
Bathroom	F4-17	1	14.62	6.90	20/25	50
Bathroom	F4-18	1	14.62	6.46	20/25	50
Bathroom	F4-19	3	14.62	9.10	20/25	50
Suite	F4-20	5	14.62	91.11	20/25	50
Bathroom	F4-21	1	14.62	6.90	20/25	50
Room 01	F4-22	2	14.62	19.96	20/25	50
Lobby	F4-23	0	14.62	170.33	20/25	50
Bathroom	F4-24	1	14.62	7.52	20/25	50
Room 09	F4-25	3	14.62	29.33	20/25	50
Main duct opening	F4-26	0	14.62	7.52	20/25	50
Balcony	F4-27	0	14.62	13.50	20/25	50
Balcony	F4-28	0	14.62	8.56	20/25	50
Balcony	F4-29	0	14.62	5.32	20/25	50
Balcony	F4-30	0	14.62	8.54	20/25	50
Balcony	F4-31	0	14.62	8.25	20/25	50
Balcony	F4-32	0	14.62	9.03	20/25	50

Table 1.9: Fourth floor specifications (from F4-15 to F4-32).

Chapter II

Thermal calculation and heat loss calculation

2.1 Introduction:

To create an HVAC system in a room, it is necessary to establish a thermal balance, which will serve as a basis for the calculation for the entire setup.

The thermal balance makes it possible to distinguish the heat gains and the heat losses for a room; it has a direct influence on the dimensioning of the HVAC system's networks. It is therefore necessary to know with utmost precision the factors that are of impact on the latter mentioned system before actually establishing it.

Therefore, we distinguish three main factors:

- **Geometrical factors:** such as room dimensions stretching from surfaces to volumes.
- **Geographical factors:** such as the latitude, altitude and orientation of the building or particular room in question.
- **Thermal factors:** from which we can mention:
    - Occupants: time of occupation, number of occupants and performed activities during occupation.
    - Lighting: lighting type and power.
    - Construction materials.
    - Machines: engines, power supplies…

2.2 Heat Transfer modes :

In the study of thermodynamics, there are three different types of heat transfer modes which are known to man today; conduction, convection and radiation.

2.2.1 Conduction heat transfer:

Conduction heat transfer is the transfer of heat by means of molecular excitement within a material without bulk motion of the matter. Conduction heat transfer in gases and liquids is due to the collisions and diffusion of the molecules during their random motion. On the other hand, heat transfer in solids is due to the combination of lattice vibrations of the molecules and the energy transport by free electrons. [2]

To examine conduction heat transfer, it is necessary to relate the heat transfer to mechanical, thermal, or geometrical properties. Consider steady-state heat transfer through the wall of an aorta with thickness Δe where the wall inside the aorta is at higher temperature (T_1) compared with the outside wall (T_2). Heat transfer, Qcond (W), is in the direction of x and perpendicular to plane of temperature difference. [2]

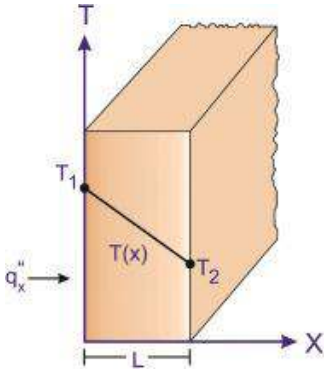


Figure 2.1: One-dimensional conduction heat transfer. [2]

As conduction heat transfer is a ''high to low'' temperature function along the wall, it is described by the following formula:

$$Qcond = \left(\lambda \frac{}{e}\right) \times S \times (T1 - T2) \qquad \text{II.1}$$

Whereas:

Q_{cond}: Heat transfer flow by conduction (W).

λ: Coefficient of thermal conductivity (W/m^2. C°).

e: Wall thickness (m).

S: Surface where the heat exchange phenomenon occurs. (m^2)

(T1-T2): The difference in temperature between the two objects from higher temperature to lower temperature respectively.

2.2.2 Convection heat transfer:

Convection is a transfer mode, which occurs by an exchange of energy between a body and a fluid. According to the kind of fluid flow, convection is termed free or forced. [2]

When the fluid currents are caused by temperature differences, the convection is called natural or free. If the fluid motion is caused by an external action, such as pump or a fan, it is termed forced convection. [2]

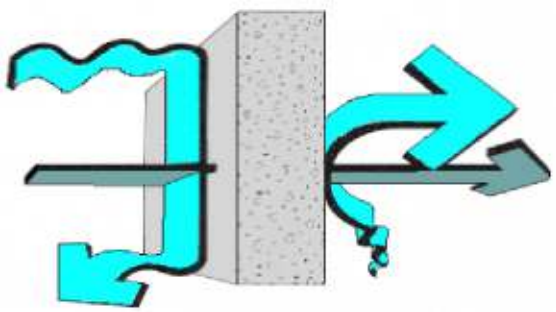


Figure 2.2: One-dimensional heat transfer by conduction. [2]

Heat transfer flow by convection is expressed in the following formula:

Qconv= hc. S. (T1-T2) II.2

Whereas:

Qconv: heat transfer flow by convection (W).

hc: Convection coefficient (W/m2. C°).

2.2.3 Radiation heat transfer:

Radiation heat transfer is the energy that is emitted by matter in the form of photons or electromagnetic waves. Radiation can be important even in situations in which there is an intervening medium. An example is the heat transfer that take place between a living entity with its surrounding.

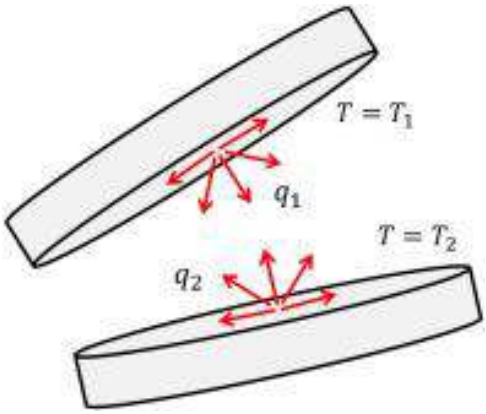


Figure 2.3: Heat transfer via radiation. [2]

Heat transfer flow by radiation varies in composure; however, we will consider the following equation for the sake of exemplification:

Qray= hr. S. (t1-t2) II.3

Whereas:

Qray : Heat transfer flow by radiation (W).

hr: radiation coefficient (W/m2. C°).

2.3 Interior/exterior surface heat transfer coefficients:

In heat transfer transmitted via convection or radiation, coefficients are generally assembled and considered as the one and same, depending on the direction and aim of the flow, this coefficient can vary also depending on the study being either interior ($h_{i)}$ or exterior ($h_{e)}$ as presented in the following table:

Direction	h_i(w/m2. ºC)		h_e(w/m2. ºC)	
Horizontal	9.09		16.66	
Vertical	Ascending	Descending	Ascending	Descending
	11.11	5.88	20	20

Table 2.1: Interior/exterior surface heat transfer coefficients. [D1]

2.4 Calculating overall heat transfer coefficients K:

Another important coefficient that needs to be taken into consideration is the overall heat transfer coefficient (K), which is a coefficient exclusive to walls comporting one or multiple layers and which is also expressed in the following manner by the following mathematic expression where each equation here on is taken from the RTD C3-4 [D1]:

$$K = \frac{1}{Rth} = \frac{1}{[\left(\frac{1}{hi}\right)+\Sigma\left(\frac{ei}{\lambda i}\right)+\left(\frac{1}{he}\right)]} \quad \text{II.4}$$

Whereas:

R_{th}: Thermal resistance of the wall (m^2.C°/w).

h_i: Internal surface heat transfer flow coefficient (W/m^2.C°).

h_e: External surface heat transfer flow coefficient (W/m^2.C°).

e_i: Thickness of the first layer of the wall (m).

λ_i : thermal conductivity of the first layer of the wall(W/m.C°).

2.4.1 Calculating overall heat transfer coefficients of the different components of the project:

Each time heat flow -which is a form of energy- clashes with a surface, it's flow is reduced depending on the characteristics of this said surface due to thermal resistance, in our project's case as well as most of buildings, this surface can be defined as a ceiling, floor, interior or exterior wall, doors, or windows.

However, each type of surface with thermal energy contact has its own specific thermal resistance, which can be calculated through the commune relation:

$$K = \frac{1}{Rtot} \qquad \text{II.5}$$

Where Rtot itself is identified as:

$$R_{tot} = R_i + \sum R + R_e \qquad \text{II.6}$$

Therefore:

$$K = \frac{1}{Ri + \sum R + Re} \qquad \text{II.7}$$

Whereas:

$$Ri = \frac{1}{hi} R = \frac{e}{\lambda} Re = \frac{1}{he} \qquad \text{II.8}$$

2.4.2 Building components:

According to our project data at hand, the components of our building as well as their thermal resistance categorized by overall surface would be as follows:

A. External walls:

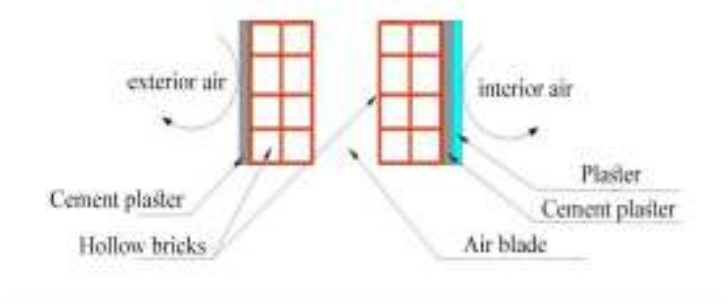


Figure 2.4: Sectional view of an exterior double partition wall.

Designation	e	λ	R
	m	W/m.C°	$m^2C°/W$
Plaster	0,005	0,35	0,014
Cement plaster	0,015	1,4	0,011
Hollow bricks	0,1	0,48	0,21
Air blade	0,1	0,625	0,16
Hollow bricks	0,1	0,48	0,21
Cement plaster	0,015	01,4	0,011
			∑R= 0,616

Table 2.2: Thermal resistance of each layer of the external wall.

B. Internal walls in between rooms:

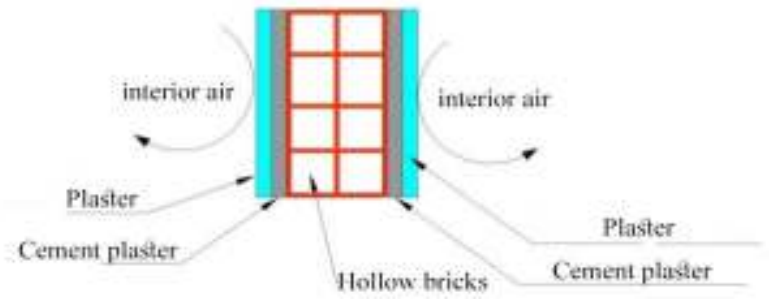


Figure 2.5: Sectional view of an interior wall between rooms.

Designation	e	λ	R
	m	W/m.C°	$m^2C°/W$
Plaster	0,005	0,35	0,014
Cement plaster	0,015	1,4	0,011
Hollow bricks	0,1	0,48	0,21
Cement plaster	0,015	1,4	0,011
Plaster	0,005	0,35	0,014
			∑ Rtot= 0,260

Table 2.3: Thermal resistance of each layer of the internal wall between rooms.

C. Internal walls in direct contact with air flow:

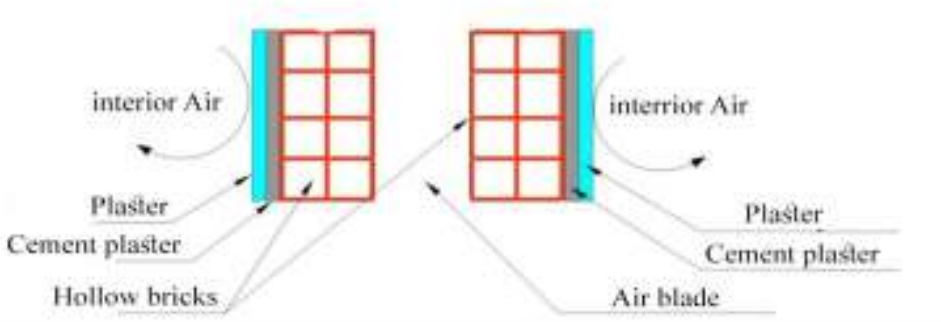


Figure 2.6: Sectional view of an interior wall in contact with traffic.

Designation	e	λ	R
	m	W/m.C°	$m^2C°/W$
Plaster	0,005	0,35	0,014
Cement plaster	0,015	1,4	0,011
Hollow bricks	0,1	0,48	0,21
Air blade	0,1	0,625	0,16
Hollow bricks	0,1	0,48	0,21
Cement plaster	0,015	01,4	0,011
plaster	0,005	0,35	0,014
			∑ Rtot=0,360

Table 2.4: Thermal resistance the internal wall layers contacted with the air. [D1]

D. Ceiling separating floors:

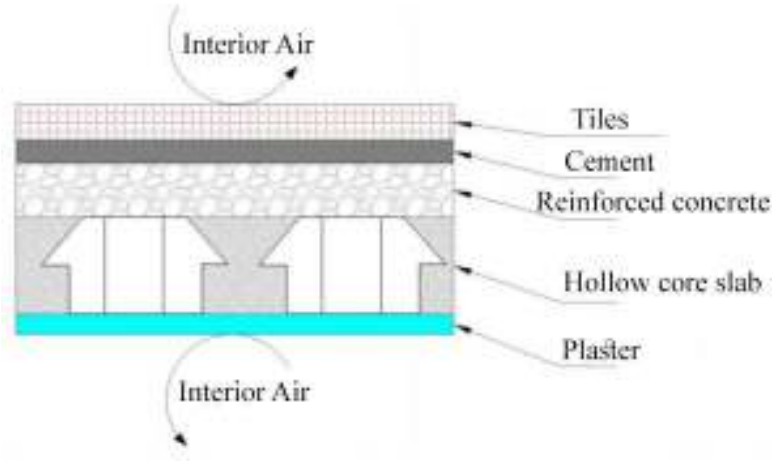


Figure 2.7: Sectional view of a floor between floors.

Designation	e	λ	R
	m	W/m.C°	$m^2C°/W$
Tiles	0,025	0,55	0,045
Cement	0,025	1,4	0,018
Reinforced concrete	0,1	1,75	0,057
Hollow core slab	0,1	0,98	0,102
Plaster	0,005	0,35	0,014
			∑ Rtot=0,236

Table 2.5: Thermal resistance of the layers of the ceiling separating floors. [D1]

E. Lower floor:

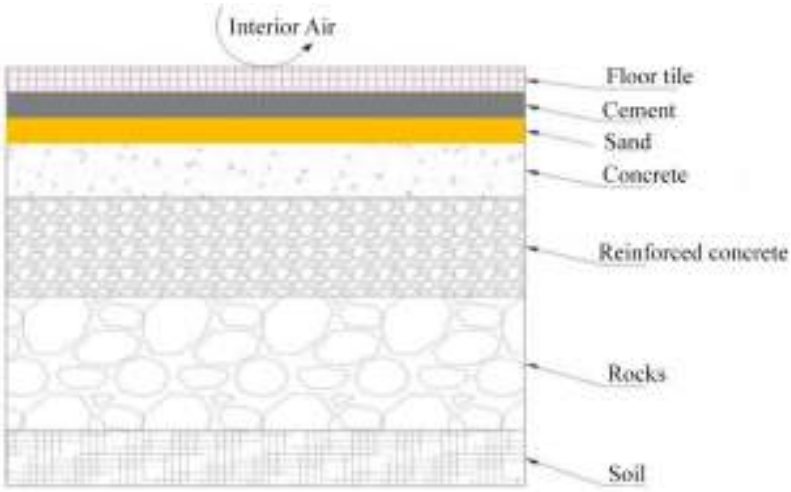


Figure 2.8: Sectional view of a Floor on ground.

Designation	e	λ	R
	m	W/m.C°	$m^2C°/W$
Floor tile	0,025	0,55	0,045
Cement	0,025	1,4	0,018
Sand	0,025	0,67	0,037
Concrete	0,075	1,4	0,053
Reinforced concrete	0,15	1,75	0,086
Rocks	0,33	2,67	0,12
Soil	0,17	2,43	0,07
			∑ Rtot=0,429

Table 2.6: Thermal resistance of the layers of the Lower floor. [D1]

F. Closure ceiling:

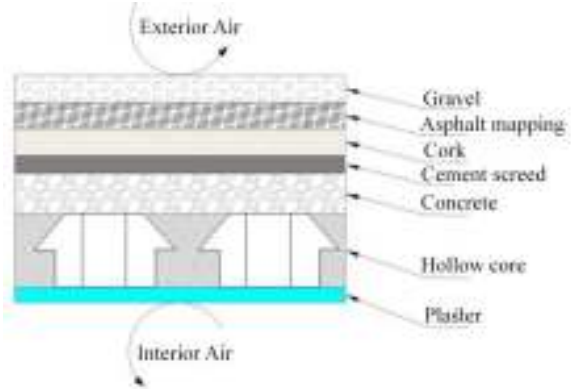


Figure 2.9: Sectional view of a roof (roof).

Designation	e	λ	R
	m	W/m.C°	$m^2C°/W$
Gravel	0,02	0,95	0,021
Asphalt mapping	0,03	0,2	0,15
Cork	0,02	0,06	0,33
Cement screed	0,01	1,63	0,006
Concrete	0,2	1,75	0,11
Hollow core	0,1	0,98	0,10
Plaster	0,01	0,35	0,057
			∑ Rtot= 0,774

Table 2.7: Thermal resistance of the layers of the roof. [D1]

Doors and Windows:

As for the carpentry elements of the project, the following coefficients are preset and therefore considered as is in the following table:

Designation		K(W/m²C°)
Doors	Wood	3.50
	Metal	5.80
	30%glazing	4.00
	30-60%glazing	4.70
	Metal or glass	5.80
Windows	Slippery glass	4.50
	Glass	5.00

Table 2.8: Coefficient K for variety of doors and windows. [D1]

2.4.3 Coefficient K calculation example:

For the sake of exemplification, we will demonstrate the method of calculating the coefficient K considering the previous formulae on our external wall of the building.

The coefficient K for our external walls would be expressed by the following development:

Ri= 0,11 m^2 °C/ W Re = 0,06 m^2 °C/ W

$$K = \frac{1}{0.11 + 0.616 + 0.06}$$

K= 1,3 (W/ m^2°C)

2.4.4 Final results of the calculated coefficient K:

After calculating the remaining coefficients K of the previously mentioned surfaces of our structure (A, B, C, D, E and F), we conclude to the following table containing the values of the coefficient K for each surface previously mentioned:

Destination	Coefficient K ((W/ $m^2C°$)
A	1.30
B	2.08
C	1.18
D	1.40
E	1.30
F	1.09

Table 2.9: Coefficient K of each building surface type.

2.5 Calculating heat balance in winter season:

In order to calculate the heat balance of a building, we need to establish a calculation that allows us to determine the calorific power necessary to maintain a stable temperature, to achieve such; we have chosen The Regulatory Technical Document C3-2 (RTD) also known in French as "Document technique règlementaire'' as a basis which will allow us through the use of its standards to calculate both heat loss and gain with great precision.

2.5.1 Presentation of the Regulatory Technical Document:

2.5.1.1 Objective of the RTD C3-2:

The Regulatory Technical Document's purpose is to establish a set of methods to insure thermal evaluations for rooms in the building, primarily those of calorific needs in winter season, the document allows:

- Determining heat loss for buildings or particular rooms.
- Verifying the compliance of buildings with thermal regulations.

- Deciding the dimensions of the heating system networks of buildings.
- The thermal conception of establishments.

2.5.1.2 Areas of usage:

The methods of finding the amount of heat gain and loss by the RTD are applicable on buildings with:

- ✓ Residential use.
- ✓ Accommodation such as dormitories, collective rooms or as in our project's case a hotel.
- ✓ Offices.
- ✓ Teaching such as classrooms, conference rooms and meeting rooms.
- ✓ Shops such as grocery stores , malls and shopping centers.
- ✓ Gastronomy as in restaurants and cafés.

2.5.1.3 Types of heat loss:

There exist two types of heat loss:

- Heat loss by transmission through the building envelopment such as walls, windows, doors and roof.
- Heat loss by air renewal which as the name suggests heat particles are moved through air traffic, causing heat loss.

2.5.1.4 Calculations

A. Calculating heat loss by transmission:

In order to calculate heat loss by transmission, we proceed with the following formula:

$$D_T = D_s + D_{li} + D_{sol} + D_{inc} \quad \text{II.9}$$

Whereas the formula consists of:

D_T: Heat loss by transmission.

D_s: Surface losses through the common parts of the walls in contact with the exterior.

D_{li}: Heat loss through wall links (wall leakage).

D_{sol}: Heat loss through the walls in contact with the soil**.**

D_{lnc}: Heat loss through walls in contact with unheated rooms.

A.1 Heat loss through the walls in contact with the exterior D_s:

The surface losses by transmission through a wall separating two different atmospheres are given by the following formula:

$$D_s = K \times A \times (T_i - T_e) \quad \text{II.10}$$

Whereas the formula contains the following:

K: The overall heat transfer coefficient measured in ($W/m^2.C°$).

A: Surface of the element in question (m^2).

T_i: Internal Temperature (C°).

T_e: External Temperature (C°).

A.2 Heat loss through wall links$D_{li:}$

The losses through links, also known as losses through thermal bridges, are given by the following formula:

$$D_{li} = K_i \times A \times (T_i - T_e) \quad \text{II.11}$$

Important Notice: In order to simplify calculations, it is commonly known to consider:

$$D_{li} = 0.2 \times D_s \quad \text{II.12}$$

A.3 Heat loss through the walls in contact with the soil D_{sol}:

In this manner of loss, three types of walls in contact with the soil can be distinguished, the upper interior surface of the room also known as ceiling, the regular envelopment walls and the lower base interior surface which can be translated into the following formula:

$$D_{sol} = K_S \times A \times (T_i - T_e) \quad \text{II.13}$$

A.4 Heat loss through walls in contact with unheated rooms D_{Inc}:

The heat loss provoked through walls that are in contact with one or more unheated rooms can be calculated through the following formula:

$$D_{Inc} = T_{au} \times [\Sigma (K \times A \times \Delta T) + 0.2\Sigma (K_i \times L \times \Delta T)] \quad \text{II.14}$$

According to the formula, we notice that this particular type of heat loss is conditioned by a coefficient (T_{au}), which is known as the temperature reduction coefficient, this coefficients value is bound to be between 0 and 1.

To calculate thetemperature reduction coefficient we use the following equation:

$$T_{au} = \frac{ti - tn}{ti - te} = \frac{de}{de - ac} \quad \text{II.15}$$

Whereas:

ti: Internal temperature (C°).

tn: Unheated volume temperature (C°).

te: External Temperature (C°).

de: Heat loss from the unheated room to the exterior (W/C°).

ac: Heat gain from the various heated rooms to the unheated room (W/C°).

In case of the temperature of the unheated volume not being given - such as in our case - we can find it throw calculation as follows:

$$tn = ti - [(ti - te) \times (\frac{de}{de - ac})] \quad \text{II.16}$$

B. Calculating Heat loss by air renewal:

When calculating Heat loss by air renewal we need to take into consideration:

- Losses due to the activity of ventilation devices; the specific ventilation flow rate is associated with these losses.
- Additional losses by infiltration due to wind effects.

Heat losses due to air renewal are expressed as follows:

$$\mathbf{Dr = 0.34 \times (Qv + Qs) \times \Delta T} \quad \text{II.17}$$

This expression contains the following elements:

0, 34: The heat volume of air.

Qv: The Fan volume flow.

The fan volume flow has an expression for itself:

$$\mathbf{Qv = MAX\ [(0.6 \times Vh).Q_{vref}]} \quad \text{II.18}$$

Whereas:

V_h: The living volume.

Q_{vref}: The reference extract of Qv.

With:

$$\mathbf{Qvref} = \frac{(\mathbf{5Qvmin + Qvmax})}{\mathbf{6}} \quad \text{II.19}$$

Qs: Additional flow by fan infiltration.

With:

$$\mathbf{Qs = \Sigma P_P \times ev} \quad \text{II.20}$$

And: $\mathbf{\Sigma P_P = P0 \times A}$ II.21

Ev: wind exposure coefficient against the wall.

P_P: Air infiltration from the wall.

P_0: Surface air infiltration from the opening.

C. Verification:

In order to verify the accuracy of our results, we follow the previous works with a verification equation expressed as follows:

$$D_T \leq 1,05\ D_{ref} \qquad \text{II.22}$$

The equation components are defined as:

Dref: Heat loss reference.

Where:

$$Dref = aS1 + bS2 + cS3 + dS4 + eS5 \qquad \text{II.23}$$

Each of the S symbols represents a particular wall surface in contact with the exterior, in order, they represent:

S1: The roofing.

S2: The floor.

S3: The walls.

S4: The doors.

S5: The windows.

Whereas the coefficients (a, b, c, d and e) are defined by the corresponding area of the establishment in question, in the case of our project, which is situated in the B area, the coefficients are to be as represented in the following table:

	Coefficients of the building				
Area	a	b	c	d	e
B	0,9	2,40	1,20	3,50	4,50

Table 2.10: Area B Dref coefficients (a, b, c, d and e). [D1]

2.6 Calculating winter season heat balance:

2.6.1 Calculating the unheated volume temperature tn:

To begin, we need to identify the unheated rooms of each floor from ground floor, first floor, second floor, third floor and fourth floor to apply the previously mentioned expression to calculate thetemperature of the unheated rooms through the previous equation:

$$\mathbf{tn} = \mathbf{ti} - [(\mathbf{ti} - \mathbf{te}) \times (\frac{de}{de-ac})$$

Ground floor	
Room	Temperature (C°)
Shop	16
Storage	16
Bathrooms	16

Table 2.11: Unheated room's temperature on the ground floor.

First floor	
Room	Temperature (C°)
Lobby	16
Storage	16
Showers	16
Bathrooms	16

Table 2.12: Unheated room's temperature on the first floor.

Second floor	
Room	Temperature (C°)
Lobby	16
Storage	16
Bathrooms	16

Table 2.13: Unheated room's temperature on the second floor.

Third floor	
Room	Temperature (C°)
Lobby	16
Storage	16
Bathrooms	16

Table 2.14: Unheated room's temperature on the third floor.

Fourth floor	
Room	Temperature (C°)
Lobby	16
Storage	16
Bathrooms	16

Table 2.15: Unheated room's temperature on the fourth floor.

2.6.2 Example of calculation of a room and final calculations results:

Take for example the room (kitchen G-07) of 227.87 m³. It is located in one of the corners of the ground floor.

The characteristics of the room are as follows:

An outer walls (double partition walls) is 14.73 m × 4.08 m (with K = 1.3 W / m² ° C)

Has 02 windows of 3.92 m² (with K = 5 W / m² ° C); 01 wooden door of 2.2 m² (with K = 3.5 W / m² ° C).

The temperatures:

- Te = 1 ° C and Ts = 10 ° C (based on baseline conditions in the region).
- Ti = 21 ° C (the desired temperature).
- Tn = 16 ° C (according to the calculations).

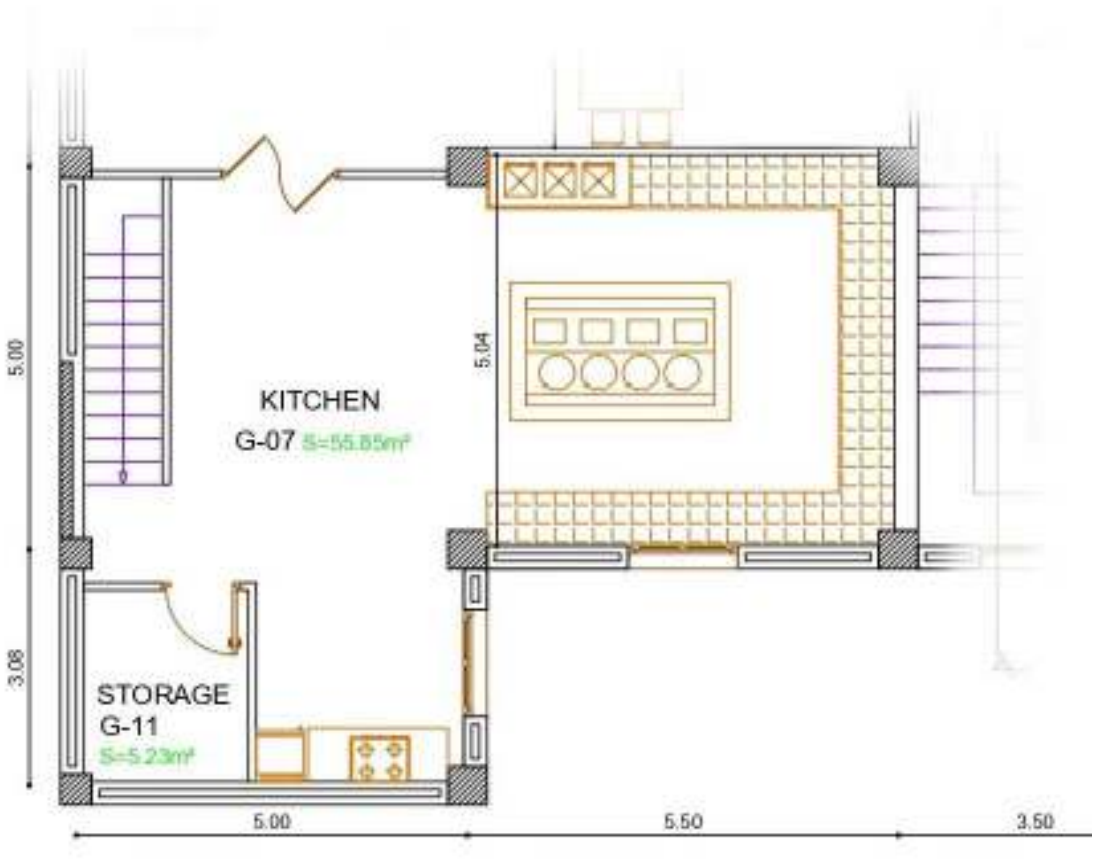


Figure 2.10: Example of calculating a room

A. Calculating heat loss by transmission:

-The basic losses are given by the following formula:

DT=Ds+ Dli+ Dsol+ Dlnc

-Surface losses by transmission through the walls D_S:

The surface losses by transmission are given by the following formula:

D_S = K × A

D_S= 1.3 ×60.13= 78.169 W/°C

-windows: D_S = 5×3.92= 19.6 W/°C

$D_{S.total}$=97.77 W/°C

-Loss through the bonds D_{li}:

D_{li}= D_S.total×0.2

D_{li}= 97.77×0.2 =19.55 W/°C

-Surface losses by transmission through the walls of unheated rooms D_{lnc}:

D_{lnc}=Tau× [(K×A) +0.2(K×A)]

Tau=0.25 (table 6.4 RTD C3-2)

D_{lnc}Wall: D_{lnc} =0.25× [(2.08×16.98) +0.2(2.08×16.98)] = 10.60 W/°C

-Door: D_{lnc} = 0.25× [(3.5×2.2) +0.2(3.5×2.2)] = 1.32W/°C

D_{lnc}floor: D_{lnc}=0.25× [(1.4×55.85) +0.2(1.4×55.85)] = 23.46 W/°C

D_{lnc} total=35.38 W/°C

D_T=152.7W/°C

B. Calculating Heat loss by air renewal:

-Loss by renewal of air Dr:

Dr= 0.34 × (Qv + Qs)

Q_V=MAX [0.6×V_h; $Q_{Vréf}$]

$$Q_{Vréf} = \frac{5Q_{Vmin} + Q_{Vmax}}{6}$$

Q_{Vmin} =120 (table 7.1 RTD C3-2)

Q_{Vmax} =210(table 7.2 RTD C3-2)

Q_V =MAX [0.6×227.87; 135]=136.72 m^3/h

Q_S =∑Q_{VO}×S_O

Q_S=14.5×3.92=56.84 m^3/h

Dr=0.34× (136.72+56.84) =65.81 W/°C

D_{tot} = (152.7+65.81) × (21-1) =218.51×20=4370.2W

D_{tot} = 4370.2W

Table 2.16: Calculation results of heat losses for the ground floor.

Locals	wall Surface DP(m²)	wall Surface SP(m²)	floor Surface (m²)	Surface of Terrace (m²)	Surface of Doors (m²)		Windows Surface (m²)	Height (m)	ΔT1 (°C)	ΔT2 (°C)	Dt (W/°C)	Dr (W/°C)	Dtot (W)
					exterior	interior							
G-01	55.73	-	161.66	-	3.08	-	48.12	4.08	20	5	374.2632	428.7358	16060
G-02	13.46	-	129.77	-	3.08	-	38.95	4.08	20	5	260.5050	375.0683	12711
G-03	unheated room												
G-04	18.80	19.58	22.33	-	-	-	-	4.08	20	5	50.9245	45.9000	1936.5
G-05	26.68	100.29	122.52	-	3.08	6.6	19.40	4.08	20	5	266.9386	320.4734	11748
G-06	43.16	-	28.82	-	-	-	1.96	4.08	20	5	88.1364	55.5628	2874
G-07	60.13	16.98	55.85	-	-	2.2	3.92	4.08	20	5	152.7012	65.8107	4370.2
G-08	unheated room												
G-09	unheated room												
G-10	unheated room												
G-11	unheated room												

Table 2.17: Calculation results of heat losses for the First floor.

Locals	wall Surface DP(m²)	wall Surface SP(m²)	floor Surface (m²)	Surface of Terrace (m²)	Surface of Doors (m²)		Windows Surface (m²)	Height (m)	ΔT1 (°C)	ΔT2 (°C)	Dt (W/°C)	Dr (W/°C)	Dtot (W)
					exterior	interior							
F1-01	127.9	64.22	223.8	-	-	9.9	-	3.54	20	5	239.3597	234.6616	9480.4
F1-02	30.9	-	31.32	-	-	-	30.9	3.54	20	5	140.9040	154.3022	5904.1
F1-03	18.87	-	19.80	-	-	-	18.87	3.54	20	5	86.0472	118.7175	4095.3
F1-04	-	-	15.2	-	-	-	-	3.54	20	5	0	62.9	1258
F1-05	-	19.49	16.47	-	-	-	-	3.54	20	5	12.1618	62.9	1501.2
F1-06	-	13.10	15.71	-	-	-	-	3.54	20	5	8.1744	62.9	1421.5
F1-07	-	4.96	11.2	-	-	2.64	-	3.54	20	5	3.0317	82.3779	1708.2
F1-08	15.36	6.02	45.67	-	-	2.64	-	3.54	20	5	31.4112	82.3779	2162
F1-09	29.03	-	18.21	-	-	-	-	3.54	39	24	45.2868	62.9	4219.3
F1-10	-	10.27	33.78	-	-	2.64	-	3.54	20	5	6.3451	82.3779	1774.5
F1-11	30.30	13.10	18.56	-	-	-	-	3.54	39	24	55.4424	62.9	4615.35
F1-12	17.7	70.4	25.39	-	-	2.64	1.96	3.54	20	5	80.1806	92.0407	3444.4
F1-13	unheated room												
F1-14	unheated room												
F1-15	unheated room												

Table 2.18: Calculation results of heat losses for the second floor.

Locals	Wall Surface DP(m²)	Wall Surface SP(m²)	floor Surface (m²)	Surface of Terrace (m²)	Surface of Doors (m²)		Windows Surface (m²)	Height (m)	ΔT1 (°C)	ΔT2 (°C)	Dt (W/°C)	Dr (W/°C)	Dtot (W)
					exterior	interior							
F2-01	unheated room												
F2-02	unheated room												
F2-03	unheated room												
F2-04	unheated room												
F2-05	37.6	23.26	28.15	-	2.64	3.74	1.96	3.20	20	5	84.0005	89.2074	3464.1
F2-06	15.68	24.99	30.50	-	2.64	3.74	1.96	3.20	20	5	50.8848	89.2074	2801.7
F2-07	9.69	18.08	18.7	-	-	3.74	9.69	3.20	20	5	55.3786	88.7297	2882.2
F2-08	16	24.64	32.00	-	-	3.74	16	3.20	20	5	88.2456	107.3947	3912.9
F2-09	17.12	25.28	31.14	-	-	3.74	17.12	3.20	20	5	93.7522	110.9443	4093.9
F2-10	13.92	22.24	25.52	-	2.64	3.74	3.84	3.20	20	5	54.7704	89.2074	3064.9
F2-11	28.7	22.27	26	-	2.64	3.74	3.84	3.20	20	5	77.8459	89.2074	3526.3
F2-12	unheated room												
F2-13	unheated room												
F2-14	unheated room												
F2-15	unheated room												

Table 2.19: Calculation results of heat losses for the second floor.

Locals	wall Surface DP(m²)	wall Surface SP(m²)	floor Surface (m²)	Surface of Terrace (m²)	Surface of Doors (m²)		Windows Surface (m²)	Height (m)	ΔT1 (°C)	ΔT2 (°C)	Dt (W/°C)	Dr (W/°C)	Dtot (W)
					exterior	interior							
F2-16	unheated room												
F2-17	unheated room												
F2-18	unheated room												
F2-19	unheated room												
F2-20	55.84	71.84	91.11	-	2.2	3.74	5.74	3.2	20	5	159.1824	104.5965	5275.6
F2-21	unheated room												
F2-22	27.04	18.85	19.96	-	2.64	3.74	3.92	3.2	20	5	73.4774	98.8702	3447
F2-23	unheated room												
F2-24	unheated room												
F2-25	17.28	61.44	29.33	-	2.64	3.74	3.92	3.2	20	5	84.8280	98.8702	3674
F2-26	unheated room												
F2-27	unheated room												
F2-28	unheated room												
F2-29	unheated room												

Table 2.20: Calculation results of heat losses for the third floor.

Locals	wall Surface DP(m²)	wall Surface SP(m²)	floor Surface (m²)	Surface of Terrace (m²)	Surface of Doors (m²)		Windows Surface (m²)	Height (m)	ΔT1 (°C)	ΔT2 (°C)	Dt (W/°C)	Dr (W/°C)	Dtot (W)
					exterior	interior							
F3-01	unheated room												
F3-02	unheated room												
F3-03	unheated room												
F3-04	unheated room												
F3-05	37.6	23.26	28.15	-	2.64	3.74	1.96	3.20	20	5	84.0005	89.2074	3464.1
F3-06	15.68	24.99	30.50	-	2.64	3.74	1.96	3.20	20	5	50.8848	89.2074	2801.7
F3-07	9.69	18.08	18.7	-	-	3.74	9.69	3.20	20	5	55.3786	88.7297	2882.2
F3-08	16	24.64	32.00	-	-	3.74	16	3.20	20	5	88.2456	107.3947	3912.9
F3-09	17.12	25.28	31.14	-	-	3.74	17.12	3.20	20	5	93.7522	110.9443	4093.9
F3-10	13.92	22.24	25.52	-	2.64	3.74	3.84	3.20	20	5	54.7704	89.2074	3064.9
F3-11	28.7	22.27	26	-	2.64	3.74	3.84	3.20	20	5	77.8459	89.2074	3526.3
F3-12	unheated room												
F3-13	unheated room												
F3-14	unheated room												
F3-15	unheated room												

Table 2.21: Calculation results of heat losses for the third floor.

Locals	wall Surface DP(m²)	wall Surface SP(m²)	floor Surface (m²)	Surface of Terrace (m²)	Surface of Doors (m²)		Windows Surface (m²)	Height (m)	ΔT1 (°C)	ΔT2 (°C)	Dt (W/°C)	Dr (W/°C)	Dtot (W)
					exterior	interior							
F3-16	unheated room												
F3-17	unheated room												
F3-18	unheated room												
F3-19	unheated room												
F3-20	55.84	71.84	91.11	-	2.2	3.74	5.74	3.2	20	5	159.1824	104.5965	5275.6
F3-21	unheated room												
F3-22	27.04	18.85	19.96	-	2.64	3.74	3.92	3.2	20	5	73.4774	98.8702	3447
F3-23	unheated room												
F3-24	unheated room												
F3-25	17.28	61.44	29.33	-	2.64	3.74	3.92	3.2	20	5	84.8280	98.8702	3674
F3-26	unheated room												
F3-27	unheated room												
F3-28	unheated room												
F3-29	unheated room												

Table 2.22: Calculation results of heat losses for the fourth floor.

Locals	wall Surface DP(m²)	wall Surface SP(m²)	floor Surface (m²)	Surface of Terrace (m²)	Surface of Doors (m²)		Windows Surface (m²)	Height (m)	ΔT1 (°C)	ΔT2 (°C)	Dt (W/°C)	Dr (W/°C)	Dtot (W)
					exterior	interior							
F4-01	unheated room												
F4-02	unheated room												
F4-03	unheated room												
F4-04	unheated room												
F4-05	37.6	23.26	28.15	28.15	2.64	3.74	1.96	3.20	20	5	121.7447	89.2074	4219.1
F4-06	15.68	24.99	30.50	30.50	2.64	3.74	1.96	3.20	20	5	90.7788	89.2074	3599.8
F4-07	10.24	18.08	18.7	18.7	2.64	3.74	1.96	3.20	20	5	62.5462	89.2074	3034.9
F4-08	16	24.64	32.00	32.00	2.64	3.74	1.96	3.20	20	5	93.0216	89.2074	3648.2
F4-09	17.12	25.28	31.14	31.14	2.64	3.74	1.96	3.20	20	5	98.6959	89.2074	3758.1
F4-10	13.92	22.24	25.52	25.52	2.64	3.74	3.84	3.20	20	5	88.1505	89.2074	3547.2
F4-11	28.7	22.27	26	26	2.64	3.74	3.84	3.20	20	5	111.8539	89.2074	4021.4
F4-12	unheated room												
F4-13	unheated room												
F4-14	unheated room												
F4-15	unheated room												

Table 2.23: Calculation results of heat losses for the fourth floor.

Locals	wall Surface DP(m²)	wall Surface SP(m²)	floor Surface (m²)	Surface of Terrace (m²)	Surface of Doors (m²)		Windows Surface (m²)	Height (m)	ΔT1 (°C)	ΔT2 (°C)	Dt (W/°C)	Dr (W/°C)	Dtot (W)
					exterior	interior							
F4-16	unheated room												
F4-17	unheated room												
F4-18	unheated room												
F4-19	unheated room												
F4-20	55.84	71.84	91.11	91.11	2.2	3.74	5.74	3.2	20	5	278.343	104.5965	7659.1
F4-21	unheated room												
F4-22	27.04	18.85	19.96	19.96	2.64	3.74	3.92	3.2	20	5	99.5851	98.8702	3969.1
F4-23	unheated room												
F4-24	unheated room												
F4-25	17.28	61.44	29.33	29.33	2.64	3.74	3.92	3.2	20	5	123.1916	98.8702	4441.3
F4-26	unheated room												
F4-27	unheated room												
F4-28	unheated room												
F4-29	unheated room												
F4-30	unheated room												

F4-31	unheated room
F4-32	unheated room

Levels	Dtot (W)
Ground floor	49699.7
First floor	41584.25
Second floor	36142.6
Third floor	36142.6
Fourth floor	41898.2
ΣDtot (W)	205467.35

Table 2.24: Total thermal losses for the hotel.

2.5.3 Verification:

Take for example the room (kitchen G-07)

$D_{réf}$ = (0.9×S1 + 2.4×S2 + 1.2×S3 +3.5×S4 + 4.5×S5)

S1= 0 m²

S2= 55.85 m²

S3= 77.11 m²

S4= 2.2 m²

S5= 3.92 m²

$D_{réf}$ = (0.9×0 + 2.4×55.85 + 1.2×77.11 +3.5×2.2 + 4.5×3.92)= 251.912 W/°C

1.05×D_{ref} = 264.51W/°C

After calculation we have:

D_T=152.7W/°C

So: 152.7<264.51

So we can say that the condition is true

Chapter III

Heat gain calculation

3.1 Introduction:

The previous chapter has covered thermal calculation and heat loss calculation, however, in order to achieve a full heat balance calculation and perform a full-on HVAC system study of any particular building, understanding heat loss alone is not enough, which raises the need for understanding heat gain and its methods of calculation.

Heat gain is the term given to a temperature rise within a space due to heat from the sun (solar radiation), heat from surfaces (long wave infrared radiation), heat originating from other sources within the space (such as heating appliances, ovens, people, mechanical systems, lights and computers) and so on. It is the heat that is gained from such sources that changes the prevailing temperature within the space.

Heat gain can be positive, for example providing 'free' heating to spaces on sunny but cold days, or it can be negative, overheating spaces in the summer so that they are uncomfortable to occupy.

3.2 Presentation of the Regulatory Technical Document:

In the previous study concerning heat loss, the document used to establish the methodical basis from which to calculate was the RTD C.3-2, whereas in the case of our current study in question which is heat gain, we will use a similar Regulatory technical document which is complimentary to the previous one named RTD C.3-4, in addition to that, in respect to our project, we will base our calculations on the European EN12831 Standard, in order to insure a well balanced, most accurate and precise result for not only the heat gain calculation but also for the resulting heat balance.

3.2.1 Objective of the RTD C3-4:

Not very different from the RTD C.3-2 the RTD C.3-4 is also a regulatory technical document dedicated to set the methodological grounds for insuring a well balanced environment heat wise, however, when the C.3-2 main focus is the winter season and its heat loss, our current document focuses on summer season with its heat gains where it allows:

- Determining heat gain for buildings or particular rooms.
- Deciding the dimensions of the air conditioning system networks of buildings.
- Just as in the previous document the the thermal conception of establishments and verifying the compliance of buildings with thermal regulations.

The areas of usage for this particular regulatory technical document do not vary and identical to those in the in the RTD C3-2 (see 2.4.1.2).

3.2.2 Calculations:

Heat gain as it stands in the calculation of heat balance in summer, as sensitive heat gain ($\mathbf{A_s}$) and latent heat gain ($\mathbf{A_l}$) which are in orderly manner represented by the following formulae:

$$\mathbf{A_s = APO + AV + Al_s + AINF_s \quad [W]} \qquad \text{III.1}$$

And:

$$\mathbf{A_l = Al_l + AINF_l \quad [W]} \qquad \text{III.2}$$

With the following representing:

- **APO:**Opaque walls heat gains.
- **AV:**Glass wall heat gains.
- **Al_s**and **Al_l:**The sensitive and latent parts of the internal gains.
- **$AINF_s$**and**$AINF_l$:**The sensitive and latent parts of heat gain due to air infiltration.

3.2.2.1 Calculating Sensitive heat gain:

A. Calculating Opaque panels heat gain:

When the need for calculating opaque panels heat rises, it would be expressed in the following general expression:

$$\mathbf{APO(t) = APO_{ext}(t) + APO_{int}(t) + APO_{soil}(t)[W]} \quad \text{III.3}$$

With :

- $\mathbf{APO_{ext}(t)}$ **:** Heat gain via external panels (W).
- $\mathbf{APO_{int}(t)}$ **:** Heat gain via internal panels (W).
- $\mathbf{APO_{soil}(t)}$ **:** Heat gain via panels in contact with the soil (W)

A.1 External opaque Panels:

$$\mathbf{APO(t)=1,2\times K_{été}\times S_{int}\times \Delta te(t)[W]} \quad \text{III.4}$$

With :

- **1,2:**A coefficient that takes into consideration the linear and lateral gains.
- $\mathbf{K_{été}}$ **:** The transmission coefficient in the straight part of the wall considered for summer according to RTD C.3-2 (W/m².°C).
- $\mathbf{S_{int}}$ **:**The total interior surface of the wall considered for roofs in slope, whilst taking the horizontal projection of the surface (m²).
- Δ**te(t) :**The equivalent difference of temperature at a given time t (C°).

In order to calculate the equivalent difference of temperature, we need to distinguish whether the wall in question is shadowed or sunny, concluding the following formulae:

- For a sunny wall:

$$\mathbf{\Delta te(t) = \Delta tes(t) + C\Delta te + \frac{\alpha}{0.9}\,[\Delta tem(t) - \Delta tes(t)]\frac{It,b}{It,b(40)}[C°]} \quad \text{III.5}$$

- For a shadowed wall 24/7:

$$\mathbf{\Delta tes(t) + C\Delta te[C°]} \quad \text{III.6}$$

We identify from these two formulae the following:

- **$\Delta tes(t)$:** Is the temperature difference equivalent to time t considering that the wall is shadowed; the values of Δtes (t) for vertical walls and from the (table3.1) for horizontal panels (C°).
- **$C\Delta te$:** Is a correction factor, where on one hand at the maximum difference (ΔTsmax) of the dry temperature of the air between the outside and the inside for the month in question, while on the other hand at the basic daytime deviation (Eb) for the month in question (C°).
- **α:** The wall absorption factor.
- **$\Delta tem(t)$:** the equivalent temperature difference at time t for the orientation of the wall in question; the values of Δtem (t) are given by the table for vertical walls and by table 3.3 for the horizontal walls. (C°).
- **$I_{t,b}$:**Is the total base radiation for the month with the latitude and orientation taken into consideration (W/m²).
- **$I_{t,b(40)}$:** The total base radiation for the month of July with a latitude of 40° North and orientation considered(W/m²).

Please note:

- Calculations according to the RTD C.3-4 consider real solar time for the month of July RST_{July} = 15h.
- Considering that our project is a hotel, colors and shades will reflect white and crème which enables α = 0,5 and Eb = 15 C°

Total Base Radiation $I_{t,b}$and $I_{d,b}$(W/m²)										
Latitude North	Month	Orientation								
		N	NE	E	SE	S	SW	W	NW	H
20°	July	59	435	514	267	44	267	514	435	791
	August	34	372	520	356	81	356	520	372	788
	September	31	273	514	441	205	441	514	273	733
30°	July	50	413	516	315	94	315	516	413	776
	August	34	340	520	406	198	406	520	340	741
	September	28	284	498	479	330	479	498	284	668
40°	July	47	400	516	394	217	394	516	400	734
	August	34	321	511	459	321	459	511	321	675
	September	28	183	470	511	441	511	470	183	577

Table 3.1: Total base radiation $I_{t,b}$and $I_{d,b}$. [D2]

A.2 Internal opaque panels:

Calculating the internal opaque panels' heat gain requires the following formula:

$$\mathbf{APO\ (t)_{int}=K_{été}\times S_{int}\times \Delta T[W]} \quad \text{III.7}$$

With :

- **ΔT :**The difference in temperature between the two rooms in question. (C°).

However, the difference in temperature between the two rooms differs depending whether the adjacent room is air conditioned or not, concluding **ΔT**being referred to as $\mathbf{\Delta t_{Inc}(t)}$ which is represented by the following definition:

$$\mathbf{\Delta t_{Inc}(t)=[(TS_e(t)-C_{inc})-TS_{b,i}][C°]} \quad \text{III.8}$$

With :

- $\mathbf{TS_e(t)}$ **:**The dry exterior temperature at a given time t (C°).
- $\mathbf{C_{inc}}$ **:**Correcting coefficient.
- $\mathbf{TS_{b,i}}$ **:**The basic interior temperature (C°).

A.3 Panels in contact with the soil:

As for the heat gain in panels in contact with the soil, they are calculated by the following formula:

$$\mathbf{APO_{sol}(t)= K_{été} \times S_{gain\times}(TS_m -TS_{b,i}) [W]} \quad \text{III.9}$$

With:

- $\mathbf{S_{gain}}$ **:**The Surface of the heat gain area (m^2).
- $\mathbf{TS_m}$ **:**Temperature of the air conditioned room (C°).

B. Calculating glass panel heat gain:

Glass panel heat gain is the combined effort of solar radiation as well as temperature gradient penetrating the panel and into the room, which is expressed as follows:

$$\mathbf{AV(t)= AVT(t)+ AVE(t)\ \ [W]} \quad \text{III.10}$$

With :

- **AVT (t):** Represents the heat gains that are due to temperature gradients penetrating the glass panels.
- **AVE (t):** Represents the heat gains that are due to solar rays penetrating the glass panels.

The heat gains that are due to temperature gradients can be calculated via the following formula:

$$\mathbf{AVT(t) = 1{,}2 \times K_{été} \times S_{ouv} \times [TS_e(t) - TS_{b.\,i}]\ [W]} \qquad \text{III.11}$$

With:

- **1,2 :** Non dimensional factor assembling heat gain through thermal bridges (thermal breaches)
- $\mathbf{K_{été}}$ **:**Transmission coefficient of the straight heat gain flow of the panel in question (W/m². °C).
- $\mathbf{S_{ouv}}$ **:**The surface of the opening in the opaque wall in question (m^2).
- $\mathbf{TS_e(t)}$ **:** The dry exterior temperature at a given time t (C°).
- $\mathbf{TS_{b.\,i}}$ **:**The basic interior temperature (C°).

While the heat gains that are due to solar rays penetrating the glass panels are calculated by the following formula:

$$\mathbf{AVE(t)=(SV_{ens} \times I_t + (SV - SV_{ens}) \times I_d) \times FS \times N_{pvi}\ [W]} \qquad \text{III.12}$$

With:

- $\mathbf{SV_{ens}}$**:** The sunny glass surface (m^2).
- $\mathbf{I_t}$**:** The real total maximum radiation (W/m^2).
- $\mathbf{SV_{omb}}$**:**The shadowed glass surface (m^2).
- $\mathbf{I_d}$**:** The real diffused maximum radiation (W/m^2).
- **FS:**The Glazing solar factor.
- $\mathbf{N_{pvi}}$**:** The amortization coefficient relating to heat gains via sunshine penetrating glass panels at a time t.

C. Calculating internal heat gains:

The quantities of calorific energy ejected in latent and sensitive states inside air conditioned rooms are distinguished as internal heat gains.

C.1 Gains due to occupants:

Heat gains caused by occupants are latent and sensitive sourced of heat, therefore the consideration and account of susceptible people -as well as animals- frequenting these air conditioned rooms is a must and thus, calculating these internal heat gains requires the following expression:

$$\mathbf{A_I\ (t)= A_{Is}(t)+A_n(t) = \Sigma\ (CS \times A_S \times N) + \Sigma\ (CS \times A_l)} \qquad \text{III.13}$$

With :

- A_s : The sensitive part of the internal gains (W).
- A_l: The latent part of the internal gains (W).
- CS: The relative coefficient of simultaneity.
- N: The relative coefficient of amortization.

C.2 Gains due to lighting:

Lighting devices share one thing in commune that is the fact of owning a sensitive heat source. If the lighting power of a lighting setup is known, the heat gains of this previously mentioned internal heat is calculated by the following formula:

$$\mathbf{AL = \sum\ (W_n \times C_{me} \times C_{cr})} \qquad \text{III.14}$$

With:

- **W_n:** The nominal power of the fluorescent tube or bulb.
- **C_{me}:**The increase coefficient for lighting gain taking 1.2 for incandescence lamps and 1.25 for the fluorescent tubes.
- **C_{cr}:**The residual heat percentage which is in our case equal to 1.

If the lighting power of the setup is unknown, the internal heat gains due to lighting are calculated whilst considering the surfaces of each and every lighted room via the values given in the following table:

Room designation and utility	Recommended luminous intensity (lux)	Power ejected by lamps or tubes (W/m_2)	
		Incandescence lamps	Fluorescent tubes
Laboratories, offices and variant rooms	250	55	16

Table 3.2: Lighting internal heat gains. [D2]

C.3 Gains due to electrical machines:

Electrical machines due to their electrical components produce and eject a considerable amount of heat depending on the electrical machine and/or its utility, thus, their heat ejection is taken into consideration and is taken by:

- Indications provided by the source of the machines (factory, producer…ect).
- Indications given in the following tables:

Devices	Heat gains		
	Sens.	Sens.	Sens.
Display stand m^3 of interior volume	640	640	640
Cash register	48	48	48
Refrigerator	1750	1750	1750
Blow drier system of 5 seats	4483	4483	4483
Blow drier system of 10 seats	6273	6273	6273
1 seat blow drier	690	690	690
Sterilizer 152x205x432 (mm)	805	805	805
Sterilizer 228x254x508 (mm)	1518	1518	1518
Sterilizer 254x305x560 (mm)	420	420	420
Sterilizer 254x305x914 (mm)	3048	3048	3048
Sterilizer 305x406x620 (mm)	2728	2728	2728
Sterilizer utensils 406x406x620 (mm)	3167	3167	3167
Sterilizer 254x305x914 (mm)	3676	3676	3676
Coffee grinder	500	500	500
Mixer	1000	1000	1000
Fabric iron	500	500	500
Vacuum	200	200	200

Table 3.3: The heat gain of some electrical devices (1). [D2]

Devices	Heat gains		
	Sens.	Sens.	Sens.
Typing machine	90	90	90
Desk computer	450	450	450
2200W to 6600W Laptop	4400	4400	4400
5000 lines/minute printer	1800	1800	1800
300 to 600 lines/minute printer	376	376	376
Terminal Screen	160	160	160

Table 3.4: The heat gain of some electrical devices (2). [D2]

D. Calculating heat gain by external air intrusion:

Heat gains that are due to external air intrusion are considered a sensitive and latent form of heat gain from which we can distinguish:

- Heat gains due to external intrusions that interfere directly with the room's thermal balance, these gains are to be associated and integrated into the infiltration flow Qv_{inf}.
- Heat gains due to the intrusion of fresh air which interferes directly with the total heat balance, these gains however are to be associated with the fresh air flow Qv_{fa}.

The sensitive heat gains that are due to air renewal $AREN_S$, and due to infiltrations $AINF_S$are given by the following formulae:

$$\mathbf{AREN_S = 0{,}32 \times qv_{an} \times (TS_e(t) - TS_{bi}(t))} \quad \text{III.15}$$

And : $$\mathbf{AINF_S = 0{,}32 \times qv_{inf} \times (TS_e(t) - TS_{bi}(t))} \quad \text{III.16}$$

Whereas :

- **0,32 :** The product of the specific heat of the air (1004 J/Kg°C) by its volumic mass (1.15 Kg/m³) as well as by the 1/3600 conversion factor with a dimension of (J/m_3. °C)**.**
- **Qv_{inf}:**Flow due to infiltration (m³/h).

- **Qv_{an}:**Flow due to fresh air (m^3/h).
- **$TS_e(t)$:**The dry exterior temperature at a given time t (C°).
- **TS_{bi} :**The dry temperature of the interior (C°).

The latent heat gains that are due to air renewal $AREN_t$, and due to infiltrations $AINF_t$are given by the following formulae:

$$\mathbf{AREN_t = 0{,}797 \times qv_{an} \times MAX[(HS_{b,e} - HS_{b,i}) ; 0]} \quad \text{III.17}$$

And: $$\mathbf{AINF_t = 0{,}797 \times qv_{inf} \times MAX[(HS_{b,e} - HS_{b,i}) ; 0]} \quad \text{III.18}$$

With:

- **0,797 :**The product of the average volumic mass the air (1,15 Kg/m^3) by the water vaporization heat(2498J/g) as well as by the 1/3600 conversion factor with a dimension of (J/m_3. °C)**.**
- **Qv_{inf}:** Flow due to infiltration (m^3/h).
- **Qv_{fa}:**Flow due to fresh air (m^3/h).
- **$HS_{b,e}$:** The basic specific humidity of the exterior ($g_v/kg_{humid\ air}$)
- **$HS_{b,i}$:** The basic specific humidity of the interior ($g_{vap}/kg_{humid\ air}$)

D.1 <u>Fresh air flow:</u>

The flow of fresh air circulating into a room is calculated by the following formula which is given by the RTD C.3-2 booklet n°1:

$$\mathbf{Qv_{an} = MAX[(0{,}6 \times V_h) ; qve_{ref}]} \quad \text{III.19}$$

$$\mathbf{Qve_{ref}} = \frac{\mathbf{5qve}_{min} + qve_{max}}{\mathbf{6}} \quad \text{III.20}$$

D.2 <u>Infiltration air flow:</u>

The flow of infiltrating air in a room is calculated by the following foruma which is given by the RTD C.3-2 booket n°1:

$$\mathbf{Qv_{inf} = (Qvo_{inf,i} \times S_{ouv,i})} \quad \text{III.21}$$

With:

- **$Qvo_{inf,i}$:** The infiltration flow corresponding to the considered opening

E. Verification:

Standard regulatory verifications for rooms specified for offices needbeing put to action in the following manner and through the following steps:

- Heat gains by opaque panels.
- Heat gains by glass panels.
- Calculating reference calorific gains.
- Verification of the conformity according to the summer thermal standard regulations of the rooms.
- The heat gains of opaque panels and glass panels combined must verify at least for the month of July at 15h (3P.M), therefore exists the following expression:

  APO (15h) +AV (15h) ≤ 1,05×$A_{réf}$**(15h)** III.22

  The reference heat gains are given by the following expression:

$$\mathbf{A_{réf} = A_{réf,ph} + A_{réf,pv} + A_{réf,pvi}} \quad \text{III.23}$$

Whereas:

- $\mathbf{A_{réf,ph}}$: The reference heat gains via horizontal opaque panels (W).
- $\mathbf{A_{réf,pv}}$: The reference heat gains via vertical opaque panels (W).
- $\mathbf{A_{réf,pvi}}$: The reference heat gains via glass panels (W).

E.1 Reference heat gains via horizontal opaque panels:

Reference calorific gains via horizontal opaque panels can be calculated by the following formula:

$$\mathbf{A_{réf,ph} = \sum (a \times S_{int} \times \Delta TS_{réf,ph})} \quad \text{III.24}$$

With:

- **a:** A coefficient related to the nature and utility of the building in question.
- $\mathbf{S_{int}}$ **:**The surface of the associated horizontal panels (m^2).

- $\Delta TS_{réf,ph}$ **:** The temperature difference of the reference for the horizontal opaque panel (C°).

E.2 Reference heat gains via vertical opaque panels:

Reference heat gains via vertical opaque panels are resolved by calculating the following formula:

$$A_{réf,pv}=\sum(c\times S_{int} \times \Delta TS_{réf,pv}) \quad \text{III.25}$$

With:

- **c:** A coefficient associated with the nature and functions of the climatic area.
- S_{int} **:**The surface of the associated vertical panels (m^2).
- $\Delta TS_{réf,pv}$**:** The temperature difference of the reference for the vertical opaque panel (C°).

E.3 Reference heat gains via glass panels:

Reference heat gains via glass panels are calculated by the following formula:

$$A_{réf,pvi}= AVE_{réf} + AVT_{réf} \quad \text{III.26}$$

With:

- $AVE_{réf}$ **:** Reference gains via sunshine penetrating the glass panels (W).
- $AVT_{réf}$ **:**Reference gains via temperature gradient (W).

Reference gains via sunshine penetrating the glass panels are given by:

$$AVE_{réf}= [SV_{ens} \text{ x } It + (SV - SV_{ens}) \text{ x } Id] FS_{réf} \text{ x } N_{PVI,réf} \quad \text{III.27}$$

With:

- **SV:** The total Glass surface (m^2).
- SV_{ens}**:** The sun exposed glass surface at 15h (3P.M) (m^2).
- **It:** The total maximal radiation for the considered latitude and orientation
- **Id:**The diffused maximal radiation.
- $FS_{réf}$**:**The solar factor of the reference.
  - 0.15 for rooms designated for housing and habitat.

- 0.38 for rooms designated for office purposes.

- **$N_{PVI,réf}$:**The amortization coefficient relative to heat gains of the glass panels in the direction of the orientation in question.

Reference gains via temperature gradients are calculated by the following expression:

$$\mathbf{AVT_{réf} = e' \times S_{ouv} \times \Delta TS_{réf,pv}} \qquad \text{III.28}$$

With:

- **e':** A relative coefficient to the nature of the room (W/m².°C)
  - 0.15 for rooms designated for housing and habitat.
  - 0.38 for rooms designated for office purposes.
- **S_{ouv}:**The surface of the openings in the wall (m^2).
- **$\Delta TS_{réf,pv}$:**The reference difference of temperature of the glass panels (C°).

3.3 Example of calculation of a room:

Take for example the suite (roomF2-20) of 291.55 m³. It is located in one of the corners of the second floor.

The characteristics of the room are as follows:

An outer wall (double partition walls) is 7 m × 3.2 m (with K_{ete} = 1.32 W / m² ° C)oriented to the west.

An outer wall (double partition walls) is 10.1 m × 3.2 m (with K_{ete} = 1.32 W / m² ° C)

facing south.

Has 03 windows of 5.88 m² (with K_{ete} = 5.2 W / m² ° C); 01 wooden door of 2.2 m² (with K_{ete} = 3.9 W / m² ° C).

An interior wall (simple partition walls) is 6.3 m × 3.2 m (with K_{ete} = 2.5 W / m² ° C)

Has 01 wooden door of 2.2 m² (with K_{ete} = 3.9 W / m² ° C).

The temperatures:

- TSbe = 37 ° C (based on baseline conditions in the region).
- TSbi = 24 ° C (the desired temperature).
- Eb = 15 ° C (based on baseline conditions in the region).

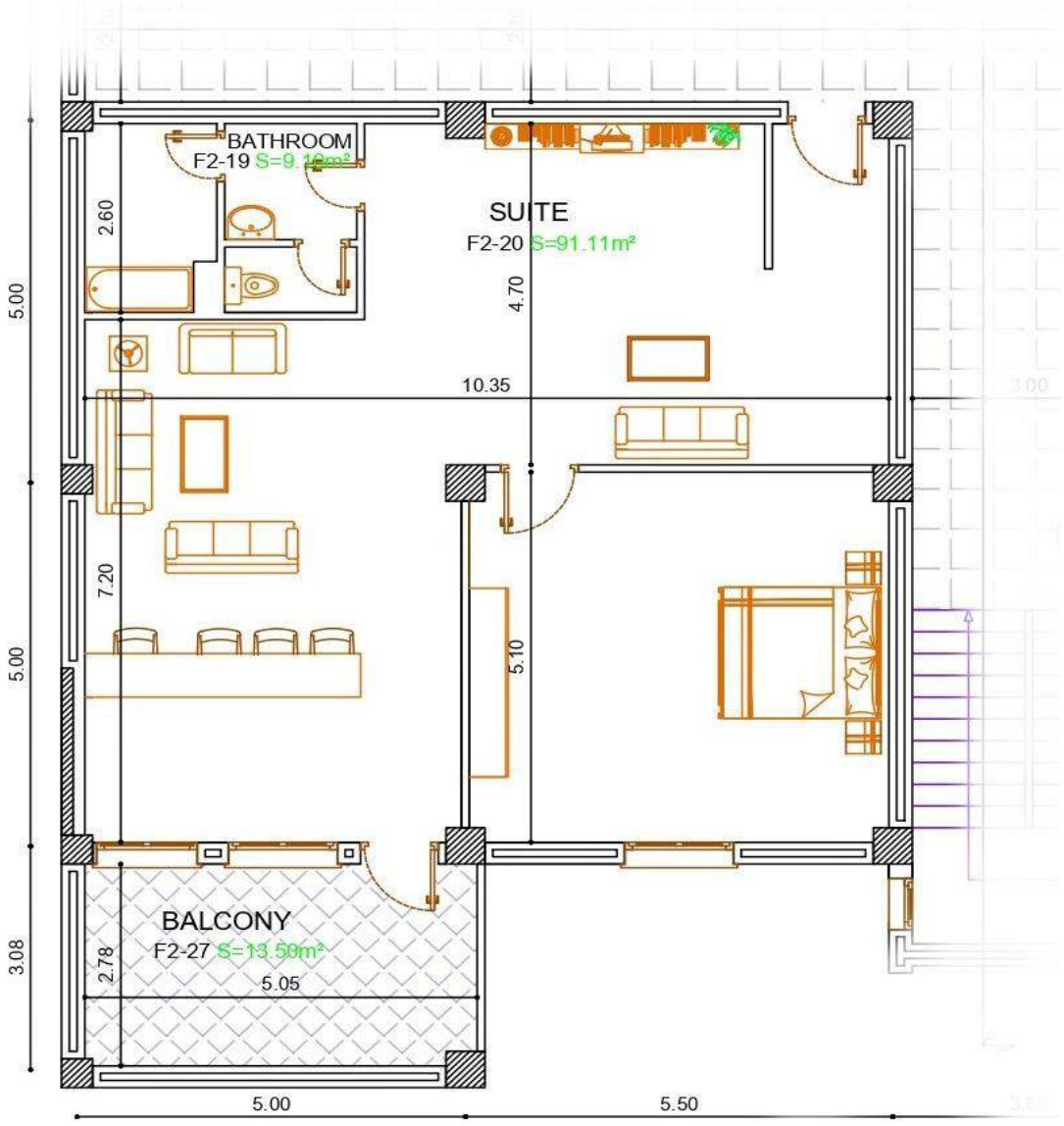


Figure 3.1: Example of calculating a room.

The sensible calorific intakes As and the latents Al:

$AS = APO(t) + AV(t) + AI_S + AINF_S(t)$ (W)

$Al = AI_l + AINF_l(t)$ (W).

A-Contributions through the opaque walls:

$APO(t) = APO(t)_{ext} + APO(t)_{int} + APO_{sol}(t)$

- The exterior walls:

$APO(t)=1,2\times K_{été}\times S_{int}\times \Delta te(t)$

$$\Delta te(t) = \Delta tes(t) + C\Delta te + \frac{\alpha}{0.9}\,[\Delta tem(t) - \Delta tes(t)]\frac{It, b}{It, b(40)}$$

Orientation	m_{surf}(kg/m²)	8	9	10	11	12	13	14	15	16
NE	300	-1,1	2,8	13,3	12,2	11,1	8,3	5,5	6,1	6,7
E	300	0	11,7	16,7	17,2	17,2	10,06	7,8	7,2	7,8
SE	300	0	7,2	11,1	13,3	15,6	14,4	13,9	11,7	10
S	300	-2,2	-1,7	-1,1	3,9	6,7	11,1	13,3	13,9	14,4
SW	300	0	0	0	1,1	4,4	6,7	13,3	17,8	19,4
W	300	0	0	0	1,1	2,2	3,9	5,5	10,6	14,4
NW	300	-2,2	-1,7	-1,1	0	1,1	3,3	4,4	5,5	6,7
N	300	-2,2	-1,7	-1,1	-0,5	0	1,7	3,3	4,4	5,5

Table 3.5: value of ΔTem (t) and ΔTes (t). [D2]

-West facing wall :

$$\Delta te(15) = 4.4 + 2.8 + \frac{0.5}{0.9}\,[10.6 - 4.4]\frac{516}{516}$$

$\Delta te(15) = 10.64$

$APO(15)=1.2\times1.323\times10.64\times22.4$

$APO(15)=378.54W$

-south facing wall :

$$\Delta te(15) = 4.4 + 2.8 + \frac{0.5}{0.9}\,[13.9 - 4.4]\frac{155.5}{217}$$

$\Delta te(15) = 10.98$

Sint=(10.1×3.2)-(3×1.96)-2.2=24.24

APO(15)=(1.2×1.323×10.98×24.24)+(1.2×3.9×10.98×2.2)=535.69 W

- The interior walls:

-two conditioned premises:

$APO(t)=K_{été}\times S_{int}\times[TSa-TSbi]$.

TSa-TSbi=24-24=0

APO=0

walls in contact with unconditioned local :

$APO\ (t)_{int}=K_{été}\times S_{int}\times\Delta T_{lnc}$

$\Delta t_{Inc}(t)=[(TS_e(t)-C_{inc})-TS_{b,i}]$

$TS_e(t)=TS_{be}-[C_{TS}(t)\times E_b]$

$TS_e(15)=37-(0\times15)=37$

$\Delta t_{Inc}(t)=(37-3)-24=10$

$S_{int}=20.1-1.65=18.51$

$APO(15)_{int}=(2.5\times10\times18.51)+(3.9\times10\times1.65)=527.1$ W

B- Calculating glass panel heat gain:

-Exterior panels:

AV(t)= AVT(t)+ AVE(t)

$AVT(t) = 1,2 \times K_{été} \times S_{ouv} \times [TS_e(t) - TS_{b.\,i}]$

AVT(15) = 1,2 × 5.2 × 5.88 × [37-24]

AVT(15)=476.98 W

$AVE(t)=(SV_{ens}\times I_t+(SV-SV_{ens})\times I_d)\times FS\times N_{pvi}$

SV_{ens}:

We have :$SV_{ens}=(H-H_0)(L-L_0)$

$L_0=Rtg\beta$; $H_0=Rtgh/cos\beta$; $\beta=a-a_m$

$Sina=cos\delta \times sinw/cosh$ $\delta=23.45\ sin(360/365(N+284))$

July 21 at 3 p.m : N=202 ; w(15h)=45°

$\delta=20.1$; $\varphi=36°$

$sinh=sin\delta sin\varphi+cos\delta cos\varphi cosw=0.739$; h=47.66

sina =0.986 ; a=80.37

West : am=90° ; $SV_{ens}=1.792m^2$

East : am=-90° ; $SV_{ens}=2.131m^2$

North : am=180° ; $SV_{ens}=0m^2$

South : am=0° ; $SV_{ens}=0.6m^2$

$I_t=[1\times0.92\times1.021\times1.05] \times155.5=153\ W/m^2$

$I_d=[1\times0.92\times1.021\times1.05] \times217=213.96\ W/m^2$

$N_{PVI}=57.5\%=0.575$

FS=0.64

AVE(15)=[(0.6×3)×153+(5.88-0.6)×213.96]×0.64×0.575

AVE(15)=422.798 W

AV(15)=899.783 W

C. Calculating internal heat gains:

C.1 Gains due to occupants:

$A_I(t)=A_{Is}(t)+A_{Il}(t)=\Sigma(CS\times A_S\times N)+\Sigma(CS\times A_l)$

CS=0.5 ; N=0.97 ; A_S=74 ; A_l=72

$A_I(15)=5(0.5\times75\times0.97)+5(0.5\times72)$

A_I (15)= 359.45 W

C.2 Gains due to lighting:

$A_I = \sum (W_n \times C_{me} \times C_{cr})$

W_n=8w/m² ;C_{me}=1.25 ; C_{cr}=1

AI=91.11×8×1.25×1

AI=911.1 W

C.3 Gains due to electrical machines:

A_I (t)= Σ (CS ×A_S × N) + Σ (CS × A_l)

television: As=150w ; Al=0 ; N=0.97 ; CS=0.5

Hotplate : As=500w ; Al=280w ; N=0.97 ; CS=0.5

A_I (15)=(0.5 ×(150+500) × 0.97) + (0.5 ×280)

A_I=455.25 W

internal heat gains:

AI(15h)=395.45+455.25+911.1=1725.8 W

D-Calculating heat gain by external air introduction:

-The sensitive heat gains :

-The sensitive heat gains that are due to air renewal $AREN_S$

$AREN_S$=0,32× qv_{an} × (TS_e(t) − TS_{bi}(t))

Qv_{an}= MAX[(0,6 × V_h) ; qve_{ref}]

Qve_{ref}= $\frac{5qve_{min}+qve_{max}}{6}$

Qve_{ref}= $\frac{5\times160+210}{6}$=168.33

$_{0.6}$×vh=0.6×3.2×91.11=174..93

Qv_{an}= MAX[174.93 ; 168.33]=174.93 m^3/h

$AREN_S$=0,32× 174.93 × (37 − 24)=727.7 W

-The sensitive heat gains that are due to infiltrations $AINF_s$

$AINF_S$=0,32× qv_{inf} × ($TS_e(t)$ − $TS_{bi}(t)$)

Qv_{inf}=($Qvo_{inf,i}$×$S_{ouv,i}$)

Qv_{inf}=(14.5×1.96×3)+(21.7×2.2)=132.565 m^3/h

$AINF_S$=0,32× 132.565 × (37 − 24)=551.47 W

-The latent heat gains :

-The latent heat gains that are due to air renewal $AREN_t$

$AREN_l$= 0,797× qv_{an} × MAX[($HS_{b,e}$ − $HS_{b,i}$) ; 0]

$AREN_l$= 0,797× 174.93 × MAX[(11–8.4) ; 0]=362.49 W

-The latent heat gains that are due to infiltrations $AINF_t$

$AINF_l$ =0,797× qv_{inf} × MAX[($HS_{b,e}$ − $HS_{b,i}$) ; 0]

$AINF_l$ =0,797× 132.565 × MAX[(11–8.4) ; 0]=274.7 W

APO	AV	AI	AINFs	ARENs	$AINF_l$	$AREN_l$	Total
1441.33	899.783	1725.8	551.47	727.7	274.7	362.49	5983.273W

Table 3.6: Total heat gains of the room(F2-20)

3.4 Heat gains calculation results:

Locals	APO(t) (w)	AV(t) (w)	AI(t) (w)	ARENs(t) (w)	AINFs(t) (w)	$AREN_l(t)$ (w)	$AINF_l(t)$ (w)	∑AP(t) (w)
G-01	3456.39	4302.6	7873.6	1646.28	715.76	820.05	356.54	19180.2
G-02	3190.11	3475.18	6477.8	1313.69	597.54	654.38	297.65	16006.4
G-03	unconditioned room							
G-04	1003.07	-	803.56	561.6	-	297.75	-	2665.98
G-05	2143.52	1521.81	2021.6	1242.4	615.59	618.87	306.64	8470.44
G-06	958.58	299.92	1512.1	561.6	118.22	297.75	58.89	3807.1
G-07	1469.32	570.608	2042.9	561.6	236.45	297.75	117.78	5296.41
G-08	270.5	344.68	580.24	561.6	130.04	297.75	64.78	2249.6
G-09	270.5	344.68	580.24	561.6	130.04	297.75	64.78	2249.6
G-10	unconditioned room							
G-11	unconditioned room							

Table 3.7: Calculation results of heat gains for the ground floor.

Locals	APO(t) (w)	AV(t) (w)	AI(t) (w)	ARENs(t) (w)	AINFs(t) (w)	$AREN_l$(t) (w)	$AINF_l$(t) (w)	∑AP(t) (w)
F1-01	1930.05	3911.36	17408.25	1977.45	2135.33	985.02	1063.66	29411.1
F1-02	489.35	2154.57	1035.54	796.6	1174.43	383.36	585.01	6618.86
F1-03	397.85	1986.6	560.03	796.6	1131.6	383.36	563.68	5819.72
F1-04	-	-	514.03	796.6	-	383.36	-	1693.99
F1-05	-	-	526.73	796.6	-	383.36	-	1706.69
F1-06	444.85	-	519.13	796.6	-	383.36	-	2143.94
F1-07	265.5	-	400.42	796.6	-	383.36	-	1845.88
F1-08	unconditioned room							
F1-09	unconditioned room							
F1-10	unconditioned room							
F1-11	unconditioned room							
F1-12	357.42	299.92	1551.36	796.6	118.22	383.36	58.89	3565.77
F1-13	1395.66	483.29	1288.9	796.6	202.67	383.36	100.95	4651.43
F1-14	unconditioned room							
F1-15	unconditioned room							

Table 3.8: Calculation results of heat gains for the first floor.

Locals	APO(t) (w)	AV(t) (w)	AI(t) (w)	ARENs(t) (w)	AINFs(t) (w)	$AREN_l$(t) (w)	$AINF_l$(t) (w)	∑AP(t) (w)
F2-01	unconditioned room							
F2-02	unconditioned room							
F2-03	unconditioned room							
F2-04	unconditioned room							
F2-05	935.79	216.56	498.03	700.27	316.83	348.82	157.82	3174.12
F2-06	608.63	216.56	521.53	700.27	316.83	348.82	157.82	2870.46
F2-07	363.1	1131.4	403.53	700.27	617.67	348.82	307.68	3872.47
F2-08	428.7	1838.6	536.53	700.27	965.12	348.82	480.75	5298.79
F2-09	460.6	1891.6	527.93	700.27	1037.5	348.82	516.8	5483.52
F2-10	580.61	216.56	543.62	700.27	316.83	348.82	157.82	2864.53
F2-11	790.23	216.56	548.42	700.27	316.83	348.82	157.82	3078.95
F2-12	unconditioned room							
F2-13	unconditioned room							
F2-14	unconditioned room							
F2-15	unconditioned room							
F2-16	unconditioned room							
F2-17	unconditioned room							
F2-18	unconditioned room							
F2-19	unconditioned room							
F2-20	1441.33	899.78	1725.8	727.7	551.47	362.49	274.7	5983.273
F2-21	unconditioned room							
F2-22	574.12	599.86	416.13	700.27	435.05	348.82	216.71	3290.96
F2-23	1528.45	283.72	2494.9	1360.46	118.23	677.68	58.89	6522.33
F2-24	unconditioned room							
F2-25	684.71	599.86	541.72	700.27	435.05	348.82	216.71	3527.14
F2-26	unconditioned room							
F2-27	unconditioned room							
F2-28	unconditioned room							
F2-29	unconditioned room							

Table 3.9: Calculation results of heat gains for the second floor.

Locals	APO(t) (w)	AV(t) (w)	AI(t) (w)	ARENs(t) (w)	AINFs(t) (w)	$AREN_l$(t) (w)	$AINF_l$(t) (w)	∑AP(t) (w)
F3-01	unconditioned room							
F3-02	unconditioned room							
F3-03	unconditioned room							
F3-04	unconditioned room							
F3-05	935.79	216.56	498.03	700.27	316.83	348.82	157.82	3174.12
F3-06	608.63	216.56	521.53	700.27	316.83	348.82	157.82	2870.46
F3-07	363.1	1131.4	403.53	700.27	617.67	348.82	307.68	3872.47
F3-08	428.7	1838.6	536.53	700.27	965.12	348.82	480.75	5298.79
F3-09	460.6	1891.6	527.93	700.27	1037.5	348.82	516.8	5483.52
F3-10	580.61	216.56	543.62	700.27	316.83	348.82	157.82	2864.53
F3-11	790.23	216.56	548.42	700.27	316.83	348.82	157.82	3078.95
F3-12	unconditioned room							
F3-13	unconditioned room							
F3-14	unconditioned room							
F3-15	unconditioned room							
F3-16	unconditioned room							
F3-17	unconditioned room							
F3-18	unconditioned room							
F3-19	unconditioned room							
F3-20	1441.33	899.78	1725.8	727.7	551.47	362.49	274.7	5983.273
F3-21	unconditioned room							
F3-22	574.12	599.86	416.13	700.27	435.05	348.82	216.71	3290.96
F3-23	1528.45	283.72	2494.9	1360.46	118.23	677.68	58.89	6522.33
F3-24	unconditioned room							
F3-25	684.71	599.86	541.72	700.27	435.05	348.82	216.71	3527.14
F3-26	unconditioned room							
F3-27	unconditioned room							
F3-28	unconditioned room							
F3-29	unconditioned room							

Table 3.10: Calculation results of heat gains for the third floor.

Locals	APO(t) (w)	AV(t) (w)	AI(t) (w)	ARENs(t) (w)	AINFs(t) (w)	$AREN_l$(t) (w)	$AINF_l$(t) (w)	∑AP(t) (w)
F4-01	unconditioned room							
F4-02	unconditioned room							
F4-03	unconditioned room							
F4-04	unconditioned room							
F4-05	1686.83	216.56	498.03	700.27	316.83	348.82	157.82	3925.16
F4-06	1422.37	216.56	521.53	700.27	316.83	348.82	157.82	3684.2
F4-07	862.01	1131.4	403.53	700.27	617.67	348.82	307.68	4371.38
F4-08	1282.46	1838.6	536.53	700.27	965.12	348.82	480.75	6152.55
F4-09	1291.42	1891.6	527.93	700.27	1037.5	348.82	516.8	6314.34
F4-10	1261.48	216.56	543.62	700.27	316.83	348.82	157.82	3545.4
F4-11	1483.91	216.56	548.42	700.27	316.83	348.82	157.82	3772.63
F4-12	unconditioned room							
F4-13	unconditioned room							
F4-14	unconditioned room							
F4-15	unconditioned room							
F4-16	unconditioned room							
F4-17	unconditioned room							
F4-18	unconditioned room							
F4-19	unconditioned room							
F4-20	3872.16	899.78	1725.8	727.7	551.47	362.49	274.7	8414.1
F4-21	unconditioned room							
F4-22	1106.65	599.86	416.13	700.27	435.05	348.82	216.71	3823.49
F4-23	6072.88	283.72	2494.9	1360.46	118.23	677.68	58.89	11066.76
F4-24	unconditioned room							
F4-25	1467.23	599.86	541.72	700.27	435.05	348.82	216.71	4309.66
F4-26	unconditioned room							
F4-27	unconditioned room							
F4-28	unconditioned room							
F4-29	unconditioned room							
Locals	APO(t) (w)	AV(t) (w)	AI(t) (w)	ARENs(t) (w)	AINFs(t) (w)	$AREN_l$(t) (w)	$AINF_l$(t) (w)	∑AP(t) (w)

F4-30	unconditioned room
F4-31	unconditioned room
F4-32	unconditioned room

Table 3.11: Calculation results of heat gains for the Fourth floor.

Floor	**AP(t) (W)**
ground floor	59925.73
first floor	57457.38
second floor	45966.54
third floor	45966.54
fourth floor	59379.67
∑AP(t) (W)	268695.86

Table 3.12: Total heat gains

Chapter IV

HVAC system selection

4.1Introduction:

Throughout chapter two and chapter three, we were able to calculate and determine the various calorific needs of our project, whether it is heat loss in winter, or heat gain in summer.

The main purpose for determining these gains and losses is to be able to select a corresponding heating and air conditioning system eligible for our project, thus, in this chapter we will discuss the multiple systems available in the market, choosing the most adequate system and calculating its power.

4.2 HVAC System selection:

4.2.1 Introduction:

As previously mentioned, thermal comfort is a necessity as crucial as the need for food and water, achieving that necessity however requires a set of installations and setups in order to insure a well balanced and comfortable thermal environment.

4.2.2 HVAC system types:

HVAC system stretch between a variety of setups that either perform solely for the purpose of heating or cooling individually, or in order to reply to the combined need of both calorific acquisition as well as cool air needs together, differing also in the fluid carrying the calorific heat varying between water, steam and air, from these systems we can distinguish the following four categories:

A. Heating and cooling split systems :

Heating and cooling split systems are known to be the most common types of HVAC systems. As the name suggests it is split into two separate units, one serves the purpose of heating and the other one is for cooling. It is to be noted that these systems come with indoor and outdoor units that can be easily identified.

Such HVAC products are equipped with a cooling system outside which uses compressors, refrigerants and coils in order to cool air and a fan to keep away the hot air. These are basically the AC units that are kept outside the home and run during the summer season.

On the other hand, the heater in the heating and cooling split system is placed in a basement and other storage space which uses gas to warm up the house. A traditional thermostat is used in this process to control the temperature to keep it within the comfort level. [4]

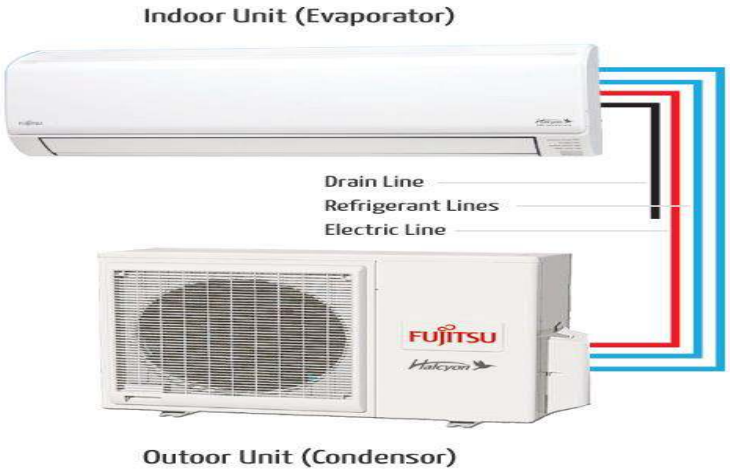


Figure 4.1: Split system air conditioning system.



Figure 4.2: Heating unit system.

B. Hybrid split system :

The hybrid system of HVACs carries similar features to the split systems but they do have some basic differences. This particular HVAC system carries a hybrid heating system which is counted as one of its unique features that helps lowering energy costs. It is up to the homeowners to set the key differences. [4]

The capacity to switch between gas power, which is faster and more complete, to electricity which is more efficient and quieter, allows homeowners to determine the way they want to heat up their homes. [4]

This system is also dependent on traditional ducts and thermostats and it offers all the advantages of a split system with an additional benefit of conserving energy and reducing utility bills. [4]

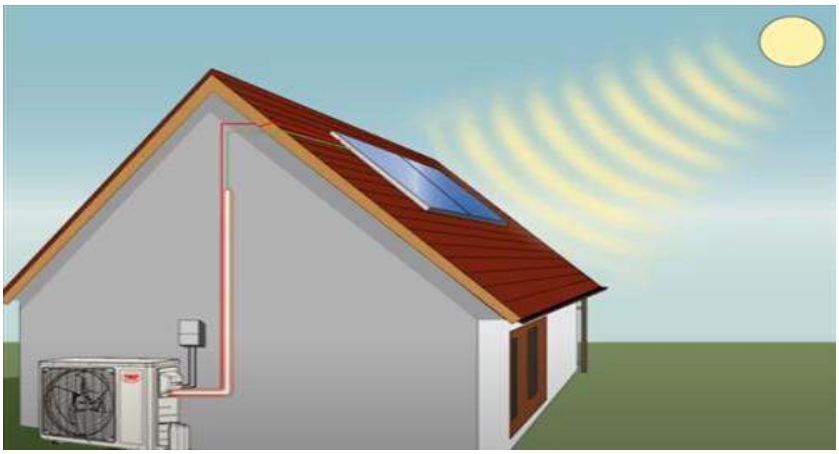


Figure 4.3: Hybrid split system infused with solar energy.

C. Duct free (Mini-split) :

A duct-free or mini-split system comes with a huge upfront cost and a list of benefits for particular needs and applications. It offers an amazing solution for areas where conventional ducted systems are not suitable. It can also be a great alternative to the existing ducted types of HVAC systems. [4]

Duct free units are usually installed directly into the particular areas of a home where heating and cooling are needed. You are allowed to have four indoor air handling units for each outdoor unit. It offers better independent control which makes it an appropriate choice for hotels or venues allowing tenants to control the individual temperature. [4]



Figure 4.4: Mitsubishi model duct free mini split system.

D. Packaged heating and air conditioning system:

A packaged HVAC system contains everything including the compressor, condenser and evaporator in a single unit which is placed on the roof or near the foundation. It fulfills both the cooling and heating needs of a house. [4]

Its compact size makes it an appropriate option for houses with a space crunch. This HVAC system is very much efficient and easy to maintain. As its heating system is not as powerful as other options, it is generally used in a warmer climate. [4]



Figure 4.5: Packaged heating and cooling unit system with return duct.

The split system is the most commonly used HVAC system whereas hybrids are preferred due to their energy-saving capacity. Similarly, Mini-split systems are a suitable choice for picky families or the service industry but the packaged heating is a perfect choice for large building offices or skyscrapers. [4]

4.3 HVAC system selection criteria:

System selection criteria can vary depending on the conditions and building data the HVAC engineers are offered, when a certain set of conditions in the building are met, the following criteria needs to be taken into consideration:

- **Realizability:** Depending on the building in question, manageability of creating the installation takes main priority over other criteria, once the building has no restrains nor conception difficulties that might intervene with the planning of the HVAC system, other factors can be taken into consideration.
- **Quality:** The quality of the system's main units as well as their components plays a huge role in the selection of the system to be of use, the branding, rigidity, availability and life span of the previously mentioned is of high priority to insure an efficient and long lasting setup.
- **Price range:** Price range differs depending on the construction enterprise or individual owner, which leads to the consideration of a price/quality ratio balance.
- **Time efficiency:** Depending on the client's needs and/or the HVAC engineer's schedule, time is of the essence as some projects are of urgent need of a system installation, which requires a system that cannot consume a long period of waiting time.

4.4. Project HVAC system comparison:

As previously mentioned, an HVAC system cannot be applied to a structure unless it is compatible with the previously given criteria, therefore, a comparison must be uphold, first in between system types to understand which type is more suitable for our project, then a comparison in between systems in the selected type.

4.4.1 Comparison between types:

As seen in HVAC system types (2.2), we can distinguish 4 main types:

- Heating and cooling split systems.
- Hybrid split system.
- Duct free mini split systems.
- Packaged heating and air conditioning systems.

According to the studies and calculations done beforehand in chapter 2 and chapter 3, our HVAC system will need to comply with a coling capacity of of at least **268695.86 W** and a heating capacity of at least **205467.35 W** since:

ΣDtot = 205467.35 **W**

ΣAPtot(t) = 268695.86 **W**

Therefore, our comparison will only include systems that have the ability to withstand the specified calorific need or more.

The comparison between all four types of HVAC systems is schematized in the following table:

HVAC System type	Criteria			
	Price estimation (DZD)	**Esthetics**	**Flexibility**	**Price/efficiency ratio**
Heating and cooling split systems	From 4000000.00 to 5000000.00	An outdated setting depending on the umbrella system which can be unpleasant to the room decoration.	Offers either heating or cooling separately and depending on at least two different cores.	-Very efficient for small buildings and fewer number of rooms (apartment, small house). -Less efficient for large buildings with multiple floors and multiple rooms.
Hybrid split system	From 2000000.00 to 2600000.00	Can cause decay to the paint of the exterior walls as well as ruins the esthetics of the façade.	Offers both heating and cooling.	-Very efficient for small buildings and fewer number of rooms (apartment, small house). -Less efficient for large buildings with multiple floors and multiple rooms.
Duct free mini split systems	From 2200000.00 to 3000000.00	-Not suitable due to the composition and utility of the system as it focuses mainly on single rooms. - Very likes to cause exterior wall paint decay.	Offers both heating and cooling.	-Very efficient for small buildings and fewer number of rooms (apartment, small house). -Less efficient for large buildings with multiple floors and multiple rooms.

Packaged heating and air conditioning systems	From 2200000.00 to 3200000.00	The main core is either on the rooftop or the basement with the duct concealed under the fake ceiling making it unnoticeable and therefore esthetically pleasant.	Offers both heating and cooling.	-Very efficient for large buildings with multiple floors and multiple rooms. -Less efficient for small buildings and fewer number of rooms (apartment, small house).

Table 4.1: Comparison table for the different types of HVAC systems.

After analyzing the data given in the table above, we come to the conclusion that the packaged heating and air conditioning systems are the most suitable for our project amongst all four types, where even though the system's price may vary drastically, it's esthetic appeal, flexibility, and efficiency make up for the most suitable system for a hotel such as our project.

Worth noting:

Traditionally, packaged heating and air conditioning systems (PHAC) with return air ducts require a certain height of ceiling, the suggested project has a height of (4.08; 3.54; 3.20; 3.20; 3.20) for the ground floor, first floor, second floor, third floor and fourth floor respectively, and thus, setting up a PHAC in the second, third and fourth floor would cause the fake ceiling to consume at least 0.5 meters of height, rendering the ceiling height below standard and therefore unpleasant.

However, we suggest the solution of creating a partial plaster fake ceiling immerging from the walls of the rooms in question in the second, third and fourth floor respectively, with a depth and width of 0.15 to 0.20 additional to the dimensions of the duct traversing it, serving as not only a carrier for the duct and outlets replacing the fake ceiling, but also adds an esthetic perspective to the decoration of the environment in addition to manageability for repairs in case of leakage, malfunction or breakdown without the need to ruin or remove the partial fake ceiling.



Figure 4.6: Example of a Partial plaster fake ceiling for duct use.

4.5 Packaged heating and air cooling unit selection:

After consideration of the criteria set ahead as well as the selection in between HVAC systems, the packaged heating and air cooling unit that will be put forth is the YLRA Air cooled heat pump scroll compressor which is an A class air handling unit powered by YORK, and following the Eurovent standard conditions.



Figure 4.7: The YORK YLRA air cooled heat pump scroll compressor. [C1]

4.5.1 Features and model selection:

YLRA are available in 6 models, from 0200 to 0330, with a nominal capacity range from 181 to 307 kW in cooling mode and from 200 to 327 kW in heating mode and up to 3.99 ESEER with EC fans. [C1]

Except for the fans all the units have the same configuration of base units (structure, electrical board, compressors and coils). Each model is available in the following acoustic versions:

• Basic Low Noise version (BLN):

These models are equipped with delta connected fans running at a fixed rpm and are fitted with compressor boxes to reduce noise emissions. [C1]

• Super Low Noise version (SLN):

Those models are equipped with special inverter fans driven by EC (electronic brushless type), fitted with a variable speed controller which allows the fans to operate at a very low rpm nonetheless, the chillers are supplied with compressor boxes and soundproof jackets on compressors reducing significantly the noise emissions. [C1]

The BLN model is also available in an EC version (developed for high seasonal efficiency) which has the same equipment as that of the standard BLN model, except that the units are equipped with special inverter fans driven by EC (electronic brushless type) motors with integrated electronic inverter, to ensure low energy consumption. [C1]

4.5.2 Options and accessories:

Depending on the model in use, the air handling unit (AHU) comes with a variety of accessories which are: [C1]

• ModBus protocol kit for BMS (standard).
• Lonwork protocol kit for BMS.
• Bacnet protocol kit for BMS.
• Soft start.
• Power factor correction capacitors.
• Compressors overload protection.
• Condensing control kit, down to -14 °C ambient temperature in cooling mode.
• Polar version (down to -18 °C ambient temperature in heating mode).
• Double set point.
• HP & LP manometers.
• E-coating Al/Cu condenser coils.
• Chiller grilles.
• Desuperheater.
• Optional hydro kits.
• Remote ON/OFF control.

• Remote keyboard panel.
• Sequencer unit, Spring isolators, Flow Switch and Water filter.

4.5.3 Nominal capacity and technical data:

As previously mentioned, the YLRA air cooled heat pump scroll compressor stretches between the 0200, 0230, 0260, 0280, 0300 and 0330 where each model has its own nominal capacity and characteristics whether it is a BLN version or SLN version which is summed up in the following table along with the technical data: [C1]

Nominal capacity

YLRA BLN versions	0200	0230	0260	0280	0300	0330
Cooling capacity (kW)	181.3	213.6	243.7	261.1	287.8	307.4
EER	2.93	2.92	2.91	2.88	2.92	2.97
Energy Efficiency Class	B	B	B	C	B	B
ESEER	3.6	3.71	3.71	3.65	3.6	3.64
EER (EC units)	2.97	2.96	2.95	2.91	2.96	3.02
ESEER (EC units)	3.71	3.83	3.83	3.78	3.71	3.71
Heating capacity (kW)	200.1	229	262.3	279.6	305.6	327.2
COP	3.22	3.23	3.21	3.20	3.27	3.21
COP (EC units)	3.28	3.27	3.26	3.25	3.27	3.26
Energy Efficiency Class	A	A	A	A	A	A
Sound power level (dBA) *	92	92	93	93	94	95
Sound pressure at 10 m (dBA) **	60	60	61	61	62	63
YLRA SLN versions	**0200**	**0230**	**0260**	**0280**	**0300**	**0330**
Cooling capacity (kW)	168.5	194.7	224	238.5	263.3	283.5
EER (EC units)	2.7	2.54	2.58	2.5	2.55	2.66
Energy Efficiency Class	C	D	D	D	D	D
ESEER (EC units)	3.86	3.99	3.95	3.93	3.86	3.79
Heating capacity (kW)	189.8	219.8	250.8	267.1	294.7	315
COP	3.27	3.27	3.25	3.24	3.26	3.25
Energy Efficiency Class	A	A	A	A	A	A
Sound power level (dBA) *	82	82	83	83	85	86
Sound pressure at 10 m (dBA) **	50	50	51	51	53	54

Table 4.2: Nominal Capacity of BLN and SLN versions of YLRA. [C1]

Technical data

YLRA BLN versions			0200	0230	0260	0280	0300	0330
Dimensions	Length	mm	3 500				4 550	
	Width	mm	2 150					
	Height	mm	2 600					
Operating weight (kg)			1 858	1 993	2 216	2 226	2 806	2 899
Additional weight EC versions (kg)			50	50	60	60	70	80
YLRA SLN versions			**0200**	**0230**	**0260**	**0280**	**0300**	**0330**
Dimensions	Length	mm	3 500				4 550	
	Width	mm	2 150					
	Height	mm	2 600					
Operating weight (kg)			1 908	2 043	2 276	2 286	2 876	2 979

Table 4.3: Technical Data of BLN and SLN versions of YLRA. [C1]

4.5.4 Dimensions and hydraulic connections:

YLRA from 0200 to 0280:

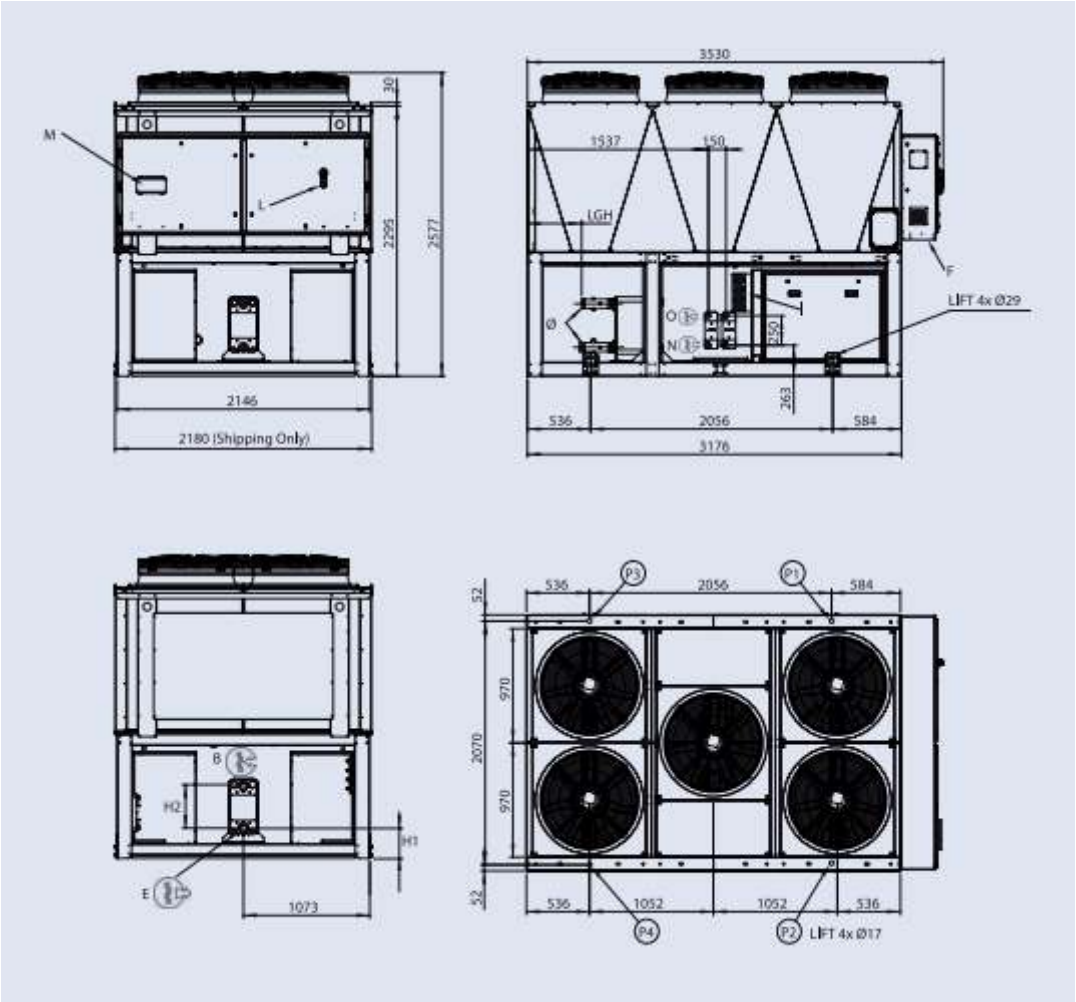


Figure 4.8: Schematics of the YLRA 0200 to 0280 models. [C1]

Notes:

All dimensions are in mm, drawings not a scale.
B, E: Water connection gas M.
F: Electrical power supply.
I: Gauge kit.
L: Main switch.
M: Control keypad/display.
N: Water inlet Ø1" gas M.
O: Water outlet Ø1" gas M.
P1, P2, P3, P4: AVM position.

YLRA from 0300 and 0330:

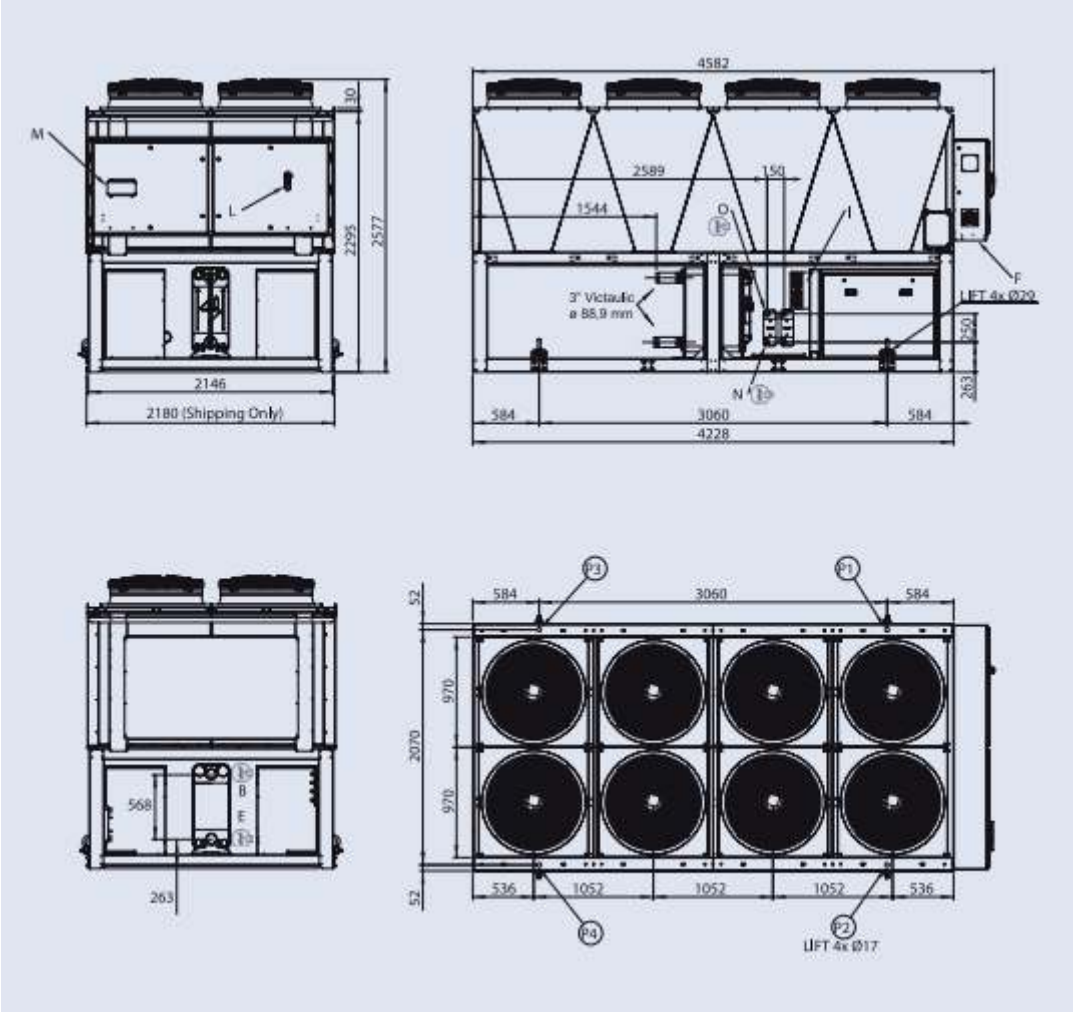


Figure 4.9: Schematics of the YLRA 0300 and 0330 models. [C1]

Notes:

All dimensions are in mm, drawings not a scale.
B, E: Water connection 3-gas M Ø88.9 mm.
F: Electrical power supply.
I: Gauge kit.
L: Main switch.
M: Control keypad/display.
N: Water inlet Ø1" gas M.
O: Water outlet Ø1" gas M.
P1, P2, P3, P4: AVM position.

4.5.5 Model selection:

After comparing between the nominal capacity and the technical date of the previously presented models, we have settled to use the **SNL version of YLRA air cooled heat pump scroll compressor model 0300**, mainly for its heating and cooling capacity which are very much compatible with our own project demands.

4.6 Diffuser units and variable air volume:

4.6.1 Definition of diffuser in HVAC:

In HVAC systems, a diffuser disperses pressurized air exiting heating ventilation and air conditioning ductwork and provides a decorative finish over the grill box hole that is holding the ductwork in place. Without an air diffuser, the air exiting the HVAC system would travel in a straight direction. This causes a concentration of conditioned air in one section of the room while the rest of the room remains unconditioned. The direction that a vent diffuser sends flowing air depends on the direction of the diffuser fins. []



Figure 4.10: Typical square wall air diffuser.

In our project, we will rely on a ductwork system based on Variable Air Volume (VAV) Terminals and diffusers. [C2]

4.6.2 Definition of Variable air volume:

Variable Air Volume (VAV) is a type of heating, ventilating, and/or air-conditioning (HVAC) system. The simplest VAV system incorporates one supply duct that, when in cooling mode, distributes approximately 13 °C supply air. Because the supply air temperature, in this simplest of VAV systems, is constant, the air flow rate must vary to meet the rising and falling heat gains or losses within the thermal zone served. [5]

The VAV Zone Controller has a built-in actuator and maintains zone temperature by operating the terminal fan and regulating the flow of conditioned air into the space. Buildings with diverse loading conditions can be supported by controlling the local terminal's supplemental heat. The VAV Zone Controller provides dedicated control functions for single duct, parallel fan box terminals and series fan box terminals with modulating heat, up to 2 stages of ducted heat, or combination baseboard and ducted heat. [5]



Figure 4.11: Typical variable air volume system box. [5]

4.6.3 Variable air volume model selection:

VAV systems although similar in conception primarily they vary in question of brands and availability in the market, with some models offer higher noise control and better air temperature precision.

Since our duct installation ends and outputs will be on the perimeter of the rooms in question, a wall diffuser model will be needed in order to insure a sufficient and cohesive air flow pattern, therefore, we have settled for the **TROX TECHNIK AURANOR** brand, a German-Scandinavian trademark specialized in HVAC equipment and air distribution products, from which we chose the **PEGASUS** diffuser, precisely the **PEGASUS-S** model for the output of our room air distribution.



Figure 4.12: The PEGASUS-S (left) and the PEGASUS-B (right) VAV diffuser. [C2]

4.6.4 Application, function and design:

- **Application:**
  Pegasus is a wall diffuser unit with VAV function. It is used as a volume flow controller unit in demand controlled ventilation systems. Pegasus has very good induction, making it ideal for variable air flow rates. [C2]
- **Function:**
  Pegasus has a built-in VAV controller for the demand controlled adjustment of air flow rates. The damper solution will choke the pressure at high flow rates and maintain a low sound level. This may reduce the need for additional dampers and sound attenuators in a duct system. Pegasus is supplied with Belimo MP-Bus, LON, Modbus or Belimo KNX for direct BUS communication with SD systems. [C2]
- **Design:**
  Pegasus is a complete measuring and control unit for the demand control of air flow rates in ventilation systems. The measuring station measures the differential pressure using measuring rods integrated into the unit. The unit is equipped with an LHV-D3 VAV controller from Belimo. The controller's specifications are provided in the table below. Full technical documentation can be downloaded from www.belimo.eu. Pegasus has a removable front panel with a choice of LØV or Opus design. LØV has two different designs, long (L) or short (K) throw length. If the Opus

design with turnable plastic nozzles is selected, (V) is the designation for the variable throw length. [C2]

Actuator	LHV-D3-MP/MOD/LON
Operating voltage	AC 24 V 50/60 Hz, DC 24 V
Power consumption	2.5 W
Dim. power	4.5 VA (max. 8 A @5 ms)

Table 4.4: Technical specification, Belimo VAV controller. [C2]

Pegasus LØV-S-K/L	m^3/h (open)		
Dim.	25 dB(A)	30 dB(A)	35 dB(A)
125	169	205	252
160	274	335	407
200	342	410	490
250	518	630	763

Pegasus LØV-S-K/L	m^3/h (75 Pa)		
Dim.	25 dB(A)	30 dB(A)	35 dB(A)
125	166	209	234
160	238	320	396
200	288	385	486
250	439	630	738

Table 4.5: Quick selection PEGASUS LØV-S-K/L measurements. [C2]

Pegasus	(m^3/h)	
Dim.	Minimum	Maximum
125	26	265
160	43	434
200	70	700
250	106	1060

Table 4.6: Adjustment and regulation range for VAV, afr in m^3/h. [C2]

Dim.	D	BF	HF	BK	BK2	HK	DB	DS	A	B	Weight [kg]
125	124	510	205	470	700	173	700	407	82	145	11
160	159	560	255	520	700	223	700	442	100	163	13
200	199	580	295	540	700	263	700	482	120	183	14.5
250	249	640	387	800	800	337	725	532	145	208	19

Table 4.7: Dimensions and weight for PEGASUS. [C2]

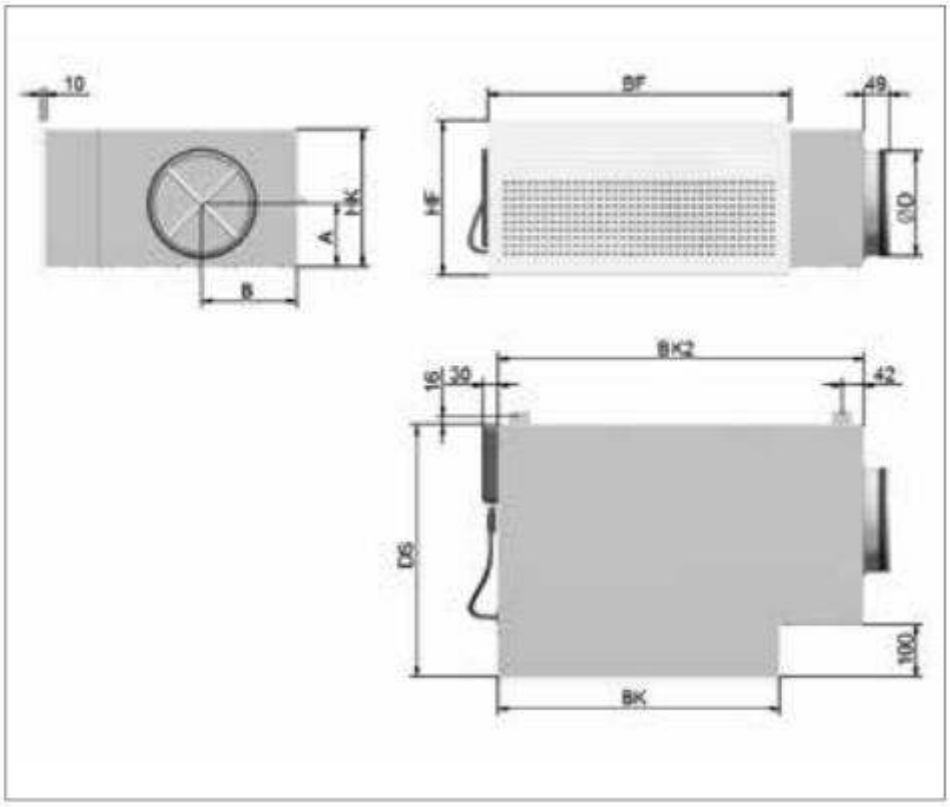


Figure 4.13: Dimensioned sketch, of the PEGASUS-S. [C2]

- **Diffusion pattern of the PEGASUS-S:**

The pattern of air diffusion of the PEGASUS-S model diffuser is flexible between long throw range and short throw range, which is better illustrated in the following figures (4.13) and (4.14): [C2]

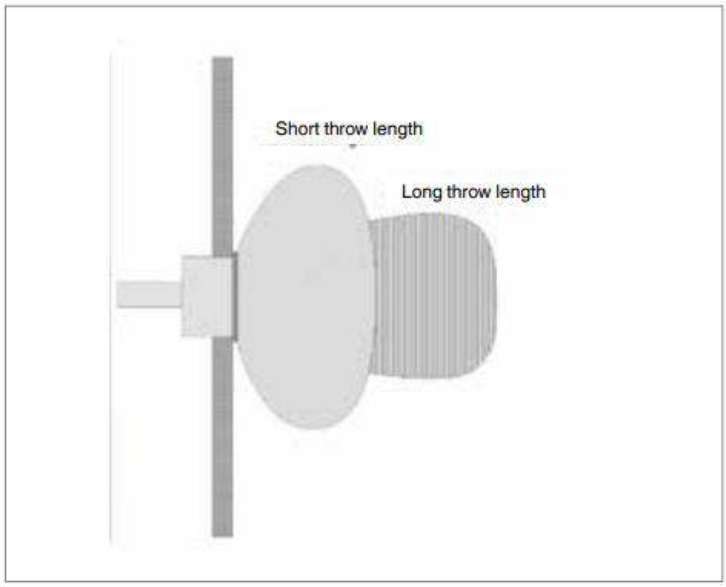


Figure 4.14: Example of the flow pattern of the PEGASUS. [C2]

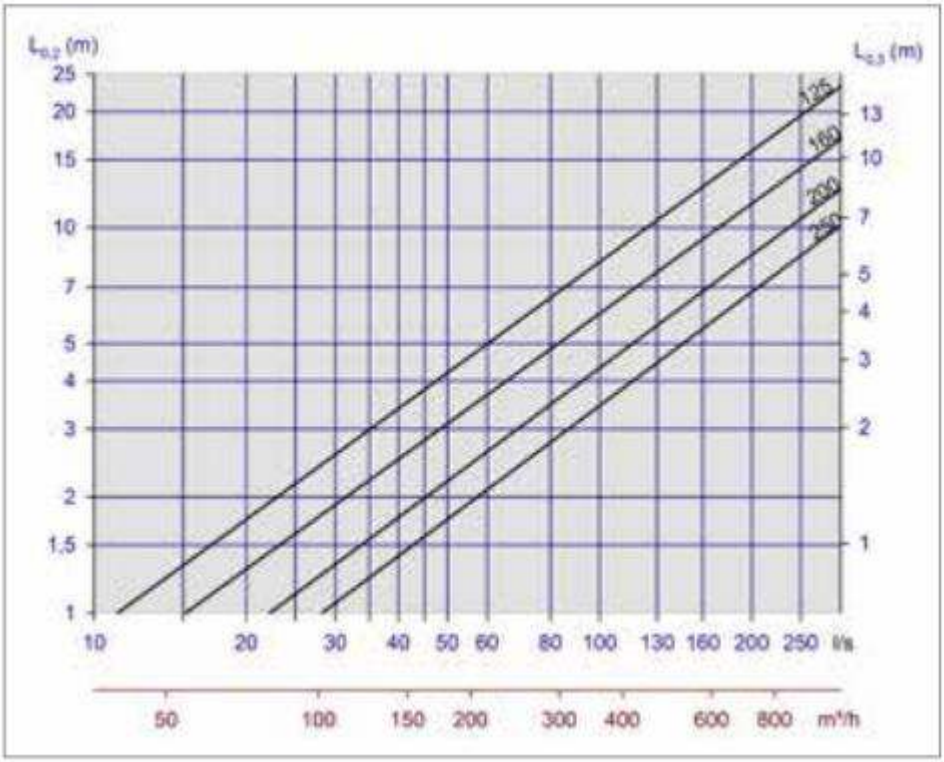


Figure 4.15: Diagram of the throw length of the PEGASUS. [C2]

- **Velocity of the PEGASUS-S:** [C2]

The velocity distribution of the PEGASUS-S descends during the air flow inside the room in a complete circle from v>0.25 at the moment of diffusion up until it reaches v<0.15 with v being the velocity of the air flow, as schematized in the following figure (4.14):

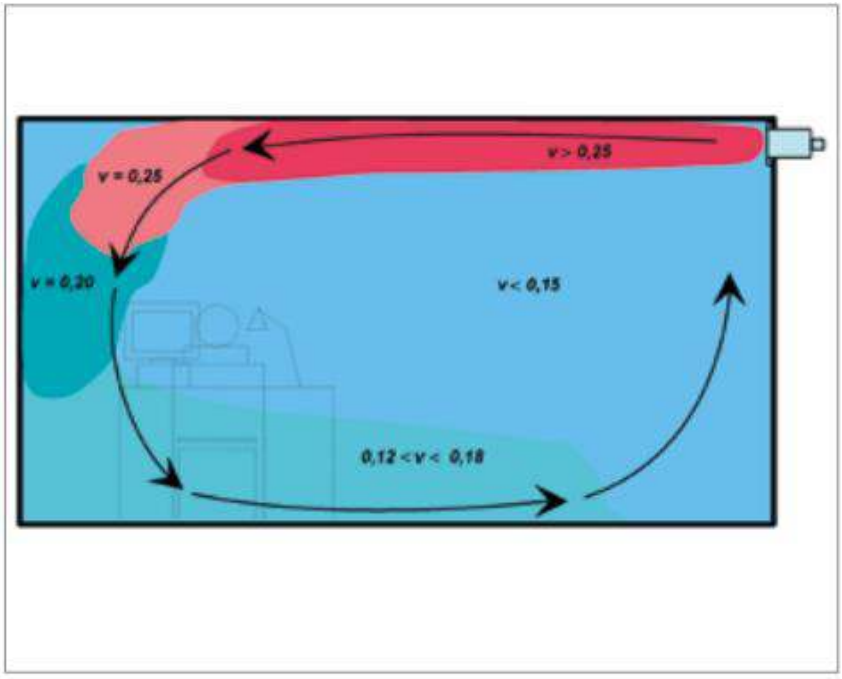


Figure 4.16: Example of velocity distribution. [C2]

- **Mounting and adjustment of the PEGASUS-S:**

Pegasus is installed as a rear diffuser in the rear of the unit using a suspension bracket and a threaded rod or strap, see figures 8 and 9. The plenum box is installed in the cutout and screwed to the studding. The adjustable installation frame is then pushed in and screwed to the plenum box. Finally, the front is pressed into position. [C2]

For which concerns the adjustment, Pegasus uses the Belimo PC-tool or ZTH-GEN to make the necessary settings. [C2]

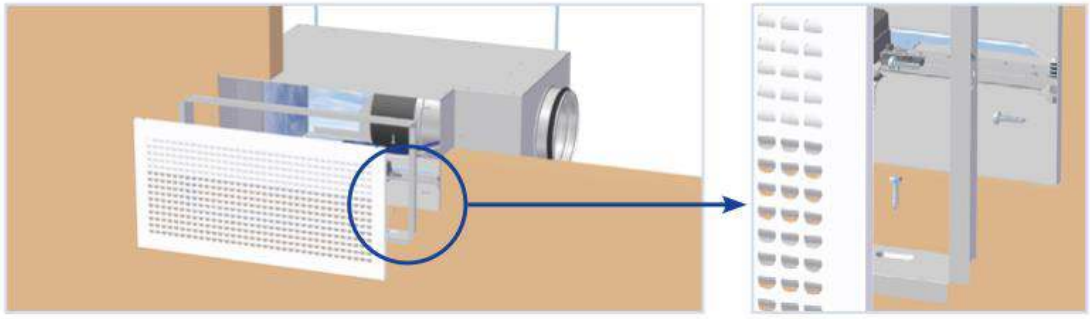


Figure 4.17: Mounting of the PEGASUS-S. [C2]

- **Duct installation of the PEGASUS-S:**

The duct work of the PEGASUS-S can be schematized in the following figure (4.16):

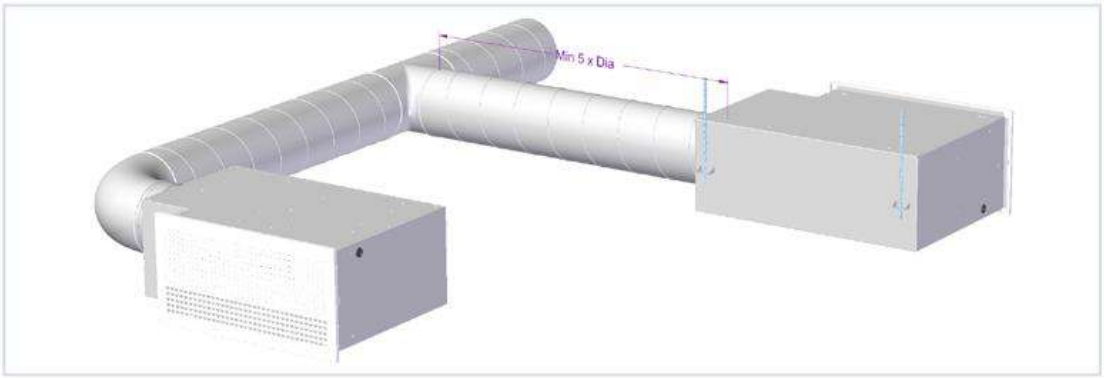


Figure 4.18: Example of duct installation of the PEGASUS-S. [C2]

Chapter V

Duct sizing and efficiency calculation

5.1 Ductwork sizing, calculation and design for efficiency:

There are many different methods used to design HVAC systems, the most common ways being:

- Velocity reduction method: (Residential or small commercial installations)
- Equal friction method: (Medium to large sized commercial installations)
- Static regain: Very large installations (concert halls, airports and industrial)

We're going to focus on the equal friction method as it is the most common method used for commercial HVAC systems

5.2 Air distribution:

-The first thing that needs to be applied is finding which the biggest load is as we need to size the system to be able to operate at the peak demand. The cooling load is usually the highest, as it is in this case (268.696 KW).

Now we need to convert the cooling loads of each room into volume flow rates but to do that we first need to convert this to mass flow rate so we use the formula:

$$\dot{m} = Q / (cp \times \Delta t). \qquad \text{V.1}$$

$\dot{m}$: mass flow rate (kg/s).

Q: cooling load of the room (kW).

Cp: specific heat capacity of the air (kJ/kg.K).

ΔT: temperature difference between the designed air temperature and the design return temperature.

- It is worth mentioning that we will use a Cp of 1.026 (kJ/kg.k) as standard and the delta T should be less than 10°C, therefore, a temperature of 8 (°C) will be used.

-Now we can convert these into volume flow rates,to do that we need the specific volume or density of the air. We will specify 21°C and assume atmospheric pressure of 101.325 kPa, in order to get the density of air being 1.2 kg/m3 and specific volume of 0.83m3/kg.

Now that we have that we can calculate the volume flow rate using the formula:

$$\dot{V} = \dot{m} \times v \qquad \text{V.2}$$

$\dot{V}$: volume flow rate (m^3/s).

$\dot{m}$: mass flow rate (kg/s).

V: specific volume(m^3/kg).

Before we advance any further, we need to consider a set of factors which will play a big role in the overall efficiency of the system.

5.2.1 Design consideration:

A. The shape of the ductwork.

Ductwork comes in round, rectangular and flat-oval shape. Round duct is the most energy efficient type,but we are forced to go with rectangular duct because we are limited by the height of the duct(max 0.6m).

B. The material being used for the ducts.

The material used for the ducts and the roughness of this material as this causes friction. For example, if we had two ducts, with equal dimensions, volume flow rate and velocity, the only difference is the material. One is made from standard galvanized steel while the other is made offiberglass, the pressure drop over a 10m distance for this example, is around 11 Pa for the galvanized steel and 16 Pa for the fiberglass, therefore we will use a galvanized steel ducts.

C. Dynamic losses caused by the fittings.

We want to use the smoothest fittings possible for energy efficiency. For example use long radius bends rather than right angles as the sudden change in direction wastes a huge amount of energy.

5.3Air distribution ducts:

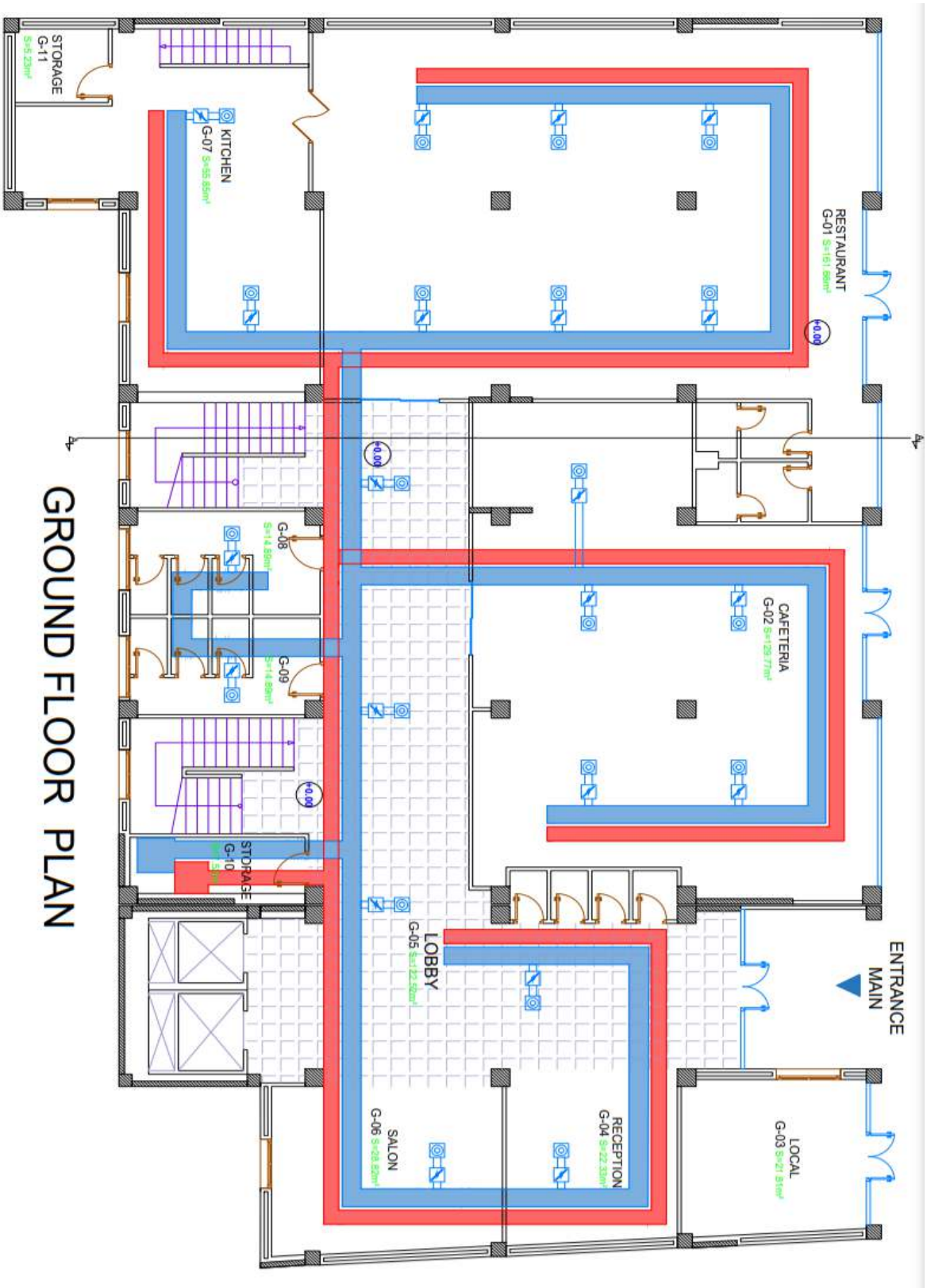


Figure 5.1: Ground floor ducts location.

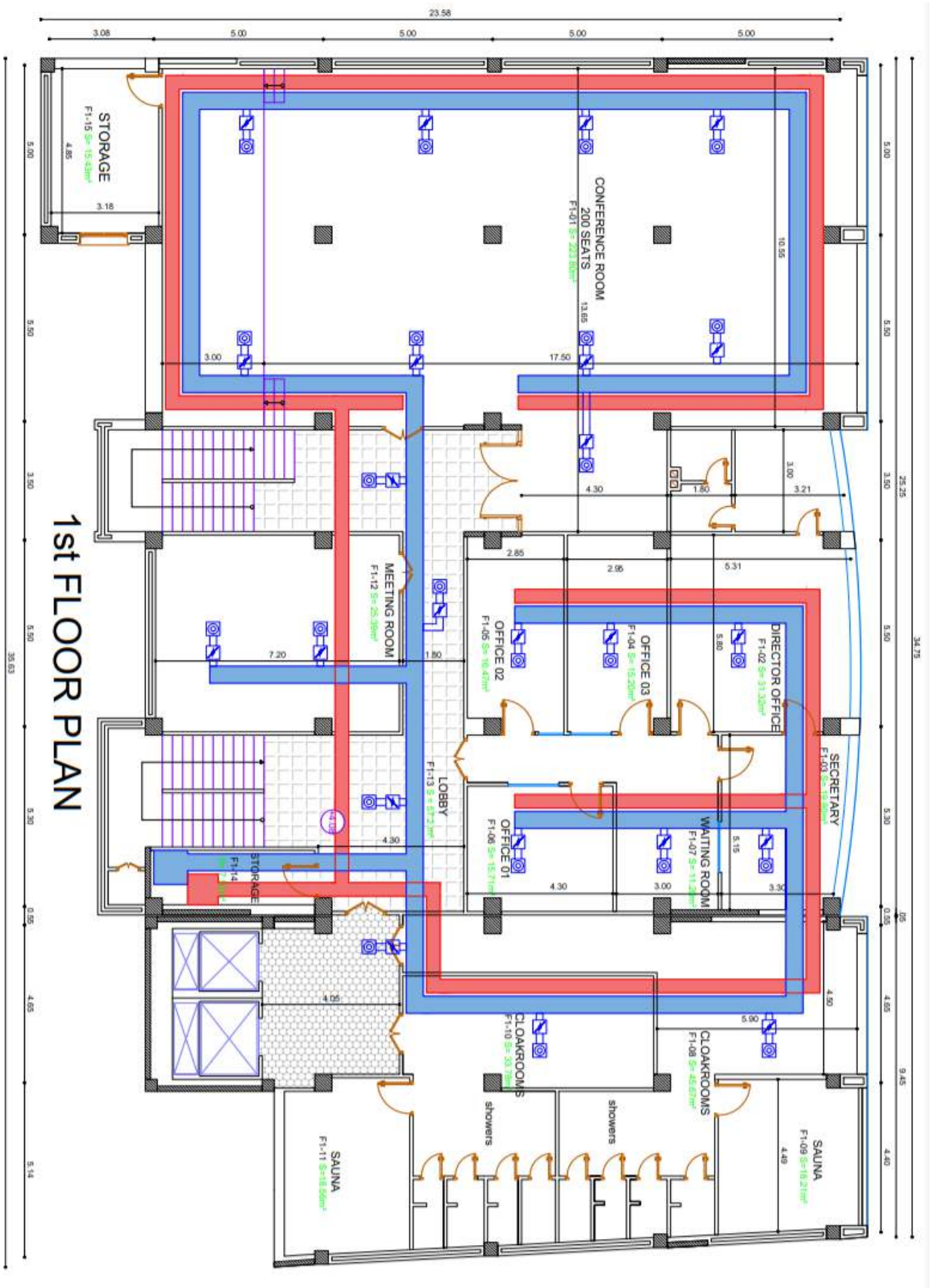


Figure 5.2: First floor ducts location.

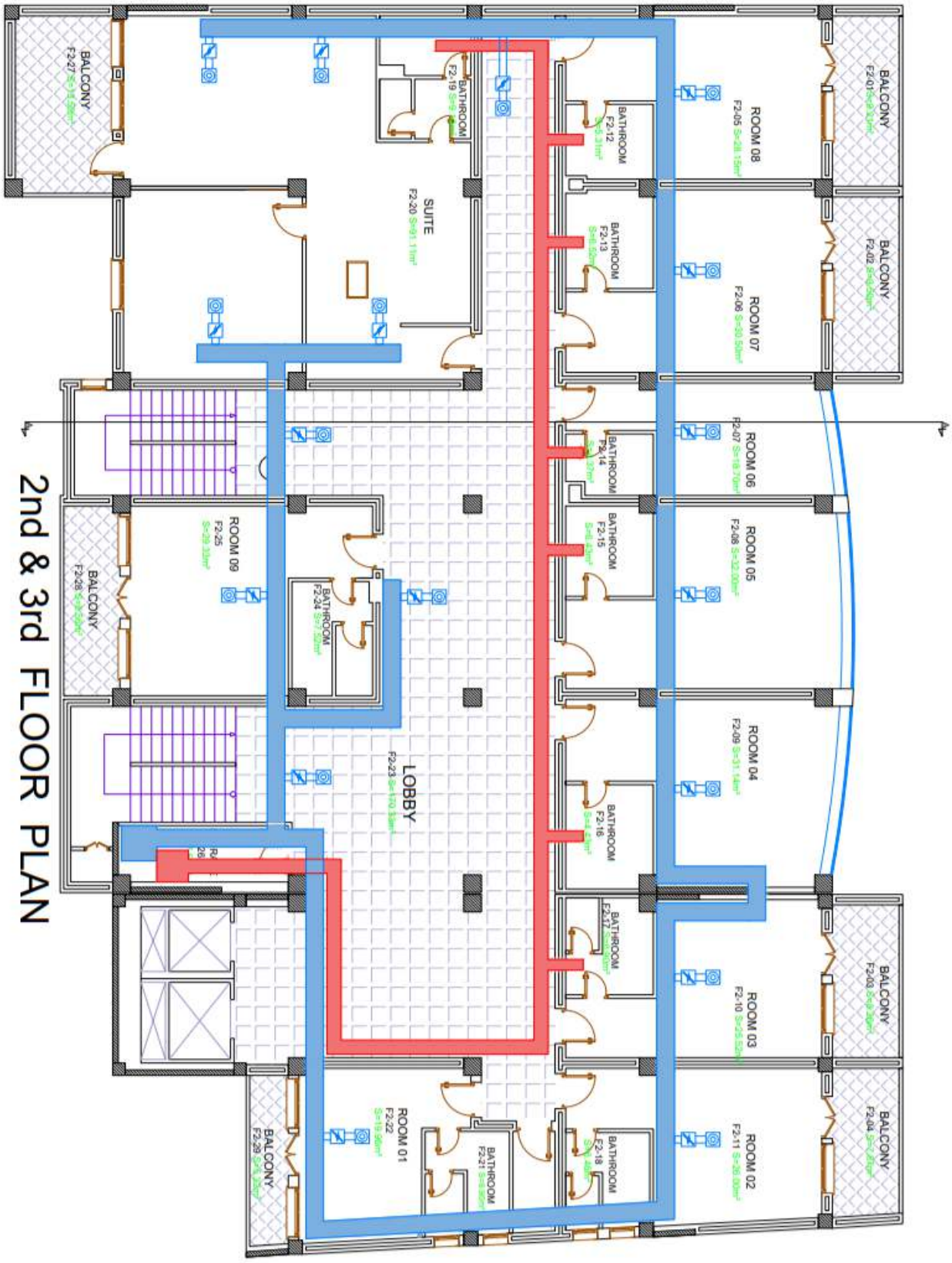


Figure 5.3: Second and third floor ducts location.

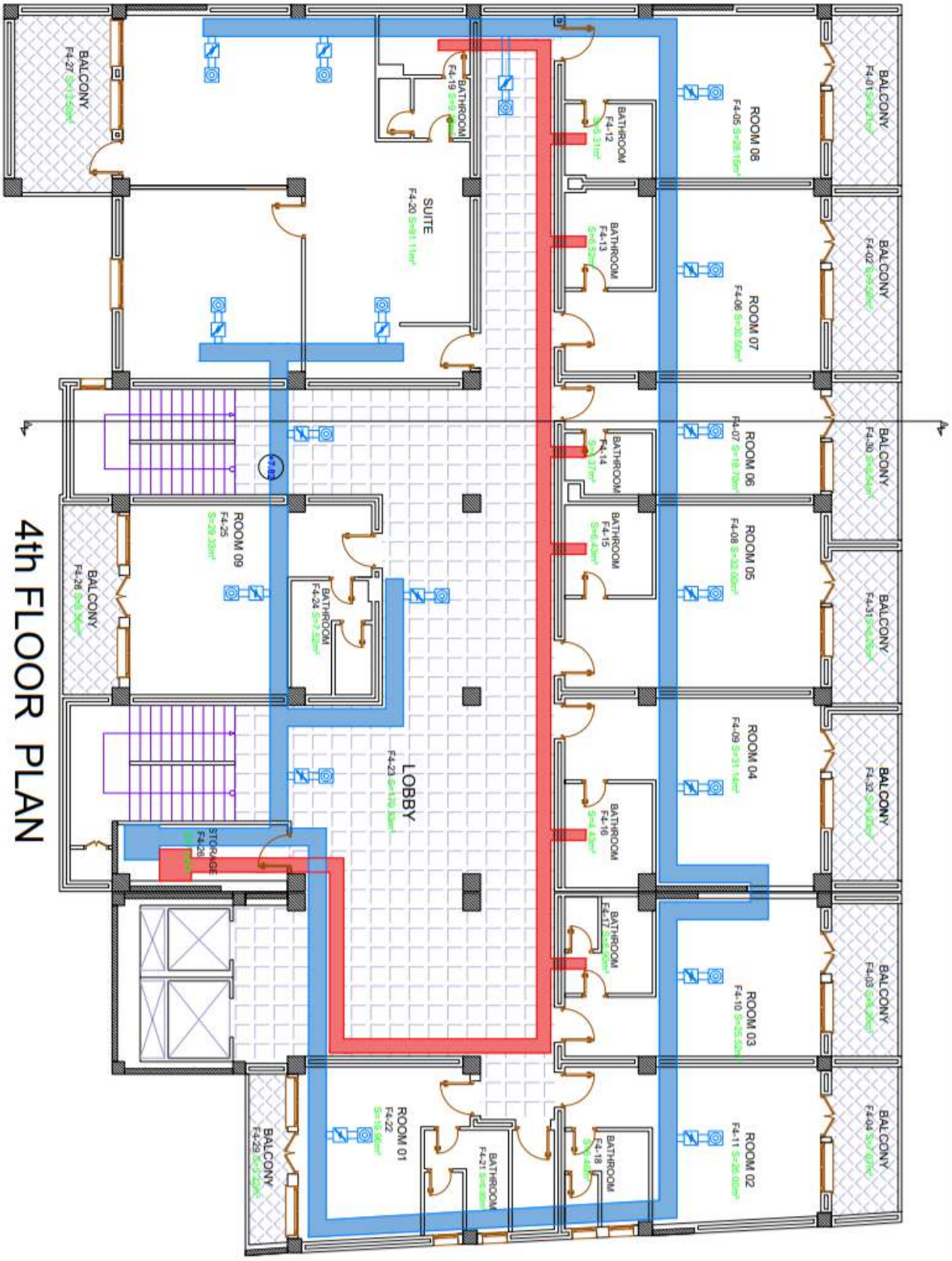


Figure 5.4: Fourth floor ducts location.

5.4 Duct sizing

First we need to label every section of ductwork as well as the fittings with a letter.

Now we want to make a table with the rows labeled so that each duct and fitting is provided its own row, if the air stream splits such as with a Tee section, then we need to include a line for each direction.We can start to fill some of the data in; we can first include the volume flow rates for each of the branches, then, we can start to size the main ducts. To do this we start at the main duct which is furthest away, after that, we just add up the volume flow rates for all the branches downstream of this.

To size the ducts you're going to need a duct sizing chart.

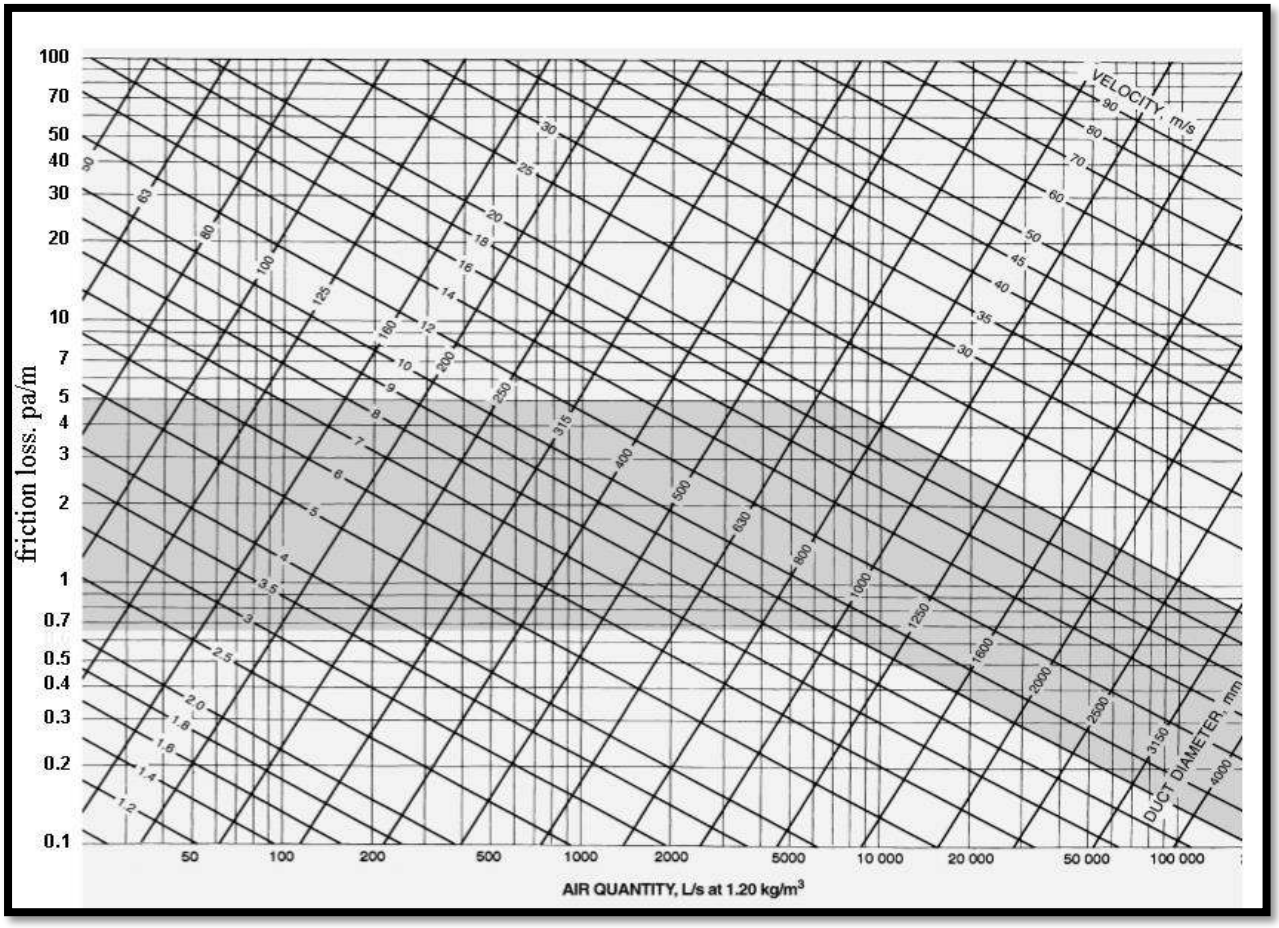


Figure 5.5: Friction Chart for Round Duct (ρ = 1.20 kg/m3 and ε = 0.09 mm)

These charts hold a lot of information. We can use them to find the pressure drop per meter, the air velocity, the volume flow rate and also the size of the ductwork. the vertical lines are for pressure drop per meter of duct. The horizontal lines are for volume flow rate. The downward diagonal lines are for velocity, the upward diagonal lines are for duct diameter.

Now find the total duct losses for each duct and branch, by multiplying the duct length by the pressure drop per meter.

5.5 Ductwork fittings Sizing

For the first fitting, we will look at the 90° bends

For this, we look up our loss coefficient for the bend from the manufacturer (2001 ASHRAE Fundamentals Handbook SI).

CD3-9 Elbow, 5 Gore, 90 Degree, r/D = 1.5

D, mm	75	150	230	300	380	450	530	600	690	750	1500
C_o	0.51	0.28	0.21	0.18	0.16	0.15	0.14	0.13	0.12	0.12	0.12

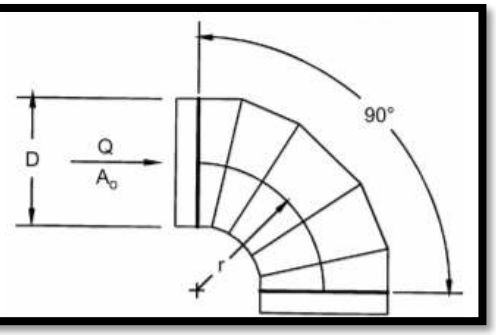


Table 5.1: Loss coefficient for the 90° elbow bend.

We then need to calculate the dynamic loss caused by the bend changing the direction of flow. For that we use the formula:

Bend loss = $C_o \times \rho \times v^2 \div 2$ V.3

- **C_o:** Loss coefficient.
- Rho: The density of the air.
- V: The velocity.

-The next fitting we'll look at is the tee which connects the main duct to the branches. Now for this we need to consider that the air is moving in two directions, straight through and also turning off into the branch so we need to perform a calculation for both directions.

If we look at the air travelling straight though first, we find the velocity ratio first using the formula velocity out divided by velocity in.

Then we perform another calculation to find the area ratio, this uses the formula diameter out squared divided by diameter in squared.

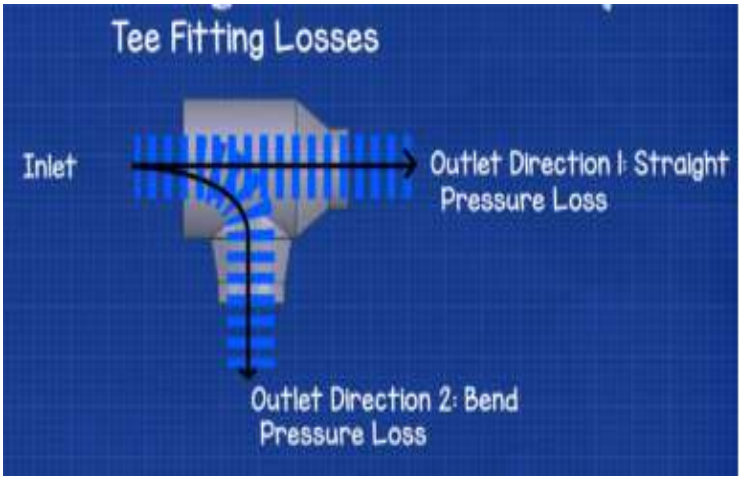


Figure 5.6: Duct tee pressure loss.

Now we look up the fitting that we are using from the manufacturer or the industry body.

SD5-9 Tee, Diverging

	C_b Values								
	Q_b/Q_c								
A_b/A_c	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.1	1.20	0.62	0.80	1.28	1.99	2.92	4.07	5.44	7.02
0.2	4.10	1.20	0.72	0.62	0.66	0.80	1.01	1.28	1.60
0.3	8.99	2.40	1.20	0.81	0.66	0.62	0.64	0.70	0.80
0.4	15.89	4.10	1.94	1.20	0.88	0.72	0.64	0.62	0.63
0.5	24.80	6.29	2.91	1.74	1.20	0.92	0.77	0.68	0.63
0.6	35.73	8.99	4.10	2.40	1.62	1.20	0.96	0.81	0.72
0.7	48.67	12.19	5.51	3.19	2.12	1.55	1.20	0.99	0.85
0.8	63.63	15.89	7.14	4.10	2.70	1.94	1.49	1.20	1.01
0.9	80.60	20.10	8.99	5.13	3.36	2.40	1.83	1.46	1.20
	C_s Values								
	Q_s/Q_c								
A_s/A_c	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.1	0.13	0.16							
0.2	0.20	0.13	0.15	0.16	0.28				
0.3	0.90	0.13	0.13	0.14	0.15	0.16	0.20		
0.4	2.88	0.20	0.14	0.13	0.14	0.15	0.15	0.16	0.34
0.5	6.25	0.37	0.17	0.14	0.13	0.14	0.14	0.15	0.15
0.6	11.88	0.90	0.20	0.13	0.14	0.13	0.14	0.14	0.15
0.7	18.62	1.71	0.33	0.18	0.16	0.14	0.13	0.15	0.14
0.8	26.88	2.88	0.50	0.20	0.15	0.14	0.13	0.13	0.14
0.9	36.45	4.46	0.90	0.30	0.19	0.16	0.15	0.14	0.13

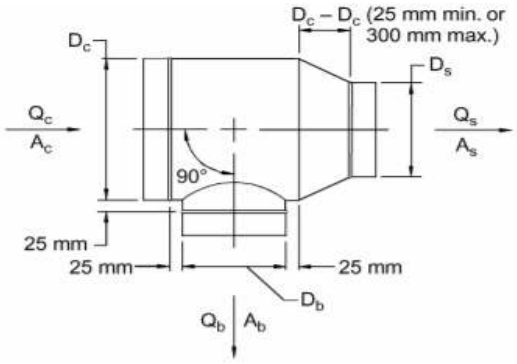


Table 5.2: loss coefficient for the Tee fitting.

Now we calculate the dynamic loss for the straight path through the tee, using the formula:

Tee fitting loss = $C_o \times \rho \times v^2 \div 2$ V.4

Then we can calculate the dynamic loss for the air which turns into the bend. For this we use the same formulas as before. Velocity out divided by velocity in to find our velocity ratio.

Then we find the area ratio using the formula diameter out squared divided by diameter in squared. We take our values from our table.

5.6 Pressure drop calculation and duct sizing results:

5.6.1 Ground floor:

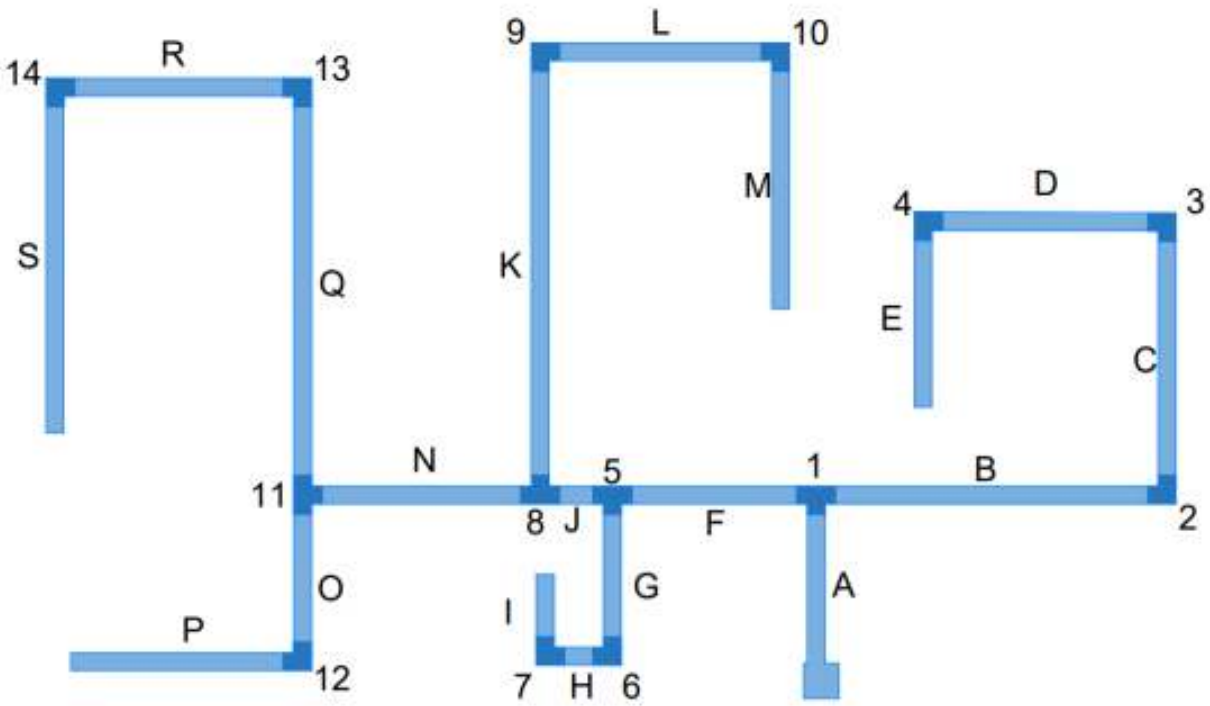


Figure 5.7: Ground floor duct and fitting labels.

ID	TYPE	Volume Flow Rate(m^3/s)	ΔP (Pa/m)	Velocity (m/s)	Diameter (m)	Lenght (m)	Duct Loss (pa)	Direction	Fitting loss (pa)
A	Duct	5.266	0.19	5	1.1	4.2	0.798		
1	Tee							A-B A-F	5.07 1.723
B	Duct	0.881	0.19	3.25	0.6	8.83	1.677		
2	Bend							B-C	0.823
C	Duct	0.390	0.19	2.7	0.43	6.6	1.254		

3	Bend							C-D	0.656
D	Duct	0.273	0.19	2.5	0.38	5.8	1.102		
4	Bend							D-E	0.6
E	Duct	0.217	0.19	2.3	0.34	4.7	0.893		
F	Duct	4.384	0.19	4.7	1.05	4.66	0.885		
5	Tee							F-G F-J	3.97 1.65
G	Duct	0.230	0.19	2.35	0.35	3.45	0.655		
6	Bend							G-H	0.563
H	Duct	0.230	0.19	2.35	0.35	0.8	0.152		
7	Bend							H-I	0.563
I	Duct	0.230	0.19	2.35	0.35	1.76	0.334		
J	Duct	4.153	0.19	4.6	1.03	0.8	0.152		
8	Tee							J-K J-N	7.018 1.442
K	Duct	1.642	0.19	3.8	0.68	11.4	2.166		
9	Bend							K-L	1.039
L	Duct	1.642	0.19	3.8	0.68	5.7	1.083		
10	Bend							L-M	1.039
M	Duct	1.642	0.19	3.8	0.68	6.7	1.273		
N	Duct	2.511	0.19	4.3	0.88	5.6	1.064		
11	Tee							N-Q N-O	1.156 4.48
O	Duct	0.543	0.19	2.85	0.48	3.6	0.684		
12	Bend							O-P	0.731
P	Duct	0.543	0.19	2.85	0.48	6	1.14		
Q	Duct	1.967	0.19	3.85	0.77	10.4	1.976		
13	Bend							Q-R	1.066
R	Duct	1.967	0.19	3.85	0.77	5.9	1.121		
14	Bend							R-S	1.066
S	Duct	1.967	0.19	3.85	0.77	9.2	1.748		

Table 5.3: Ground floor pressure drop results.

5.6.2 First floor:

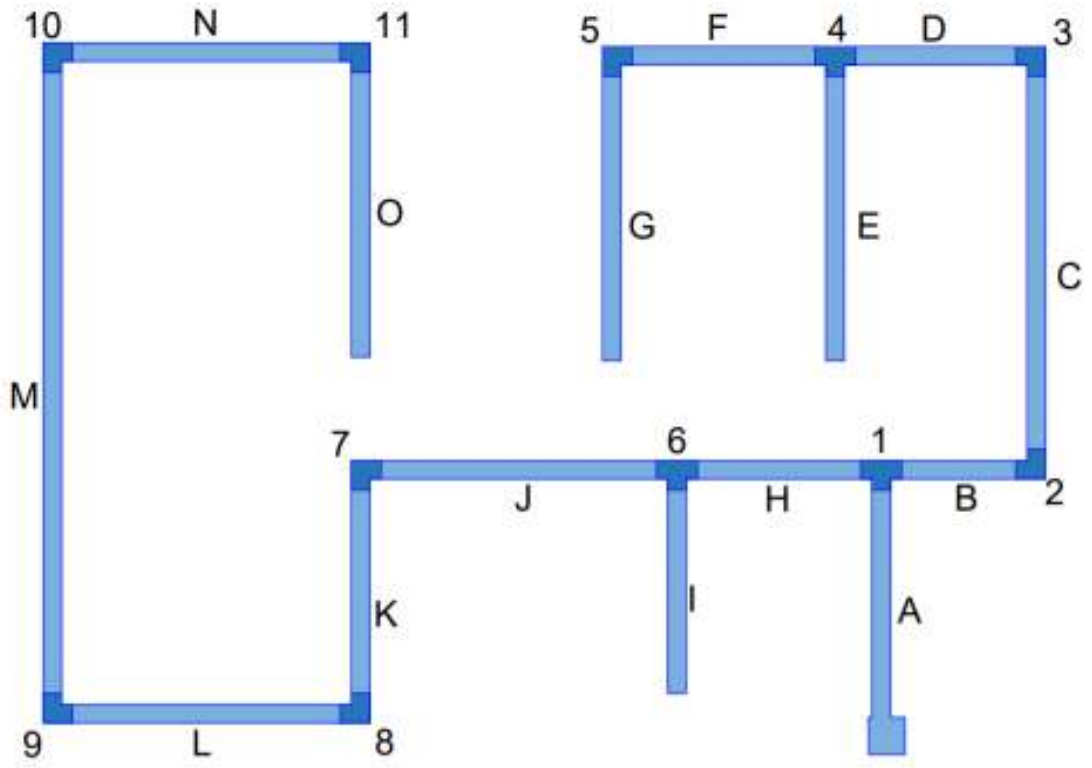


Figure 5.8: First floor duct and fitting labels.

ID	TYPE	Volume Flow Rate(m^3/s)	ΔP (Pa/m)	Velocity (m/s)	Diameter (m)	Lenght (m)	Duct Loss (pa)	Direction	Fitting loss (pa)
A	Duct	6.51	0.26	6	1.1	6.15	1.599		
1	Tee							A-B A-H	13.12 2.109
B	Duct	2.645	0.26	4.7	0.83	3.1	0.806		
2	Bend							B-C	1.59
C	Duct	2.645	0.26	4.7	0.83	10.1	2.626		
3	Bend							C-D	1.45
D	Duct	2.034	0.26	4.5	0.74	4.35	1.128		
4	Tee							D-E D-F	10.39 1.12
E	Duct	1.006	0.26	3.8	0.57	7.7	2.002		
F	Duct	1.027	0.26	3.8	0.59	4.95	1.287		

5	Bend							F-G	1.12
G	Duct	1.027	0.26	3.8	0.59	7.7	2.002		
H	Duct	3.859	0.26	5.2	0.95	4.45	1.157		
6	Tee							H-I H-J	4.44 2.028
I	Duct	0.366	0.26	2.9	0.38	5.5	1.43		
J	Duct	3.494	0.26	5.1	0.9	7.5	1.14		
7	Bend							J-K	1.728
K	Duct	3.017	0.26	4.9	0.85	5.5	1.43		
8	Bend							K-L	1.728
L	Duct	3.017	0.26	4.9	0.85	7.25	1.885		
9	Bend							L-M	1.728
M	Duct	3.017	0.26	4.9	0.85	16.8	4.368		
10	Bend							M-N	
N	Duct	3.017	0.26	4.9	0.85	7.25	1.885		
11	Bend							N-O	1.728
O	Duct	3.017	0.26	4.9	0.85	7.7	2.002		

Table 5.4: First floor pressure drop results.

5.6.3 Second floor:

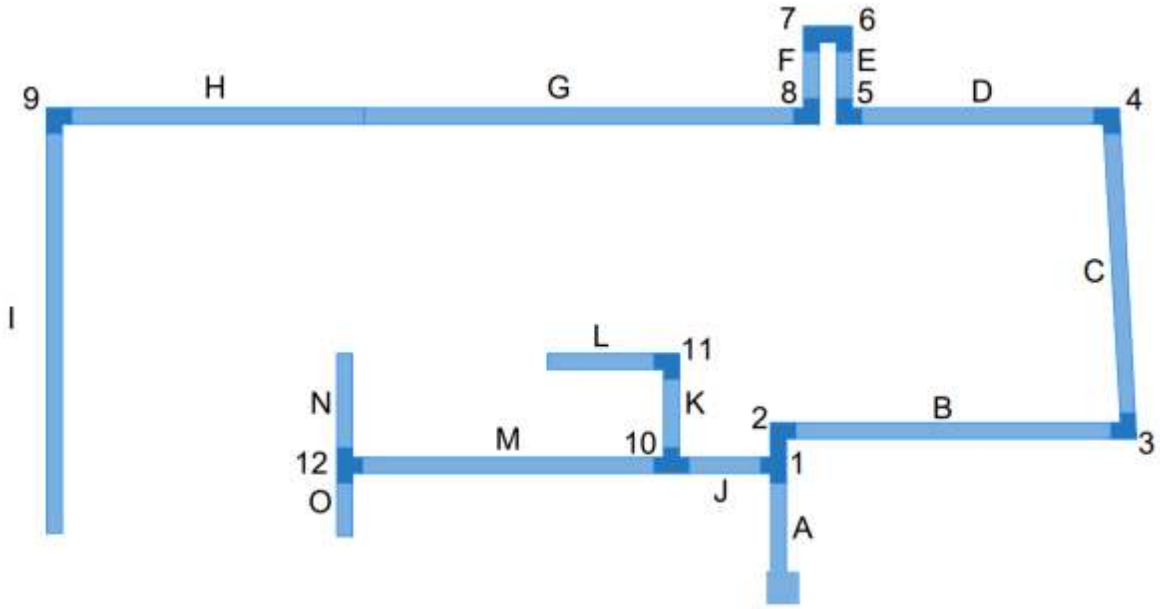


Figure 5.9: Second floor duct and fitting labels.

ID	TYPE	Volume Flow Rate(m^3/s)	ΔP (Pa/m)	Velocity (m/s)	Diameter (m)	Lenght (m)	Duct Loss (pa)	Direction	Fitting loss (pa)
A	Duct	4.711	0.32	6	1	2.85	0.912		
1	Tee							A-2 A-J	2.35 11.15
2	Bend							1-B	2.178
B	Duct	3.375	0.32	5.5	0.9	10.05	3.36		
3	Bend							B-C	2.099
C	Duct	3.038	0.32	5.4	0.85	8.91	2.851		
4	Bend							C-D	2.099
D	Duct	3.038	0.32	5.4	0.85	7.4	2.368		
5	Bend							D-E	1.8
E	Duct	2.429	0.32	5	0.75	1.5	0.48		
6	Bend							E-7	1.8
7	Bend							6-F	1.8
F	Duct	2.429	0.32	5	0.75	1.5	0.48		
8	Bend							F-G	1.8
G	Duct	2.429	0.32	5	0.75	13.7	4.384		
H	Duct	0.926	0.32	4	0.55	9.3	2.976		
9	Bend							H-I	0.691
I	Duct	0.306	0.32	3.1	0.35	12.7	4.064		
J	Duct	1.336	0.32	4.4	0.6	2.3	0.736		
10	Tee							J-M J-K	1.31 4.32
K	Duct	0.223	0.32	2.8	0.32	2.2	0.704		
11	Bend							K-L	0.564
L	Duct	0.223	0.32	2.8	0.32	3.4	1.088		
M	Duct	1.113	0.32	4.1	0.55	9.3	2.976		
12	Tee							M-N M-O	3.73 3.73
N	Duct	0.153	0.32	2.6	0.27	3	0.96		
O	Duct	0.153	0.32	2.6	0.27	1.7	0.544		

Table 5.5: Second floor pressure drop results.

5.6.4 Third floor:

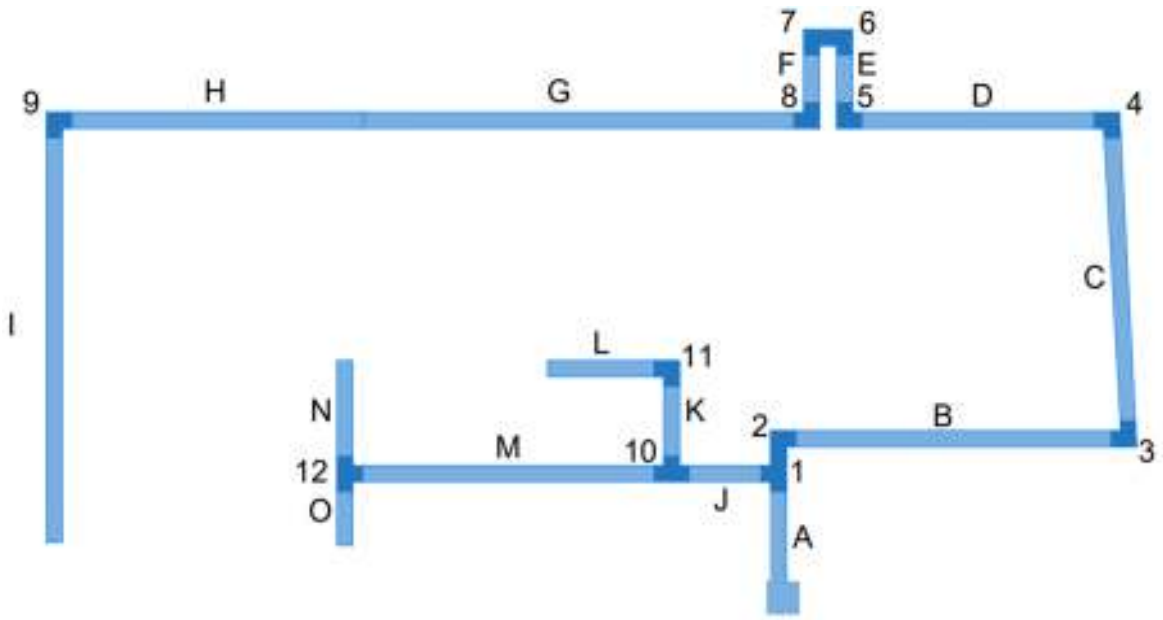


Figure 5.10: Third floor duct and fitting labels.

ID	TYPE	Volume Flow Rate(m^3/s)	ΔP (Pa/m)	Velocity (m/s)	Diameter (m)	Lenght (m)	Duct Loss (pa)	Direction	Fitting loss (pa)
A	Duct	4.711	0.32	6	1	2.85	0.912		
1	Tee							A-2 A-J	2.35 11.15
2	Bend							1-B	2.178
B	Duct	3.375	0.32	5.5	0.9	10.05	3.36		
3	Bend							B-C	2.099
C	Duct	3.038	0.32	5.4	0.85	8.91	2.851		
4	Bend							C-D	2.099
D	Duct	3.038	0.32	5.4	0.85	7.4	2.368		
5	Bend							D-E	1.8
E	Duct	2.429	0.32	5	0.75	1.5	0.48		
6	Bend							E-7	1.8
7	Bend							6-F	1.8
F	Duct	2.429	0.32	5	0.75	1.5	0.48		
8	Bend							F-G	1.8

G	Duct	2.429	0.32	5	0.75	13.7	4.384		
H	Duct	0.926	0.32	4	0.55	9.3	2.976		
9	Bend							H-I	0.691
I	Duct	0.306	0.32	3.1	0.35	12.7	4.064		
J	Duct	1.336	0.32	4.4	0.6	2.3	0.736		
10	Tee							J-M J-K	1.31 4.32
K	Duct	0.223	0.32	2.8	0.32	2.2	0.704		
11	Bend							K-L	0.564
L	Duct	0.223	0.32	2.8	0.32	3.4	1.088		
M	Duct	1.113	0.32	4.1	0.55	9.3	2.976		
12	Tee							M-N M-O	3.73 3.73
N	Duct	0.153	0.32	2.6	0.27	3	0.96		
O	Duct	0.153	0.32	2.6	0.27	1.7	0.544		

Table 5.6: Third floor pressure drop results.

5.6.5 Fourth floor:

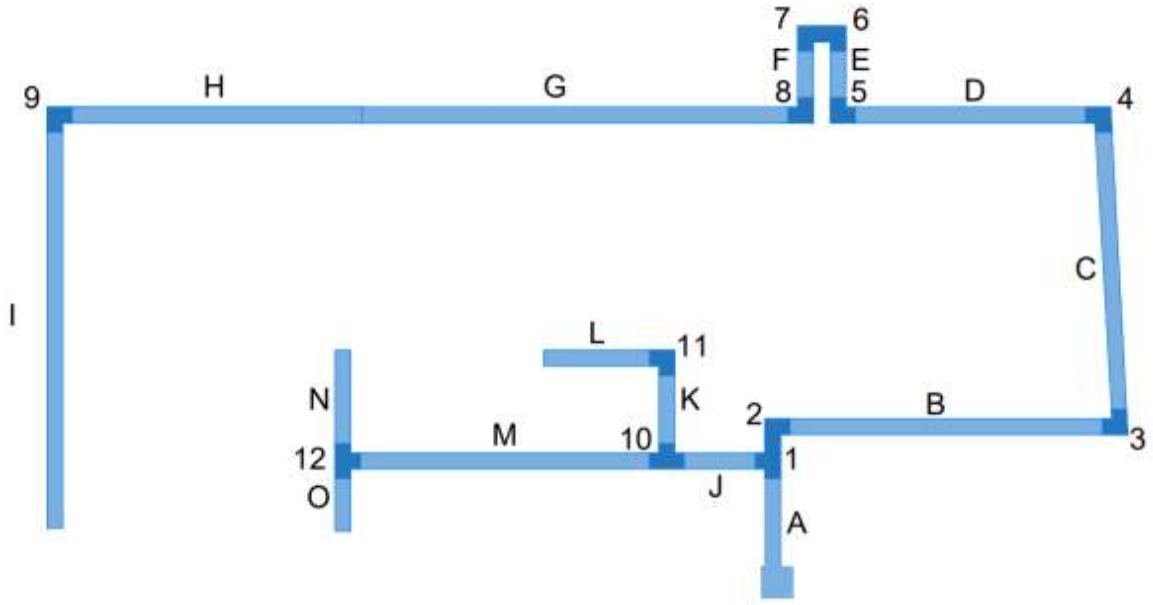


Figure 5.11: Fourth floor duct and fitting labels.

ID	TYPE	Volume Flow Rate(m^3/s)	ΔP (Pa/m)	Velocity (m/s)	Diameter (m)	Lenght (m)	Duct Loss (pa)	Direction	Fitting loss (pa)
A	Duct	5.825	0.29	6	1.1	2.85	0.826		
1	Tee							A-2 A-J	2.43 11.55
2	Bend							1-B	2.256
B	Duct	4.171	0.29	5.6	0.95	10.05	2.914		
3	Bend							B-C	2.174
C	Duct	3.754	0.29	5.5	0.88	8.91	2.584		
4	Bend							C-D	2.174
D	Duct	3.754	0.29	5.5	0.88	7.4	2.146		
5	Bend							D-E	1.864
E	Duct	3.002	0.29	5.1	0.78	1.5	0.435		
6	Bend							E-7	1.864
7	Bend							6-F	1.864
F	Duct	3.002	0.29	5.1	0.78	1.5	0.435		
8	Bend							F-G	1.864
G	Duct	3.002	0.29	5.1	0.78	13.7	3.973		
H	Duct	1.144	0.29	2	0.6	9.3	2.697		
9	Bend							H-I	0.715
I	Duct	0.378	0.29	3.3	0.37	12.7	3.683		
J	Duct	1.651	0.29	4.5	0.63	2.3	0.667		
10	Tee							J-M J-K	1.357 4.475
K	Duct	0.275	0.29	2.9	0.34	2.2	0.638		
11	Bend							K-L	0.584
L	Duct	0.275	0.29	2.9	0.34	3.4	0.986		
M	Duct	1.375	0.29	4.2	0.55	9.3	2.697		
12	Tee							M-N M-O	3.864 3.864
N	Duct	0.189	0.29	2.8	0.28	3	0.87		
O	Duct	0.189	0.29	2.8	0.28	1.7	0.493		

Table 5.7: Fourth floor pressure drop results.

5.6.6 Vertical duct:

floor	Volume Flow Rate(m^3/s)	ΔP (Pa/m)	Velocity (m/s)	Diameter (m)	Lenght (m)	Duct Loss (pa)	Fitting loss (pa)
Fourth	5.825	0.29	6	1.1	3.2	0.826	
bend							2.623 24.38
Third	10.536	0.22	6	1.45	3.2	0.704	
Tee							8.59 42.136
Second	15.247	0.17	6	1.75	3.2	0.544	
Tee							18.02 61.5
Firstt	21.757	0.14	6	2.1	3.54	0.495	
Tee							34.39 94.35
Ground	27.023	0.12	6	2.3	4.08	0.489	
Tee							56.862 102.04
					Total	3.061	444.89

Table 5.8: Vertical duct pressure drop.

Floor	Pressure drop (pa)
Ground floor	51.943
First floor	69.745
Second floor	70.204
Third floor	70.204
Fourth floor	68.943
Vertical duct	447.95
Total pressure drop	774.989

Table 5.9: Total pressure drop.

5.7 Blower fan's Pressure drop:

The pressure drops before the blower that is to say at the suction of the fan which must include the pressure drops of the following elements:

a) Coil Pressure drop of the air handler unit ΔP_{coil}:

On usual bases, these losses are between 150-230pa, we take 200pa.

b) Air filter pressure drop ΔP_{AF}:

According to the tables below, we choose an average group and class filter M5 the pressure drops are 450 Pa with average particle efficiency (Em) between 40 and 60.

Classification of air filters [1]

Group	Class	Final pressure drop (test) Pa	Average arrestance (A_m) of synthetic dust %	Average efficiency (E_m) for 0.4 µm particles %	Minimum efficiency [2] for 0.4 µm particles %
Coarse	G1	250	$50 \leq A_m < 65$	–	–
	G2	250	$65 \leq A_m < 80$	–	–
	G3	250	$80 \leq A_m < 90$	–	–
	G4	250	$90 \leq A_m$	–	–
Medium	M5	450	–	$40 \leq E_m < 60$	–
	M6	450	–	$60 \leq E_m < 80$	–
Fine	F7	450	–	$80 \leq E_m < 90$	35
	F8	450	–	$90 \leq E_m < 95$	55
	F9	450	–	$95 \leq E_m$	70

Table 5.10: Air filters pressure drop.

Therefore: ΔP_{AF}= 450 Pa

The pressure drops at the suction of the blower fan will be:

$\Delta P_{AHU} = \Delta P_{coil} + \Delta P_{AF} = 450 + 200 = 650pa$

$\Delta P_{blow} = \Delta P_{AHU} + \Delta P_{Duct} = 650 + 774.989 = 1424.98$ pa

5.8 Calculation of the blower fan power

General form:

$$\text{Fan kW}=\left(\frac{Flow\times Pressure}{power\ conversion\times Efficiencies}\right)\times\text{HP Conversion} \quad \text{V.5}$$

Applied forms:

$$\text{Fan kW}=\left(\frac{Air\ Flow\times Inches\ of\ static}{6356\times\eta Fan\times\eta Belt\times\eta Motor\times\eta Drive}\right)\times 0.746$$

Where:

kW: Electrical power supplies to fan motor system.

Air Flow: Flaw rate in cubic feet per minute (CFM).

Inches of static: Duct static pressure in inches of water column (w.c).

η_{Fan}=Fan efficiency=80%.

η_{Belt}=Drive belt efficiency (if belt-driven)=90%.

η_{motor}=Motor efficiency =85%.

η_{VFD}=VFD efficiency =95%.

6356: A power conversion constant to HP.

0.746: A power conversion constant to kW.

-We have 1 m^3/s=2 118.88CFM and we have 27.17m^3/s.

So Air Flow = 57570.31CFM.

And: in (w.c) = 248.84 pa we have 1424.98pa.

So Inches of static = 5.73 (w.c).

$$\text{Fan kW}=\left(\frac{57570.31\times5.73}{6356\times0.8\times0.9\times0.85\times0.95}\right)\times0.746$$

Fan Kw=66.59

5.8.1 Fan selection:

The fan used in the air handling unit is a forward Curved Centrifugal Fan for AHU - Wheel Type H – YFFCDH



From Manufacturer ANFINAIR

Airflow Range: 1,800～80,000 m³/h (1059～47,059 CFM)

Static Pressure Range: 100～1,400 Pa (0.4～5.62 in.WG.)

Drive Configurations: Belt Drive

Applications: AHU fans, HVAC fans, air purifier fans and ventilation fans among others

Figure 5.12: Wheel Type H – YFFCDH

5.8.2 Pulley selection:

Fast Shaft Rpm	Min Pulley Diameter (mm) Motor Power(kW)													
	≤1	3	4	5.5	7.5	11	15	18.5	22	30	37	45	55	75
2880	56	60	67	67	80	80	85	90	100	112	125	132	160	170
1800	56	63	71	75	80	85	95	106	112	125	150	155	170	190
1400	56	63	75	80	85	85	100	112	125	140	160	165	190	212
1200	56	71	80	80	95	95	106	118	132	150	160	175	200	236
960	56	75	80	85	95	100	112	132	150	180	180	190	224	250
720	56	80	85	100	100	106	132	150	160	190	200	224	250	

Table 5.11: Pulley diameter and RPM.

Eventually, we conclude a fan power of 66.59KW, nonetheless it is necessary to maintain an RPM of 1400 rpm to reduce fan noise equivalent and suited to both the YLRA Air handling Unit as well as the Variable Air Volume system installed.

Thus, the pulley diameter cannot be less than 205mm.

Conclusion

Conclusion

The results obtained from this study allowed us to conclude that the architecture of the edifice in question is in calorific standards, balanced in heat gain as well as heat loss with the ability to adjust its thermal comfort via a centralized system; this is justified by regulatory verification of the RTD C3-2 and RTD C3-4.

The choice of a centralized packaged heating and air conditioning air conditioning system with an air-based variable air volume diffusing mechanism is justified thanks to its modernity as well as the multiple advantages it offers.

Not only have we focused on the calculations and setup only, the tools and machinery used is considered of high quality and the most adequate in terms of noise control, odor disposition and hygiene, nonetheless, we were able to find a multitude of solutions to a variability of problem that have faced us, from which we can mention the insufficient height of the 2^{nd}, 3^{rd} and 4^{th} floor which we were able to correct by creating a partial plaster fake ceiling to allow us to not only carry our duct work through the building but also serve as an esthetic decoration for our interior design.

This project has proven that heating, ventilating and air conditioning knows no boundaries to for innovation and creativity, not only is this project a financial and economical landmark for the province of Medea, but also an example for this city and its neighbors to open up for more modern and more cost-efficient solutions for thermal comfort.

Bibliographical References

Bibliographical References

Documents:

[D1] -Regulatory technical document (D .T.R C) 3.2 thermal regulation for residential buildings rule for calculating heat losses (French edition).

[D2] -Regulatory technical document (D .T.R C) 3.4 air conditioning regulation, rules and laws for calculating the heat gain of buildings (French edition).

Websites:

[1] - **Green Education Foundation** on thermal comfort on the link: http://www.greeneducationfoundation.org/green-building-program-sub/learn-about-green-building/1239-thermal-comfort.html, visited on 05/03/2021 at 8:00 P.M.

[2] - **Science Direct**, Definitions of heat transfer modes on the link: https://www.sciencedirect.com/topics/index, visited on 18/03/2021 at 10:00 P.M.

[3] - **Designing Buildings**, Definition of heat gain on the link: https://www.designingbuildings.co.uk/wiki/Heat_gain, visited on 13/04/2021 at 2:00 P.M.

[4] - **Ebair Control**, The different types of HVAC systems defined on the link: https://ebaircontrol.com/blog/introduction-to-different-types-of-hvac-system, visited on 02/05/2021 at 3:30 P.M.

[5] - **Carrier**, Definition of Variable air volume systems on the link: https://www.carrier.com/commercial/en/in/products/commercial-products/controls/vav/, visited on: 22/05/2021 at 8:30 PM.

[6] -**The Engineering Mindset**, Ductwork sizing and efficiency design on the link: https://theengineeringmindset.com/ductwork-sizing-calculation-and-design-for-efficiency/?fbclid=IwAR0eHxMAmuDon3tQO0RsDDOfKZk09bGm0hpTWUGjlIQlz7jDv9OtBndVLNE, visited on 29/05/2021 at 8:00 P.M.

Catalogues:

[C1]- Series 19195, YLRA Air cooled heat pump scroll compressor (YORK-Jonson Controls).

[C2]- GB0606-PEGASUS TROX Variable Air Volume catalogue.

ADDENDUM

ADDENDUM

ZONE	Altitude (m)	t_{be} (en °C)	ZONE	Altitude (m)	t_{be} (en °C)
A	< 300 300 à 500 500 à 1000 ≥ 1000	6 3 1 -1	C	500 à 1000 ≥ 1000	- 2 - 4
B	< 500 500 à 1000 ≥ 1000	2 1 -1	D	< 1000 ≥ 1000	5 4
B'	<500 ≥ 500	0 voir Zone B	D'	< 1000	5

Annex Table 1: Climatic zone [D1]

	K [W/m²°C]
Portes : Porte extérieure en métal.	5,8
Porte extérieure en bois.	3.7
Portes intérieures.	3.00
Fenêtres : Fenêtres verre simple	3.3
Double Fenêtres	2.6

Annex Table 2: Doors and windows' coefficient K values [D1]

Valeurs de a (en W/m². °C)		
Zone climatique	Logement Individuel	Logement en immeuble collectif, bureaux, locaux à usage d'hébergement
A	1,10	1,10
B	1,10	0,90
B'	1,10	0,90
C	1,10	0,85
D1, D2 et D3	2,40	2,40

Annex Table 3: Values of a [D1]

Valeurs de c (en W/m². °C)		
Zone climatique	Logement Individuel	Logement en immeuble collectif. Bureaux. Locaux à usage d'hébergement
A	1,4	1,2
B	1,2	1,2
B'	1,2	1,2
C	1,2	1,2
D	1,4	1.4
D1, D2 et D3	1,4	1,4

Annex Table 4: Values of c [D1]

Hauteur H (1) (m)	Classes de rugosité (2) e_v				
	V	IV	III	II	I
H 4	0,40	1,47	2,71	4,06	6,36
4 < H 7	1,10	2,30	3,51	4,82	7,08
7 < H 11	1,76	3,00	4,19	5,46	7,67
11 < H 18	2,57	3,87	4,97	6,17	8,32
18 < H 30	3,50	4,80	5,80	6,93	9,02
30 < H 50	4,47	5,78	6,66	7,71	9,72

Annex Table 5: Value of e_v [D1]

Type de parois	Valeurs de P_O (m /h.m² sous P = 1 Pa)
Fenêtre ou porte fenêtre	4,0
Porte avec seuil et joint d'étanchéité	1,2
Porte	6,0
Double fenêtre	2,4

Annex Table 6: Value of P_0 [D1]

Nombre de pièces principales	1	2	3	4	5	> 5
Qv_{min} (m^3/h)	25	50	75	100	110	On ajoute 10 m^3 h par pièce supplémentaire

Annex Table 7: Minimal extraction flow rate [D1]

Nombre de pièces principales par logement	Q_{vmax}			
	Cuisine	Salle de bains	Autre salle d'eau	Cabinet d'aisance
1	75	15	15	15
2	90	15	15	15
3	105	30	15	15
4	120	30	15	30
5 et plus	135	30	15	30

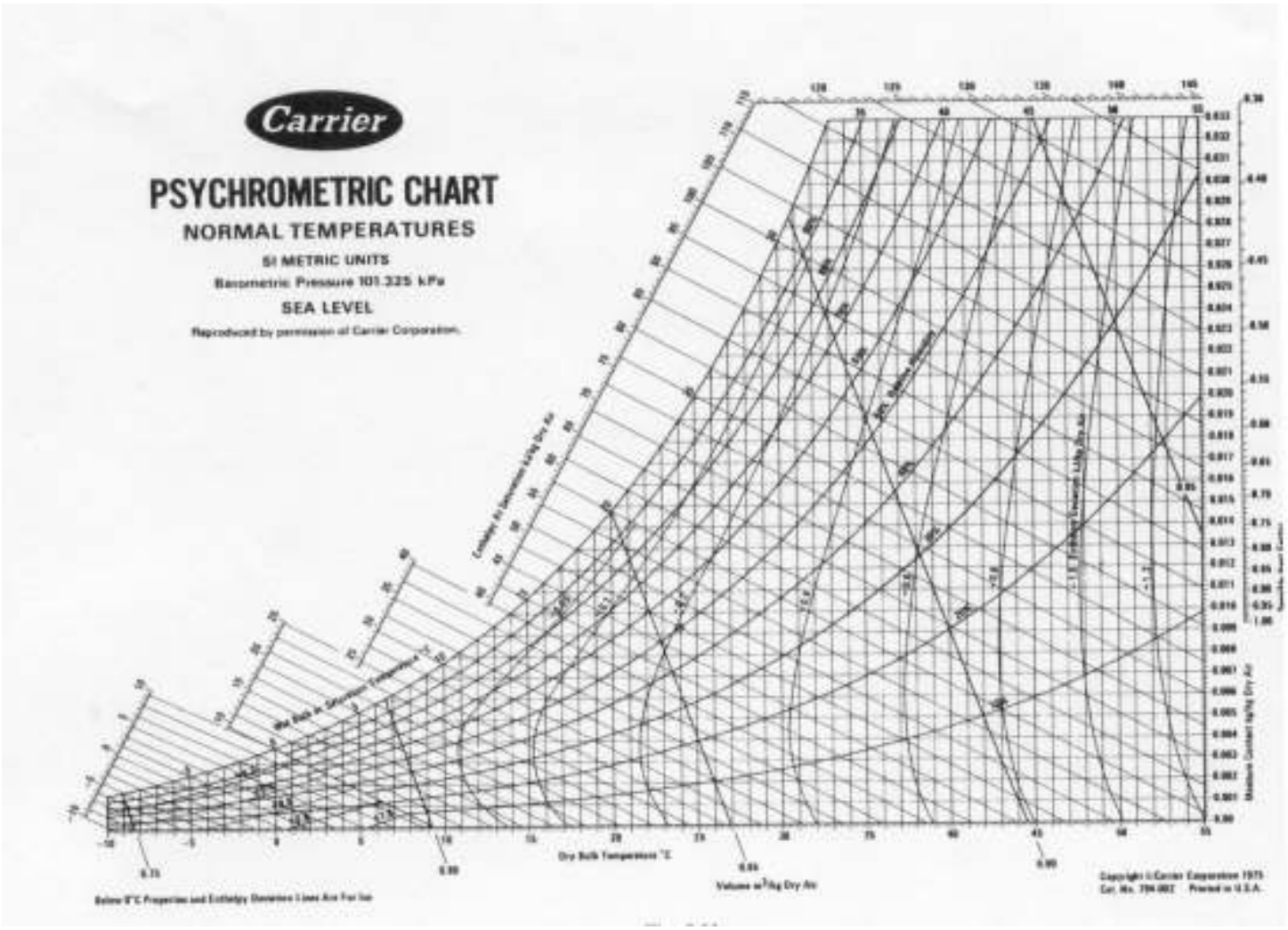
Annex Table 8: Maximal extraction flow rate [D1]

Zone	Logement individuel					Logement en immeuble collectif				
	a	B	c	D	e	a	b	c	d	e
A	1,10	2,40	1,40	3,50	4,50	1,10	2,40	1,20	3,50	4,50
B	1,10	2,40	1,20	3,50	4,50	0,90	2,40	1,20	3,50	4,50
B'	1,10	2,40	1,20	3,50	4,50	0,90	2,40	1,20	3,50	4,50
C	1,10	2,40	1,20	3,50	4,50	0,85	2,40	1,20	3,50	4,50
D	2,40	3,40	1,40	3,50	4,50	2,40	3,40	1,40	3,50	4,50
D'	2,40	3,40	1,40	3,50	4,50	2,40	3,40	1,40	3,50	4,50

Annex Table 9: values of a, b, c, d, e [D1]

1/h en m².°C/W	Paroi en contact avec : - l'extérieur, - un passage ouvert, - un local ouvert.			Paroi en contact avec : - un autre local, conditionné ou non conditionné, - un combe, - un vide sanitaire.		
	$1/h_i$	$1/h_e$	$1/h_i + 1/h_e$	$1/h_i$	$1/h_e$	$1/h_i + 1/h_e$
 Latéral (Mur) $\alpha > 60°$	0,10	0,04	0,14	0,10	0,11	0,21
 (toiture) $\alpha \leq 60°$	0,16	0,04	0,20	0,16	0,17	0,33
 (plancher)	0,08	0,04	0,12	0,08	0,09	0,17

Annex Table 10: Coefficients of superficial transfer in summer [D1]



Annex chart 1: The psychrometric diagram. [5]

Différence équivalente de Δtes(t) ou Δtem(t) (en °C)- murs ensoleillés ou à l'ombre																								
Orientation	m_{surf} Kg/m²	Temps solaire vrai																						
		6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	1	2	3	4	5
NE	≤100	2,8	8,3	12,2	12,8	13,	10,6	7,8	7,2	6,7	7,2	7,8	7,8	7,8	6,7	5,5	4,4	3,3	2,2	1,1	0	-1,1	-1,7	-2,2	-1,1
	300	-0,5	-1,1	-1,1	2,8	13,	12,2	11,	8,3	5,5	6,1	6,7	7,2	7,8	7,2	6,7	6,1	5,5	4,4	3,3	2,2	1,1	0,5	0	-0,5
	500	2,2	1,7	2,2	2,2	2,2	5,5	8,9	8,3	7,8	6,7	5,5	6,1	6,7	6,7	6,7	6,1	5,5	5,0	4,4	3,9	3,3	3,3	2,8	2,8
	≥700	2,8	2,8	3,3	3,3	3,3	3,3	3,3	5,5	7,8	8,9	7,8	6,7	5,5	5,5	5,5	5,5	5,5	5,5	5,5	5,0	5,0	4,4	3,9	3,9
SE	≤100	5,5	7,2	7,2	10,6	14	15,0	15,6	14,4	13,3	10,6	8,9	8,3	7,8	6,7	5,5	4,4	3,3	2,2	1,1	0	-0,5	-0,5	-1,1	-1,1
	300	0,5	0	0	7,2	11	13,3	15,6	14,4	13,9	11,7	10	8,3	7,8	7,2	6,7	6,1	5,5	4,4	3,3	2,8	2,2	1,7	1,7	1,1
	500	3,9	3,3	3,3	3,3	3,3	6,1	8,9	9,4	10	10,6	10	9,4	7,8	7,2	6,7	6,1	12,2	5,5	5,5	5,0	5,0	4,4	4,4	3,9
	≥700	5,0	4,4	4,4	4,4	4,4	3,9	3,3	6,1	7,8	8,3	8,9	10	8,9	8,3	7,8	7,2	11,1	6,7	6,7	6,1	6,1	5,5	5,5	5,0
SO	≤100	-1,1	-2,2	-2,2	-1,1	0	2,2	3,3	10,	14,4	18,9	22,	22,	23,	16,	13,	6,7	3,3	2,2	1,1	0,5	0,5	0	-0,5	-0,5
	300	1,1	0,5	0,5	0	0	0,5	1,1	4,4	6,7	13,3	17,	19,4	20	19,	18,	11,1	5,5	3,9	3,3	2,8	2,2	2,25	1,74	1,7
	500	3,9	2,8	2,8	2,8	2,2	2,8	3,3	3,9	4,4	6,7	7,8	10,6	12,2	12,	13,3	12,8	12,2	8,3	5,5	4,4	5,0	4,4	4,4	3,9
	≥700	4,4	4,4	4,4	4,4	4,4	3,9	3,3	3,6	3,3	3,9	4	5,0	5,5	8,3	10	10,6	11,1	7,2	4,4	4,4	4,4	4,4	4,4	4,4
NO	≤100	-1,7	-2,2	-2,2	-1,1	0	1,7	3,3	5,5	6,7	10,6	13,	18,3	22,2	20	18,	10	3,3	2,2	1,1	0	-0,5	-0,5	-1,1	-1,1
	300	-1,1	-1,7	-2,2	-1,7	-1,1	0	1,1	3,3	4,4	5,5	3	11,7	16,7	17,	17,	11,7	6,7	4,4	3,3	3,9	3,9	0,5	0	-0,5
	500	2,8	2,2	2,2	2,2	2,2	2,2	2,2	2,2	2,2	2,8	6,7	5	6,7	9,4	11,1	11,7	12,2	7,8	4,4	3,9	3,9	3,3	3,3	2,8
	≥700	4,4	3,9	3,3	3,3	3,3	3,3	3,3	4,4	3,3	3,3	3,3	3,9	4,4	5,0	5,5	7,8	10	10,	11,1	8,9	7,2	6,1	5,5	5,0
N (à l'ombre)	≤100	-1,7	-1,7	-2,2	-1,7	-1,1	0,5	2,2	4,4	4,4	6,7	7,8	7,2	6,7	5,5	4,4	3,3	2,2	1,1	0	0	-0,5	-0,5	-1,1	-1,1
	300	-1,7	-1,7	-2,2	-1,7	-1,1	-0,5	0	1,7	1,7	4,4	5,5	6,1	6,7	6,7	6,7	5,5	4,4	3,3	2,2	1,1	0,5	0	-0,5	-1,1
	500	0,5	0,5	0	0	0	0	0	0,5	0,5	1,7	2,2	2,8	2,8	2,8	4,4	3,9	4,4	3,3	2,2	1,7	1,7	1,1	1,1	0,5
	≥700	0,5	0,5	0	0	0	0	0	0	0	0,5	1,1	1,7	1,7	2,8	3,3	3,9	4,4	3,3	2,2	2,2	1,7	1,1	1,1	0,5

Annex table 11: Equivalent difference of Δtes(t) or Δtem(t) (°C)- shadowed or enlightened panels. [D2]

	Valeurs de CΔt en (°C)												
ΔTS =	Ecart diurne de base Eb												
She - TShi	7	8	9	10	11	12	13	14	15	16	17	18	19
-8	-14,3	-14,8	-15,3	-15,8	-16,2	-16,7	-17,1	-17,6	-18,0	-18,5	-19,0	-19,4	-19,9
4	-10,3	-10,8	-11,3	-11.8	-12,2	-12,7	-13,1	-13,6	-14,0	-14,5	-15,0	-15,4	-15,9
0	-6,1	-6,6	-7,1	-7,6	-8,0	-8,5	-8,9	-9,4	-9,8	-10,3	-10,8	-11,2	-11,7
2	-4,2	-4,7	-5,2	-5,6	-6,1	-6,6	-7,0	-7,5	-7,9	-8,4	-8,9	-9,3	-9,8
4	-2,2	-2,7	-3,2	-3,6	-4,1	-4,6	-5,0	-5,5	-5,9	-6,4	-6,9	-7,3	-7,8
10	3,6	3,1	2,6	2,2	1,7	1,2	0,8	0,3	-0,1	-0,6	-1,1	-1,5	-2,0
12	5,7	5,2	4,7	4,3	3,8	3,3	2,9	2,4	1,8	1,3	0,8	0,4	-0,1
14	7,7	7,2	6,7	6,3		5,3	4,9	4,4	3,8	3,3	2,8	2,4	1,9
16	9,7	9,2	8,7	8,3	7,8	7,3	6.9	6,4	5,8	5,3	4.8	4,4	3.9
18	11,7	11,2	10,7	10,3	9,8	9;3	8,9	8,4	7,8	7,3	6,8	6,4	5,9
20	13,7	13,2	12,7	12,3	11,8	11,3	10,9	10,4	9,8	9,3	8,8	8,4	7,9
22	15,8	15,3	14,8	14,4	13,9	13,4	13,0	12,5	11,9	11,4	10,9	10,5	10,0

Annex Table 12: Values of $C_{\Delta t.}$ [D2]

Nature du matériau	Facteur d'absorption	Nature du matériau	Facteur d'absorption
Asphalte feutre	0.90	pierre calcaire	0.60
bitumé	0.90	granit poli	0.45
Sable	0.70	acier poli	0.06
Ardoise	0,93	cuivre poli	0.04
Tuile	0.80	aluminium poli	0.04
carrelage clair	0.30	tôle galvanisée	0.70
marbre poli	0,60	amiante-ciment	0.96
tôle oxydée	0,80	peinture	0,60
tôle rugueuse	0,94	fonte brute	0.80
béton, brique	0,75	Gravier	0,80

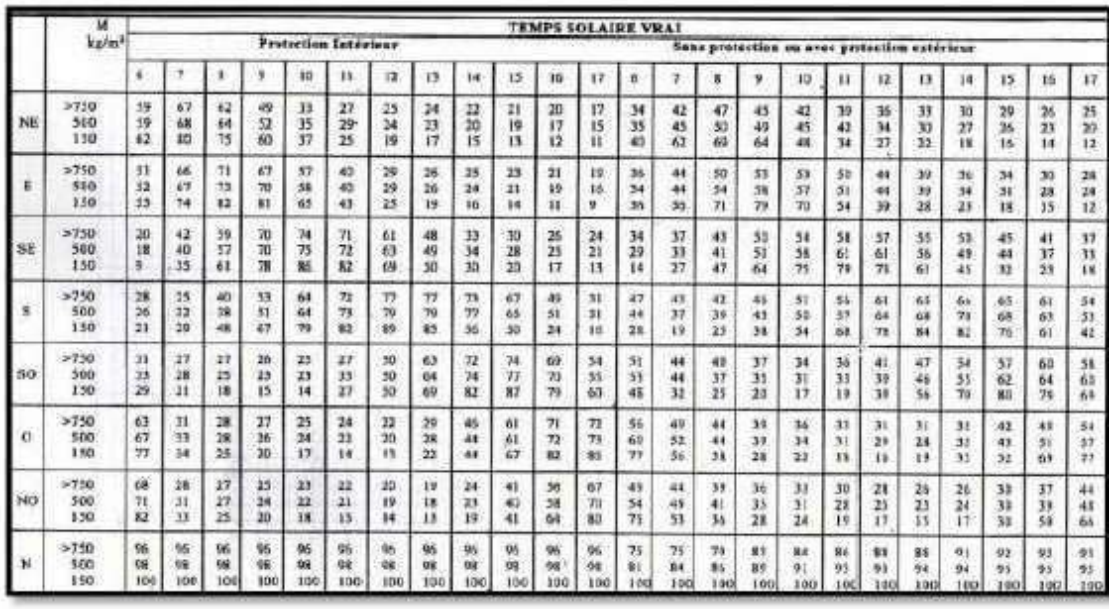
Annex Table 13: Absorption factors. [D2]

zone climatique	Température sèche $TS_{b,e}$ (°C)	Humidité spécifique $HS_{b,e}$ (g_{vap}/kg_{as})	Ecart diurne E_b (°C)	Température moyenne TS_m (°C)	Ecart annuel de température EAT (°C)
A alt≤500m 500≤ alt ≤ 1000m alt ≥ 1000m	34 33,5 30,5	14,5 13 13	9 10 9	25,5 25 22,5	31 32,5 31,5
B alt≤500m 500≤ alt ≤ 1000m alt ≥ 1000m	38 37 35	12,5 9 10	15 15 14	26,5 26,5 25	36 36 36
B' alt < 500m alt ≥ 500m	41 Voir zone B	11 Voir zone B	18 Voir zone B	29 29	41
C alt < 1000m alt ≥ 1000m	39,5 36	8,5 8,5	20 18	27 25	41,5 40
D1	44	6,5	15,5	33	38
D2	48	5,5	16,5	36,5	43
D3	39	5,0	12,0	29,6	35

Annex Table 14: Basic external conditions. [D2]

Rayonnement total de base It, b et diffus de base Id, b (en W/m²)									
Latitude Nord	Mois	ORIENTATION							
		N	NE	E	SE	S	SO	O	NO	Hori
20°	Juillet	59	435	514	267	44	267	514	435	791
	Août	34	372	520	356	81	356	520	372	788
	Septembre	31	273	514	441	205	441	514	273	733
30°	Juillet	50	413	516	315	94	315	516	413	776
	Août	34	340	520	406	198	406	520	340	741
	Septembre	28	284	498	479	330	479	498	284	668
40°	Juillet	47	400	516	394	217	394	516	400	734
	Août	34	321	511	459	321	459	511	321	675
	Septembre	28	183	470	511	441	511	470	183	577

Annex Table 15: Basic solar radiation. [D2]



	M kg/m²	TEMPS SOLAIRE VRAI																							
		Protection Intérieur												Sans protection ou avec protection extérieur											
		6	7	8	9	10	11	12	13	14	15	16	17	6	7	8	9	10	11	12	13	14	15	16	17
NE	>750	59	67	62	49	33	27	25	24	22	21	20	17	34	42	47	45	42	39	35	33	30	29	26	25
	500	59	68	64	52	35	29	24	23	20	19	17	15	35	45	50	49	45	42	34	30	27	26	23	20
	150	62	80	75	60	37	25	19	17	15	13	12	11	40	62	69	64	48	34	27	22	18	16	14	12
E	>750	31	66	71	67	57	40	29	26	25	23	21	19	36	44	50	53	53	50	44	39	36	34	30	28
	500	32	67	73	70	58	40	29	26	24	21	19	16	34	44	54	58	57	51	44	39	34	31	28	24
	150	33	74	82	81	65	43	25	19	16	14	11	9	36	50	71	79	70	54	39	28	23	18	15	12
SE	>750	20	42	59	70	74	71	61	48	33	30	26	24	34	37	43	50	54	58	57	55	53	45	41	37
	500	18	40	57	70	75	72	63	49	34	28	25	21	29	33	41	51	58	61	61	56	49	44	37	33
	150	9	35	61	78	86	82	69	50	30	23	17	13	14	27	47	64	75	79	75	61	45	32	25	18
S	>750	28	25	40	53	64	72	77	77	73	67	49	31	47	43	42	46	51	55	61	65	66	65	61	54
	500	26	22	38	51	64	73	79	79	77	65	51	31	44	37	39	43	50	57	64	68	73	68	63	53
	150	21	20	48	67	79	82	89	85	56	50	24	10	28	19	23	38	54	68	78	84	82	70	61	42
SO	>750	31	27	27	26	23	27	50	63	72	74	69	54	51	44	40	37	34	36	41	47	54	57	60	58
	500	33	28	25	23	23	33	50	64	74	77	70	55	53	44	37	35	31	33	39	46	55	62	64	63
	150	29	21	18	15	14	27	50	69	82	87	79	60	48	32	25	23	17	19	30	56	79	80	75	64
O	>750	63	31	28	27	25	24	22	20	46	61	71	73	56	49	44	39	36	33	31	31	31	42	48	54
	500	67	33	28	26	24	23	20	38	44	61	72	73	60	52	44	39	34	31	29	28	32	43	51	57
	150	77	34	25	20	17	14	15	22	44	67	82	80	77	56	38	28	22	15	16	19	31	52	69	77
NO	>750	68	28	27	25	23	22	20	19	24	41	56	67	49	44	39	36	33	30	28	26	26	33	37	44
	500	71	31	27	24	22	21	19	18	23	40	58	71	54	49	41	35	31	28	25	23	24	33	39	48
	150	82	33	25	20	18	15	14	13	19	41	64	80	75	53	36	28	24	19	17	15	17	33	58	66
N	>750	96	96	96	96	96	96	96	96	96	96	96	96	75	75	79	81	84	86	88	88	91	92	93	91
	500	98	98	98	98	98	98	98	98	98	98	98	98	81	84	86	89	91	93	93	94	94	95	95	95
	150	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100

Annex Table 16: Relative damping coefficient for 12 hours of work. [D2]

Dépendances	
Maison individuelle, toutes dépendances (caves, garage, cellier, ...)	0,5
Immeuble collectif d'habitation	
- Circulation commune	
sans ouverture directe sur l'extérieur	0,5
avec ouverture directe sur l'extérieur	2
avec bouche ou gaine de désenfumage ouverte en permanence	4
	4
- Hall d'entrée	1
- Garage collectif	0,5
-Autres dépendances	

Annex Table 17: values of N (h_{-1}). [D2]

Type de vitrage	Epaisseur (mm)	FS
Verre à vitre : normal	2 à 3	1
fort	4	0,93
épais	5	0,96
Glace claire ordinaire	6	0,95
	8,6	0,92
	10,5	0,90
	12	0,88
Verre absorbant	5,5	0,71
Glace absorbante		
couleur bronze	6	0,64
	8	0,60
	10	0,54
couleur grise	6	0,65
	8	0,60
	10	0,55
couleur verte	6	0,63
	8	0,58
	10	0,53
Verre coloré		
rouge fumé	3 à 6	0,56
bleu foncé	3 à 6	0,60
gris foncé	3 à 6	0,52
gris-vert	3 à 6	0,46
opale vert	3 à 6	0,43
opale foncé	3 à 6	0,37
ambre	3 à 6	0,70
Glaces réfléchissantes		
claires	6	0,72
	8	0,71
	10	0,70
bronze	6	0,55
	8	0,49
	10	0,45
gris	6	0,55
	8	0,50
	10	0,46
vert	6	0,53
	8	0,49
	10	0,45
Film réflecteur		
sur glace claire		
film A18		0,25
film A33		0,42
sur glace absorbante		
bronze		
film A18		0,33
film A33		0,41
gris		
film A18		0,12
film A33		0,21

Annex Table 18: Glass solar factor. [D2]

Appareils	Gains à admettre (en W)		
	Sens.	Sens.	Sens.
Machine à écrire	90	90	90
Micro-ordinateur (écran compris)	450	450	450
Mini-ordinateur (les valeurs peuvent varier de 2200 à 6600 W)	4400	4400	4400
Imprimante Plus de			
5000 lignes/minute	1800	1800	1800
300 à 600 lignes/minute	376	376	376
Terminal écran	160	160	160

Annex Table 19 : Heat gain due to office devices. [D2]

Appareils		Gains à admettre (en W)		
		Sens.	Sens.	Sens.
Présentoir (par m3 de volume intérieur)		640	640	640
Caisse enregistreuse		48	48	48
Réfrigérant à eau		1750	1750	1750
Sèche-cheveux central	5 casques	4483	4483	4483
	10 casques	6273	6273	6273
Casque sèche-cheveux		690	690	690
Stérilisateur	152x205x432mm	805	805	805
d'instruments. taille :	228x254x508	1518	1518	1518
	254x305x560 mm	1420	1420	1420
	254x305x914 mm	3048	3048	3048
	305x406x620 mm	2728	2728	2728
Stérilisateur ustensiles. Taille :				
	406 x 406 x 620 mm	3167	3167	3167
	508 x 508 x 620 mm	3676	3676	3676
Moulin à café		500	500	500
Mixer		1000	1000	1000
Fer à repasser		500	500	500
Aspirateur		200	200	200

Annex Table 20 : Heat gain due to various devices. [D2]

Valeurs de $\Delta TS_{réf, PH}$ (en °C)			
(alt = altitude) Zone climatique		Paroi en contact avec l'extérieur	Paroi en contact avec un local non conditionné
A	alt < 500 m	13,39	4
	500 ≤ alt < 1000 m	12,49	3,5
	alt ≥ 1000 m	9,99	0,5
B	alt < 500 m	14,54	5
	500 ≤ alt < 1000 m	13,59	4
	alt ≥ 1000 m	12,09	5
B'	alt < 500 m	16,09	6
	alt ≥ 500 m	voir zone B (en fonction de l'altitude)	
C	alt < 1000 m	13,49	6,5
	alt ≥ 1000 m	11,24	3
D1		13,09	9
D2		12,09	13
D3		12,09	6

Annex Table 21: Values of $\Delta TS_{réfph}$ for horizontal panels. [D2]

Valeurs de $\Delta TS_{réf,pv}$ (en °C) - Latitude 40°N									
Zone	Paroi en contact avec l'extérieur Orientation								Paroi en contact avec un I.n.c
(alt en m)	N	NE	E	SE	S	SO	O	NO	
A alt < 500	4.66	5.44	5.89	8.22	9.81	10.02	8.75	5.66	4
5005 alt <	3.76	4.54	4.99	7.32	8.91	9.12	7.85	4.76	3.5
alt k 1000	1.26	2.04	2.49	4.82	6.41	6.62	5.35	2.26	0.5
B alt < 500		6.59	7.04	9.37	10.96	11.17	9.9	6.81	5
5005. alt		5.64	6.09	8.42	10.01	10,22	8.95	5.86	4
1000 alt k 1000		4.14	4.59	6.92	8.51	8.72	7.45	4.36	5
B' alt < 500	7.36 8.14 8.59 10.92 12.51 12.7211.45 8.36								6
alt k 500	voir zone B (en fonction de l'altitude)								

Annex Table 22: Values of $\Delta TS_{réf,pv}$ for vertical panels. [D2]

Valeurs de NPVI								
Types de locaux	N	NE	E	SE	S	SO	O	NO
logement, locaux à usage d'hébergement	0,95	0,26	0,31	0,44	0,68	0,62	0,43	0,30
Bureaux	0,98	0,19	0,21	0,28	0,65	0,77	0,61	0,40

Annex Table 23: Values of $N_{PVI;\ réf}$. [D2]