

Design and Analysis of Office Ventilation System Integrated with Liquid Based Air Filtration System A-Case of Batu Caustic Soda and Cement Factory



Leliso Hobicho Dabelo

A Thesis Submitted to the Department of Mechanical Engineering

School of Mechanical, Chemical and Materials Engineering

Presented in Partial Fulfillment of the Requirement for the Degree of Master

of science in Thermal Engineering

Office of Graduate Studies

Adama Science and Technology University

June, 2022

Adama, Ethiopia

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Declaration

I hereby declare that this MSc. thesis entitled “*Design and analysis of office Ventilation system integrated with liquid based air filtration system A-Case of Batu caustic soda and cement factory*” in partial fulfillment of the requirement for the award of the degree of Master of Science in thermal engineering is my original work under the supervision of Dr. Addisu Bekele, Associate Professor of Mechanical Engineering Department.

The matter embedded in this thesis has not been submitted for the award of a degree or diploma in any other university. All relevant resource information used in this paper has been duly acknowledged.

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This is to certify that the above statement made by the candidate is correct to the best of my knowledge and belief. This thesis has been submitted for examination with my approval.

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We, the undersigned, members of the Board of Examiners of the final open defense by Leliso Hobicho Dabelo has read and evaluated his thesis entitled “*Design and analysis of office Ventilation system integrated with liquid based air filtration system A-Case of Batu caustic soda and cement factory*” and examined the candidate. This is, therefore, to certify that the thesis has been accepted in partial fulfillment of the requirement of the Degree of Master of Science in Thermal Engineering.

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DEDICATED TO
MY FATHER HOBICHO DABELO,
MY MOTHER SHUKARE ERKISO,
MY SISTERS KUFTU HOBICHO and DEMU HOBICHO

ACKNOWLEDGEMENT

First of all, I would like to give special thanks to God for giving me the patience, courage and strength in my life and during this work. I would like to express my deepest gratitude to my advisor, Dr. Addisu Bekele for his continuous support, encouragement, invaluable comments, and supervision that helped me during the time of the thesis work and he has been an enormous resource to me in all aspects of my academic life. I also would like to thank Dr. Gezahegn Habtamu for sharing his academic experience, and knowledge during this work. I express my sincere thanks to Mr. Sintayewu Shifarawu and Mr. Adis Tadele of the Thermal and Aerospace Engineering staff and Automotive Engineering staff, for their outstanding suggestions and continuous inspiration during my work. I would also like to thanks all my best friends for their encouragement, suggestion, and assistance, and gave a worthy comment on my work and through the process of researching. I am also grateful to express my sincere gratitude to Adama Science and Technology University for this opportunity. Finally, I extend my deep gratitude to thank my family for their support during tough times and motivating me throughout my life, and for their constructive suggestions and inspiration.

ABSTRACT

Recently the problem of air pollution is emerging due to increasing demand of fossil fuel, vehicle and factory. Especially one of the biggest problems is increase of a fine dust, and smokes from different factory. Since fine dust and other polluting particle was very fine particles of $10\mu\text{m}$ or less and it is difficult to remove. Adiabatic cooling plays an essential role in cleaning polluted air and ensuring human thermal comfort. Due to its dependence on a natural process of air cooled by water it has many advantages over other cooling processes, those includes creating non-pollution and of the fact that it uses fresh air and replaces the air time to time, quite frequently. This study presents design and analysis of central liquid based air filtration system integrated with ventilation system for Batu Caustic Soda and Cement factory office building. The basic principles including numerical dispersion modeling of air pollution, numerical analysis of ventilation system and experimental evaluation of liquid (water) based air filtration system saturation effectiveness (efficiency), achievable temperature drop, achievable temperature, evaporative rate, sensible heat transfer rate and air quality evaluation of the systems are evaluated. This paper also presents the results of false pollution creation and performance evaluation for the air filtration system. Comparative studies on the performance of liquid based air filtration system for atmospheric air pollution test with (4-8) mm thick cooling pad and false air pollution test with 6mm and 8mm thick cooling pad are conducted by using water as a cooling agent during experimental test. The result showed that the outlet temperature, maximum saturation effectiveness, maximum outlet relative humidity, maximum achievable temperature drop, maximum achievable temperature, maximum evaporative rate, maximum sensible heat transfer rate, CO_2 , HCHO and TVOC during atmospheric air pollution test are found to be 19.7°C , 80.02%, 69%, 3.9°C , 20.9°C , 4.92, 9.3 kW, 410ppm, 0.007 mg/m^3 and 0.002 mg/m^3 , with false air pollution test 20°C , 74%, 62%, 6.4°C , 21.5°C , 1.54, 0.25 kW, 862ppm, 0.11 mg/m^3 and 0.321 mg/m^3 respectively.

Keywords: Batu caustic soda and cement factory, liquid based air filter, saturation effectiveness, sensible heat transfer, cooling load and air quality index.

Table of Contents

ABSTRACT	vi
List of Figures	xii
List of Tables	xiv
ABBREVIATIONS	xvi
NOMENCLATURE	xvii
SYMBOL	xviii
CHAPTER ONE	1
INTRODUCTION	1
1.1 Background of study	1
1.2 Statement of the problem	3
1.4 Objectives of study	3
1.4.1 General objective	3
1.4.2 Specific objectives	3
1.4 Significance of the study	4
1.5 Scope of the Study and Its Limitation	4
1.6 Organization of the Study	4
CHAPTER TWO	5
LITERATURE REVIEW	5
2.1 Introduction	5
2.2 Air pollution	5
2.2.1 Particulate Matter (PM)	6
2.2.2 Ozone (O ₃)	7
2.2.3 Nitrogen dioxide (NO ₂)	8
2.2.4 Sulfur dioxide (SO ₂)	8

2.2.5 Carbon monoxide (CO)	9
2.2.6 Carbon dioxide (CO ₂).....	10
2.3 Air quality standards	12
2.3 The sources of criteria ambient air pollutants	13
2.3.1 Industrial emissions	14
2.4 Pollution Modeling.....	16
2.4.1 The Proximity model.....	17
2.4.2 Land use regression model (LUR).....	17
2.4.3 Interpolation model.....	18
2.5 Assessment of air pollution exposure in epidemiological studies	18
2.6 Batu city Air pollution.....	19
2.7 Mechanical Ventilation system	22
2.7.1 Human Comfort and Ventilation system	22
2.7.2 Ventilation Requirements	25
2.7.3 Air Infiltration.....	26
2.7.4 Building Office Air Distribution Systems	35
2.7.5 Duct systems	38
2.8 Evaporative cooling system	41
2.9 Literature summary	53
2.10 Research Gap.....	54
CHAPTER THREE	55
METHODOLOGY and MATERIALS	55
3.1Materials.....	55
3.2 Methodology	55
3.2.1 Data sources and collection techniques	57

3.2.2 Design parameters	57
3.2.3 Design Data	57
CHAPTER 4	59
THEORETICAL FORMULATION OF AIR QUALITY, VENTILATION SYSTEM AND CENTRAL LIQUID BASED AIR FILTRATION SYSTEM.....	59
4.1 Outdoor Air Quality	59
4.1.1 Atmospheric Dispersion Models	59
4.2 Theoretical Modeling of Outdoor Air Pollution	69
4.2.1 Calcium oxide (CaO_2) Plant Emission	70
4.2.2 Caustic Soda (NaOH) Plant Emission	79
4.2.3 Boiler Plant Emission	81
4.2.4 Ground Level Emission Concentration	83
4.3 Indoor Air Quality Modeling	87
4.4 Ventilation System	92
4.4.1 Heating and Cooling Loads	92
4.5 Duct Design.....	112
4.6 Design of liquid based air filtration System.....	128
CHAPTER FIVE	135
EXPERIMENTAL SETUP AND PROCEDURE OF LIQUID BASED AIR FILTRATION SYSTEM (DIRECT EVAPORATIVE COOLING).....	135
5.1 Introduction	135
5.1.1 Conceptual Model	135
5.3 Working Principle	136
5.4 Manufacturing Process	137
5.4.1 Construction of the cooling chamber	137
5.4.2 Water Tank	138

5.4.3 Cooling pad	139
5.4.4 Return water cleaner.....	139
5.4.3 Construction of Supporter Structure	140
5.4.4 Pipes and fittings	140
5.4.5 Water distribution header	141
5.4.6 Fan.....	141
5.4.7 Pump.....	142
5.5 Experimental setup	143
5.6 Experimental Procedure	144
5.7 Measuring Instruments	145
CHAPTER SIX.....	146
RESULTS AND DISCUSSIONS.....	146
6.1 Introduction	146
6.2 Ambient air Temperature and Air quality	146
6.3 Experimental Results of the air filtration system.	148
6.3.1 Outlet Temperature Variation.....	148
6.3.2 Relative Humidity Variation.....	149
6.3.3 Saturation effectiveness compression.....	150
6.3.4 Achievable temperature drop.....	151
6.3.5 Achievable temperature	152
6.3.6 Dew point temperature	152
6.3.7 Evaporation rate	153
6.3.8 Compression of Moisture absorption rate by air	153
6.3.9 Compression of sensible heat transfer	154
6.3.10 Compression of mass flow rate and air flow rate.	155

6.3.11 Experimental Results of the ambient air quality.....	156
6.4 Experimental Results of the false pollution	157
6.4.1 Experimental Results of the false pollution air quality	161
6.5 Economic Analysis.....	Error! Bookmark not defined.
CHAPTER SEVEN	162
CONCLUSIONS AND RECOMMENDATIONS	162
7.1 Conclusions	162
7.2 Recommendations	164
REFERENCES	165
APPENDIX.....	176

List of Figures

Figure2. 1 Comfort zones for summer and winter (Bradshaw, 2015).	24
Figure2. 2 Acceptable range of operative temperature and air speeds.	27
Figure2. 3: Diffuser model	28
Figure2.4:Schematic of the mechanically ventilated space and sampling points.	31
Figure2. 5: Stack effect and NPL on a building wall (Antretter & Pallin, 2019).	34
Figure2. 6: Typical infiltration and ventilation airflow (Hayati, 2017).	35
Figure 2. 7: Air washer (air filter) (Klein & Reindl, 2017).	45
Figure 2. 8: Psychometric chart	47
Figure 2. 9: Direct evaporative cooling system	49
Figure 2. 10: Psychometric graph.....	51
Figure 3. 1: Work flow diagram of evaporative cooling system integrated with ventilation system.	56
Figure 3. 2: Direct evaporative cooling.....	56
Figure 4. 1: A contaminant plume emitted from a point source (Laxmipriya & Narayanan, 2019).	61
Figure 4. 2: Calcium oxide (CaO ₂) plant annual sulfur dioxide Emission concentration,	71
Figure 4. 3: Calcium oxide (CaO ₂) plant annual Nitrogen dioxide Emission concentration,.....	73
Figure 4. 4: Calcium oxide (CaO ₂) plant annual Carbon monoxide Emission concentration,	74
Figure 4. 5: Calcium oxide (CaO ₂) plant Annual Ozone Emission concentration,	76
Figure 4. 6: Calcium oxide (CaO ₂) plant Annual Particulate Matter (PM ₁₀) Emission concentration.	77
Figure 4. 7: Calcium oxide (CaO ₂) plant Annual Particulate Matter (PM _{2.5}) Emission concentration.....	79
Figure 4. 8: Caustic Soda Plant Emission concentration.....	80
Figure 4. 9: Boiler Plant Emission concentration.....	82
Figure 4. 10: Ground level Annual Emission rate.....	84
Figure 4. 11: CaCO ₃ plant Emission compound Versis Ethiopia air quality standard.	85
Figure 4. 12: NaOH plant Emission compound Versis Ethiopia air quality standard.	86
Figure 4. 13: Boiler plant Emission compound Versis Ethiopia air quality standard.....	86
Figure 4. 14: Groung Level Emission compound Versis Ethiopia air quality standard.....	86
Figure 4. 15: General Box modeling out line	87
Figure 4. 16: Floor site of ground floor.....	97
Figure 4. 17: Psychometric analysis result for ventilation system of ground floor.	102
Figure 4. 18: Floor site of first floor.	103
Figure 4. 19: Psychometric analysis result for ventilation system of first floor.	109
Figure 5. 1: Conceptual schematic of a Liquid Based Air Filtration System	135

Figure 5. 2: Schematic layout of the Liquid Based Air Filtration System.	136
Figure 5. 3: Air filtration chamber.....	137
Figure 5. 4: Air filter Water tank.....	138
Figure 5. 5: Cooling pad.....	139
Figure 5. 6: Return water cleaner.....	139
Figure 5. 7: Fitting and pipes	140
Figure 5. 8: Water distribution header	141
Figure 5. 9: Fan	141
Figure 5. 10: Pump	142
Figure 5. 11: Experimental setup of the central liquid based air filtration system.	143
Figure 5. 12: False pollution creation.	144
Figure 5. 13: Measuring instruments (a) air quality detector, (b) pitot tube, (c) Digital thermometer.	145
Figure 6. 1: Ambient temperature and humidity versus time.	146
Figure 6. 2: Atmospheric air quality versus time.	147
Figure 6. 3: Ambient air inlet temperature and outlet temperature versus time.....	148
Figure 6. 4: Ambient air relative humidity and outlet temperature versus time.	149
Figure 6. 5: Saturation effectiveness versus time.....	150
Figure 6. 6: Achievable temperature drop versus time.	151
Figure 6. 7: Achievable temperature versus time.....	152
Figure 6. 8 Dew point temperature's versus time.....	152
Figure 6. 9: Evaporative rate versus time.	153
Figure 6. 10: Moisture absorption rate versus time.	154
Figure 6. 11: Sensible heat transfer versus time.....	154
Figure 6. 12: Air flow rate versus time.	155
Figure 6. 13: Air mass flow rate versus time.....	155
Figure 6. 14: CO ₂ versus time.....	156
Figure 6. 15: HCHO versus time.....	156
Figure 6. 16: TVOC versus time.	157
Figure 6. 17: Saturation effectiveness compression.	158
Figure 6. 18 Temperature compression.....	158
Figure 6. 19: Achievable temperature drop compression.....	159
Figure 6. 20: Temperature drop compression.	159
Figure 6. 21: Evaporative rate compression.	160
Figure 6. 22: Relative humidity compression.	160

List of Tables

Table 2. 1: Ethiopia Air Quality StandardsFigure 2. 1	13
Table 2. 2: Summary of studies measuring air pollutants in Ethiopia/East shoa/Batu	20
Table 2. 3: Estimation of parameters in Equation.	32
Table 2. 4: A summary of room air distribution types.	36
Table 3. 1: Emission data of pollutant (2010-2021)	57
Table 3. 2: Metrology data	58
Table 3. 3: Outdoor dry bulb and wet bulb temperature	58
Table 4. 1: Pasquill Stability Classes (Awasthi et al., 2006).	63
Table 4. 2: Wind speed annual average of Batu city (2010-2021)	64
Table 4. 3: Plume standard deviation coefficient (Offermann et al., 1985).	64
Table 4. 4: The following Emission rate data was for 10 year on each pollutant.	68
Table 4. 5: Outdoor air quality modeling parameter	69
Table 4. 6: Calcium oxide (CaO ₂) plant theoretical modeling analysis parameter.	70
Table 4. 7: Caustic Soda (NaOH) Plant theoretical modeling analysis parameter.	79
Table 4. 8: Boiler Plant theoretical modeling analysis parameter.	81
Table 4. 9: Batu Caustic Soda and Cement Factory Indoor pollutant emission rate	88
Table 4. 10: Emission rate of each indoor portion of the building.	89
Table 4. 11: Emission from each office of the building	90
Table 4. 12: Total indoor concentration of the pollutant.	91
Table 4. 13: Abridged outdoor air requirements listed in ANSI/ASHRAE Standard 62-1989....	95
Table 4. 14: Dimension of office, wall, roof and window.	96
Table 4. 15: Zone Sizing Data	98
Table 4. 16: Space Loads and Airflows	99
Table 4. 17: Ventilation size summer Result.	100
Table 4. 18: Peak load detail summary	101
Table 4. 19: Zone Sizing Data	104
Table 4. 20: Space Loads and Airflows	105
Table 4. 21 Ventilation system design summery	107
Table 4. 22: Peak load detail summary	108
Table 4. 23: Main supply duct sizing for ground floor of the building.	114

Table 4. 24: Main supply duct sizing for first floor of the building.	115
Table 4. 25: Branch supply duct sizing result for ground floor of the building.	116
Table 4. 26: Branch supply duct sizing result for first floor of the building.	117
Table 4. 27: Main Return duct sizing for ground floor of the building.	118
Table 4. 28: Main Return duct sizing for first floor of the building.	119
Table 4. 29: Branch return duct sizing for ground floor of the building.	120
Table 4. 30: Branch return duct sizing for first floor of the building.	121
Table 4. 31: Supply duct diffuser sizing result of ground floor of building.	124
Table 4. 32: Supply duct diffuser sizing result of first floor of building.	125
Table 4. 33: Temperature and heat loose in the duct where:	127
Table 4. 34: Size of evaporative cooling equipment (ASHRAE hand book 2007)	129
Table 4. 35: Size of evaporative cooling equipment (ASHRAE hand book 2007)	129
Table 4. 36: Air change standard data.....	133
Table 6. 1: Ethiopian and international indoor air quality standard	147
Table 6. 2: Cost of components of the solar air dryer.....	Error! Bookmark not defined.

ABBREVIATIONS

WHO	World health organization.	EER	Energy Efficiency Ratio.
PM	Particulate matter.	DBT	Dry bulb temperature.
TPd	Tone per day.	WBT	Wet bulb temperature.
NaOH	Sodium hydroxide.	DPT	Dew point temperature.
CaO ₂	Calcium oxide.	IAQ	Indoor air quality.
SSAC	Sub Saharan African country.	O ₃	Ozone.
GBD	Global Burden of Disease.	GLC	Ground level concentration
EPA	Environmental protection association.	CFM	Cubic fit per minute
VOCs	Volatile Organic Compounds.	ODB	Outdoor dry bulb temperature
NO _x	Nitrogen oxides.	OWB	Outdoor wet bulb temperature
NO ₂	Nitrogen dioxide.	LDB	Living dry bulb temperature
SO ₂	Sulfur dioxide.	LWB	Living wet bulb temperature
CaCO ₃	Calcium carbonate.	x, y, z	Direction
CO ₂	Carbon dioxide.	a, b, c, d, f	Constant
PC	Precast concrete.		
AQA	Air Quality Act.		
Pb	Lead.		
CO	Carbon monoxide.		
LUR	Land use regression.		
CFD	Computational fluid dynamics.		
AQI	Air quality index.		

NOMENCLATURE

$C_{(x, y, z)}$	Continuous point source of pollutants.	H_h	Heating load (kW)
Q	Emission rate (mg/s)	t_s	Supply air temperature (K)
U	Air speed (m/s)	q_{mh}	Amount of supply air (m ³ /s)
V	Building volume (m ³)	H_d	Heat loss in a duct (kW)
q	Mechanical ventilation flow rate (m ³ /s)	t_1	Initial temperature in duct (K)
$C_{OA} (t, d)$	Outdoor air concentration (mg/m ³)	t_2	Final temperature in duct (K)
$P (d)$	Penetration factor	k	Heat loss coefficient of duct walls
K	Deposition loss	t_r	Room temperature in duct (K)
$E (d)$	Filtration efficiency (%)	e_c	Saturation effectiveness (%)
P	Pressure (Pa)	V_s	Plume speed (m/s)
C_p	Pressure coefficient	d	Stack diameter (m)
m_a	Mass flow rate of air (kg/s)	DB	Dry bulb temperature (K)
P_s	Packing power (W)	WB	Wet bulb temperature (K)
V_p	Total volume of packing (m ³)	q_h	Volume of air for heating (m ³ /s)
t^1	Thermodynamic wet bulb temperature (K)	A_p	Area of packing (m ²)
X	Concentration of pollutant (mg/m ³)	ΔH	Plume height (m)
t	Time (s)	W_o	Relative humidity of outlet air (%)
H	Effectiveness stack height (m)	W_i	Relative humidity of inlet air (%)
RH	Relative humidity (%)	T_s	Stack Temperature (K)
x	Distance from stack (m)		
E	Emission rate (g/s)		

SYMBOL

σ_x	Standard deviation in X direction
σ_y	Standard deviation in y direction
σ_z	Standard deviation in Z direction
ρ	Density (Kg/m ³)
τ	Optimum time (s)
ξ	Fluid speed (m/s)
η	Evaporative cooler efficiency (%)
σ	Number of particle

CHAPTER ONE

INTRODUCTION

1.1 Background of study

“There's so much pollution in the air now that if it weren't for our lungs there'd be no place to put it all,” Robert Orben. Our lung which inhale and exhale oxygen and carbon di oxide 12 to 20 times within a minute between our body and our environment have responsibility to cleans waste gases, such as carbon dioxide and other from our blood. If it were not for our lung there is no way we could handle this all. Friedrich Nietzsche also wrote about the pollution saying that, “one must be a sea, to receive a polluted stream without becoming impure.”

As stated on new WHO global air quality guide line on 22 September 2020 provides clear evidence of the damage air pollution inflict on human healthy, at even lower concentration than previously understood, since 2005 global update there has been a marked increase of evidence that show how air pollution affect different aspects of healthy (Lehtomäki et al., 2020). Every year exposure to air pollution is estimated to cause 7 million premature deaths and result in the loose of million more than healthy years of life, they also state that air pollution is the cause of one-third of deaths caused due to stroke, lung cancer, and chronic respiratory diseases approximately, 92% of the world's population is exposed to air pollution levels that are beyond the permissible limits prescribed by WHO (Gao et al., 2018).

People mostly spend their time in indoor environment such as school, office, home, etc. Researches revealed that indoor air is more polluted than outdoor air about 2-5 times. It is known that indoor air quality has direct effect on human health (Shrestha et al., 2019). There are three main pollution sources in case of indoor, the first was Materials and equipment's, (such as computer, electronic devices, oven, paint, coating, furniture etc.), the second was People in the building and the last and the most hazardous source of indoor air pollution was the polluted air from outdoor. The outdoor atmospheric air where subjected to a lot of contaminating particulate matter (PM) depending on the environmental activity, in case of chemical producing factory the air quality will be low and this is highly risk for the worker in such area without clean air (Staszowska, 2020).

Adiabatic cooling is the use of adiabatic process to cool the air passing through a wetted cooling media or a spray of water. The ancient Egyptians hung wet mats on their doors and windows, and wind blowing through the mats cooled the air-the first attempt at air conditioning. This basic idea was refined through the centuries: mechanical fans to provide air movement in the 16th century, cooling towers with fans that blew water cooled air inside factories in the early 19th century, swamp coolers in the 20th century. Modern technology has dramatically increased the efficiency and effectiveness of adiabatic cooling (WESCOR, 1992), (Andriamamonjy et al., 2019).

There is a lot of way to take control the outdoor polluted air from affecting indoor air to create a comfortable working place for worker; the first is Source Control wherein, the source of pollutants is considered, and steps are taken to eliminate or decrease pollutants right from the source (Roy et al., 2018). Adiabatic saturation are good for cost cutting budgets because it is designed without a refrigerant, condenser, pressurized pipes to provide comfort cooling, not only the materials are cheap but lesser energy consumption. Another is it uses 100% fresh air so it is healthy because it brings in outside air and exhaust stale air, smoke, odors and germs. It helps maintain natural humidity levels, which benefits both people and furniture and cuts static electricity. Lastly it does not need an air tight structure for maximum efficiency, so building occupants can open doors and windows (VehiCool, 1999), (Zhao & Yang, 2003).

Batu Caustic Soda and cement factory was established in 1989 with a starting capital of 400 million birr. With the design output of 10,000TPd with the NaOH mass fractions of 100%. Currently it has more than 700 employers, it is found in east Africa Ethiopia and it is located at 163 km south from the capital city Finfine, at Batu in oromia region. The factory is constructed by two different foreign companies with different period of construction. First, the main part (i.e. the evaporation and filtration section and slacking section) made by Russian in 1989. After a few years of working the company need to construct the kiln(furnace) Because of easily availability of calcium carbonate and more economical(expensiveness) of calcium oxide. It is used to convert calcium carbonate into calcium oxide. And the other part is built by Germans. They have planned new project on preparation of sodium hydroxide and chlorine by electrolysis. The production unit consists of one process stream and caustic soda is produced by the lime process. The technical grade sodium hydroxide is a NaOH water solution with NaOH mass fraction of number less than 45.5%.

1.2 Statement of the problem

The purpose of this study is to identify the effect that air pollution have on the field and office worker of Batu caustic soda and cement factory in Batu town. That air pollution in Batu caustic soda and cement factory rise due to high accumulation of polluting particulate matter (PM_{2.5}, PM₁₀) such as dust, dirty and soot. The other was gaseous pollution from calcium oxide (CaO₂) plant, sodium hydroxide (NaOH) plant and boiler steam generation plant and the liquid sludge from those plant highly boosting factory air pollution level. The analysis includes, the first section of this research report introduces the causes of particulate matter (PM) and Gaseous air pollution in Batu caustic soda and cement factory. It explains both how to measure and how to model indoor and even outdoor air pollution. The second section is a compilation of the most viable mitigation strategies of indoor air pollution in Batu caustic soda and cement factory office building: designing central liquid (water) based air filtration system integrated with ventilation system. The effects, implementation strategies, and specific strengths and weaknesses associated with each approach and models are described, including a comparison of solid air cleaning (filtration) system and liquid based air washing system using different literature review analysis approach. The final section provides a prototype development of liquid (water) based air filtration system and its performance analysis. In general, the associated findings of this research study are located within this final section.

1.4 Objectives of study

1.4.1 General objective

The overall goal of this study was Design and analysis of ventilation system integrated with central liquid (water) based air filtration system for Batu caustic soda and cement factory offices.

1.4.2 Specific objectives

- Identify, Measure and collect the polluting matter (gas, particulate matter and chemical) data detail in Batu caustic soda and cement factory outdoor air.
- Determine the air quality and volume requirement of each office.
- Design a liquid based air filtration system.
- Design ventilation system for office building of Batu caustic soda and cement factory.
- Design and develop a prototype for the liquid based air filtration system.
- Experimentally test the performance of the liquid based air filter system prototype.

1.4 Significance of the study

Air is the inevitable transfer medium of environmental pollution to people. Like polluted water and other polluted substance it is not easy to control air pollution. Batu caustic soda was one of the largest four factory in the Batu city that cause the air pollution and the measurement to control the pollution doesn't taken. in this study the cause of Batu caustic soda and cement factory air pollution will be studied in deep, identify of the particulate matter details in Batu caustic soda and cement factories outdoor air, the pollution level will be measured by using the right measuring equipment and modeled by using Two different modeling approaches were followed, Lagrangian modeling and Eulerian modeling, air quality and the volume requirement of each offices will be determined and modeled, design and model of liquid based air filtration system, ventilation system design and model design for each office. After all design and develop for the liquid (water) based air filtration system and an experimental performance analysis will take place. After this all Batu caustic soda and cement factory will notice how much environmental pollution it's causing and will take measurement for its worker safety and for Batu city also.

1.5 Scope of the Study and Its Limitation

The scope of this study will be designing and selecting components for liquid based air filtration system, development of the prototype from easily and locally available materials for conducting an experimental investigation. During the thesis works, some limitation makes the study hard. Lack of accessibility of measuring equipment like O₃, NO₂, SO₂ and CO detectors, flow meter, and others.

1.6 Organization of the Study

This thesis is composed of seven chapters. After the previous chapter introduces the concept of air pollution, air filtration, and ventilation system, a literature survey on the same was presented in chapter two with its summary. The third chapter presents the material and methodology section of the work. The fourth chapter formulates the theoretical design and sizing of system components for individual performance investigation. Then, the fifth chapter describes the conceptual design, construction process, and the experimental setup of the prototype. Chapter six presents the major findings and results of the experimental analysis conducted with their implication. The last chapter includes the conclusion of the whole work and recommendations for further study.

CHAPTER TWO

LITERATURE REVIEW

2.1 Introduction

Air pollution is defined as the state in which the atmospheric condition contains certain concentration of substances that may produce harmful effects on man and the environment (Chattopadhyay et al., 2014). The following chapter presents the literature review which discusses the most common ambient air pollutants affecting public health building in Adama, Oromia, Ethiopia Country, African continent especially sub Saharan African country (SSAC) and globally. The exposure levels of the air pollutants measured locally and globally together with their sources are discussed in this chapter.

The following literature discuss the need for design and analysis of office building ventilation system integrated with central liquid (water) based air filtration system a case of Batu caustic soda and cement factory. However, little is written in detail about resource sharing as a way to bridge the shortage of information resources among the liquid based air filtration system. I am grateful to some authors (Ni, 2002), (Andriamamonjy et al., 2019), (Elkamhawy & Jang, 2020), (Bustamante et al., 2017), (Abdullah & Damanhuri, 2014), (Version, 2017), (Czarski et al., 2020), (Shukla et al., 2021), (Vanpariya et al., 2019), (Perumal et al., 2021), (Rocha et al., 2020), (Mamaghani et al., 2020), (R et al., 2021), who provide some basic critical design and modeling idea of the liquid (water) based air filtration system, although their discussions outline funding as a major factor in this research work, effective designing and modeling of the air filtration system. By studying other relevant literature on air quality modeling, air pollution modeling, ventilation system design and modeling and integration of air filtration and ventilation system, literature helped me understand more fully how all those systems play a big role in delivering purified air to office building of Batu caustic soda and cement factory.

2.2 Air pollution

Mortality was extremely high in the industrial cities of the 19th century, but little is known about the role played by pollution in generating this pattern, due largely to a lack of direct pollution measures. Industrial pollution explains 30-40% of the relationship between mortality and population density in 1851-60, and nearly 60% of this relationship in 1900. Growing industrial

coal use from 1851-1900 reduced life expectancy by at least 0.57 years. A back-of-the envelope estimate suggests that the value of this loss of life, expressed as a one-time cost, was equal to at least 0.33-1.00 of annual GDP in 1900. Overall, these results show that industrial pollution was a major cause of mortality in the 19th century, particularly in urban areas, and that industrial growth during this period came at a substantial cost to health (Walker Hanlon, 2015).

Particulate matter air pollution is widely considered as the leading environmental cause of premature mortality ambient air pollution, especially particulate matter, is among the biggest environmental health risks globally. Exposure to air pollution is linked with non-communicable diseases such as cardiovascular and respiratory diseases, and cancers. While different assessments recognize the importance of ambient air pollution as a health risk, there are substantial differences in the estimated deaths. For instance in the Global Burden of Disease (GBD) study it was estimated that 2.9 million deaths were related to fine particle ($PM_{2.5}$) exposure in 2017 (Lehtomäki et al., 2020).

2.2.1 Particulate Matter (PM)

Particulate matter is the most common air pollutant comprising of a mixture of particles from natural and anthropogenic sources (WHO, 2021). These particulates contain both liquid and solid constituents. The size of the particle is important in determining the impact of particulate matter on health. Thus, particulate matter is categorized into two types: coarse particulates (2.5-10) and fine particles (<2.5). Coarse particles are PM_{10} particles with an aerodynamic diameter of 10 μm or less and fine particles are $PM_{2.5}$ particles with an aerodynamic diameter of 2.5 μm or less. Fine particles ($PM_{2.5}$) are considered to have a greater risk of causing human health effects compared to coarse particles (PM_{10}) (Wang et al., 2015). Sources of particulate matter emission include power plants industries and diesel trucks. Particulates are emitted by either natural or anthropogenic sources and could be emitted directly into the atmosphere as primary particles or formed by secondary processes.

Furthermore, PM_{10} is emitted from biological particles, particles generated mechanically from factory, mining construction, road traffic and other related sources. $PM_{2.5}$ include particles from motor vehicle combustion, burning coal, wood, fuel and dust from the road and soil (I & G, 2013). However, combustion sources of $PM_{2.5}$ are considered more harmful and risky to health compared to other sources (Kim et al., 2021).

The effect of PM on human health includes breathing problems through penetration of the alveolus and oxidative stress leading to cardiovascular and neurological problems (Amaral, 2015). Mortality rates and health risks are considered to be greater for PM_{2.5} than for PM₁₀ because PM_{2.5} has the ability to penetrate the alveoli which are the gas exchange region deeper in the lungs, subsequently causing lung diseases (Lehtomäki et al., 2020).

2.2.2 Ozone (O₃)

It has been decades since exposure to a high concentration of ground-level ozone has been identified as a harmful pollutant on both the environment and human health (Han et al., 2011). Exposure to ozone has increased in recent years due to the increase in motor traffic, factory emission and agricultural activities (Malakooti, 2010). Ozone is a reactive oxidant and respiratory irritant colorless gas that significantly contributes to atmospheric smog associated with adverse health events (Arciszewska & McClatchey, 2001).

Ground level ozone, unlike most other primary pollutants, is not emitted directly into the ambient air. It is a secondary pollutant formed in the presence of chemical reactions of NO_x family in the ambient air and Volatile Organic Compounds. Ground level ozone is associated with impaired lung function and breathing health effects on humans (Hay et al., 2015).

Environmental exposure to high levels of O₃ is a concern due to effects on human health and ecological system (Lai, 2010). Local meteorological conditions have a direct bearing and significant impact on ground-level ozone formation (Lee et al., 2014). A chemical reaction between Volatile Organic Compounds (VOCs) and nitrogen oxides (NO_x) in the presence of sunlight (photochemical) yield ozone formation (Munir et al., 2020).

Meteorological conditions influence the formation of physical and chemical processes involved in the formation of secondary pollutants such as ozone. Thus, O₃ is regarded as a secondary pollutant because of its formation reliance on sunlight as a precursor. It is argued that high levels of O₃ are normally observed in densely populated areas with high exposure to sunshine (Tian et al., 2021).

2.2.3 Nitrogen dioxide (NO₂)

Nitrogen oxide are well known composite gases of NO_x family abundant in the atmosphere. Nitrogen oxides form part of the highly reactive gases of nitrogen oxides (NO_x) family and also contributes to the atmospheric reaction in the formation of the ozone pollution. The well-known pollutants from this group are nitric oxides and nitrogen dioxide. Nitrogen gas is described as reddish brown and very reactive in the atmosphere with a pungent suffocating smell. (Shrestha et al., 2019) assert that nitrogen gas exposure is pervasive in developing countries in comparison to countries that are developed. While nitric oxide is described as a color less and odor less gas (Gurung et al., 2017).

Sources of NO₂ are thought to be of primary or secondary pollutants. Primary pollutant comes directly from emission sources, while secondary pollutant is formed during chemical reactions in the atmosphere (J. Zhong et al., 2016). Primary NO_x gases are the product of combusting fuel at high temperatures which lead to emissions from, for example, metallurgical furnace, blast furnace emissions and also emissions from vehicles (Tian et al., 2021). (Koukouli et al., 2020) further associate NO₂ with point source combustion site, vehicle traffic, domestic fuel combustion and industrial activities.

NO_x are strong oxidizing agents and capable of reacting with water in the atmosphere to form nitric acid that significantly contributes towards acid rain. High NO₂ personal exposure of 21µg/m³ was reported to result in asthma exacerbation (Roy et al., 2018). The WHO AQG and the SA-NAAQS values for short-term NO₂ exposures for an average 1-hour are 200µg/m³ and 40µg/m³ for the annual average (Taylan et al., 2021).

2.2.4 Sulfur dioxide (SO₂)

Sulfur dioxide is a color less gas with a sharp odor existing in both aerosol and gaseous form on the surface of the earth (WHO, 2021). SO₂ emissions in the developing countries have been relatively increasing compared to developed countries which have recorded a decrease in the last decade (Q. Zhong et al., 2020).

The 93% of the SO₂ emissions are from the northern hemisphere, while the southern hemisphere is only responsible for 7% emissions. Sources of Sulfur just like many other gases include both natural and anthropogenic sources. However, Sulfur is predominantly present in the atmosphere in a gaseous state as SO₂ (Tian et al., 2021). Additionally, SO₂ discharged into the atmosphere can transform through oxidation reaction processes and be converted into fine particulate sulfate (De, 2012). Fine particulate sulfate contributes towards harmful fine components of both PM₁₀ and PM_{2.5} (Elkamhawy & Jang, 2020).

Oxidation of SO₂ in the atmosphere form sulfate aerosols that contribute to acid rain deposition and high concentration of sulfate ions (SO₄₋₂) found in rainwater is the result of intense SO₂ emissions (Taylan et al., 2021). In South Africa, crushing calcium carbonate for production of calcium carbonate (CaCO₃) containing sulfur content is comprehensively used for calcium oxide production, poultry feed and sugar factory. The Major Sources of SO₂ in different country include factories and power station. SO₂ affect human health in the respiratory systems, further cause's inflammation of the respiratory tract that results in mucus secretion, coughing, exacerbating asthma condition and chronic bronchitis (WHO, 2021).

The WHO Air Quality Guidelines recommends that SO₂ concentration of 500µg/m³ must not be exceeded over a 10-minutes mean duration and the long-term average exposure in 24-hours should not exceed 20µg/m³ (IFC, 2007). The South African National Ambient Air Quality Standards (SA-NAAQS) for SO₂ mean concentration at 10-minutes and 24-hours is 500µg/m³ and 125µg/m³ (RSA, 2004).

2.2.5 Carbon monoxide (CO)

Carbon monoxide is a colorless and odorless poisonous gas produced by incomplete combustion of fossil fuels such as natural gas, petrol and oil. In Ethiopia, major sources of carbon monoxide include motor vehicles, cement and sugar factory and industrial processes. Other sources include cigarette smoke, wood stoves and fires from the forest. Carbon monoxide exposure is reportedly higher in the urban area and displays a habit to increase with traffic density.

The high greatest exposure to carbon monoxide concentrations tends to be recorded in different factories during process and commuter's hours (Saadi et al., 2021). The incomplete combustion of carbon in vehicles is responsible for more than 90% of carbon monoxide emissions resulting in air pollution. In the cities, motor vehicles are considered the greatest exposure sources of carbon monoxide. Studies have indicated that where concentration levels of carbon monoxide reach an average of 3-4 parts per million (ppm) along the roadways, the concentration inside the motor vehicles (Shrestha et al., 2019).

2.2.6 Carbon dioxide (CO₂)

The attention to environmental impacts of cement production has grown fast in recent decades. The cement and chemical industry is a significant greenhouse gases emitter mainly due to the calcinations of raw materials and the combustions of fuels. To reduce emissions of greenhouse gases (e.g., carbon dioxide, nitrous oxide and methane), which are responsible for the greenhouse effect. In particular, the cement industry contributes about 5% of global anthropogenic CO₂ emissions excluding land-use change, as the production of the binder is a highly energy-intensive and emitting process (Durastanti & Moretti, 2020).

CO₂ emissions continue to increase worldwide. In particular, in the recent construction field, large-scale buildings such as logistics centers are rapidly increasing. Most of these large buildings are built using the precast concrete (PC) method. However, in general, PC methods are produced in factories and transported to the site. In order to reduce CO₂ emissions generated in this process, the on-site production of PC members is essential. In other words, in terms of eco-friendliness, if the PC member is produced in the field, CO₂ emission generated during the transportation process can be reduced (Na et al., 2021).

The balance of evidence suggests that anthropogenic emissions of greenhouse gases out of which carbon dioxide is the most significant are having a discernible impact on the global climate (IPCC, 2007, 2013, 2021). Although it is well-known that the uniformly mixed nature of greenhouse gas emissions makes the geographical distribution of the emissions irrelevant, a per capita distribution of emission rights may nevertheless influence the political economy of negotiating multilateral agreements (Aldy, 2006).

Emissions of carbon dioxide to the atmosphere come from three main sources: (i) oxidation of fossil fuels, (ii) deforestation and other land-use changes, and (iii) carbonate decomposition. Cement the largest source of emissions from the decomposition of carbonates is a binding material that has been used since ancient times. But it was following World War II that the production of cement accelerated rapidly worldwide, with current levels of global production equivalent to more than half a ton per person per year (Andrew, 2019).

Global cement production has increased more than 30 fold since 1950 and almost 4-fold since 1990, with much more rapid growth than global fossil energy production in the last 2 decades. Since 1990 this growth has largely been because of rapid development in China where cement production has grown by a factor of almost 12 such that 73 % of global growth in cement production since 1990 occurred in China (van Oss, 2017).

There are two aspects of cement production that result in emissions of CO₂. The first is the chemical reaction involved in the production of the main component of cement, clinker, as carbonates (largely limestone, CaCO₃) are decomposed into oxides (largely lime, CaO) and CO₂ by the addition of heat. Stoichiometry directly indicates how much CO₂ is released for a given amount of CaO produced. Recent estimates are that these so-called “process” emissions contribute about 5 % of total anthropogenic CO₂ emissions excluding land use change (Boden et al., 2017).

The second source of emissions is the combustion of fossil fuels to generate the significant energy required to heat the raw ingredients to well over 1000 °C, and these “energy” emissions, including those from purchased electricity, could add a further 60 % on top of the process emissions (IEA, 2016). Total emissions from the cement industry could therefore contribute as much as 8 % of global CO₂ emissions. These process (sometimes called “industry” or “industrial process”) and energy emissions are most often reported separately in global emissions inventories (Na et al., 2021).

2.3 Air quality standards

The air quality standards are significant for effective air quality management. These air quality standards are used to determine the maximum concentration levels of a substance in the atmosphere. The air quality standards provide a guideline which assists to determine and indicate a difference between polluted and non-polluted atmosphere. Furthermore, these standards indicate what is considered to be a safe exposure level for public health and for general welfare purposes. Higher concentrations measured against air quality standards are likely to affect public health and cause environmental pollution. These air quality standards and their resultant effects are based on scientific findings (Koukouli et al., 2020).

The WHO is responsible for providing countries with scientific findings that serve as reference and guidance for developing air quality standards cognitive of health effects. However, the challenge with WHO scientific guidelines is that they are generic in approach and they do not consider individual economic status per country (Table 2.1). The Ethiopian National Air Quality Standards are given in Schedule 2 of the Ethiopian National Environmental Management: Air Quality Act (AQA) (Act No. 39 of 2004). The Act covers the following substances and their concentrations together with their average periods, sulfur dioxide (SO₂), carbon monoxide (CO), ozone (O₃), particulate matter less than 2.5 µm in aerodynamic diameter (PM_{2.5}) and Nitrogen dioxide (NO₂) as shown in (Table 2.2):

Table 2. 1: Ethiopia Air Quality Standards

Compound	Guideline value ($\mu\text{g}/\text{m}^3$)	Averaging time
Sulphur dioxide (SO_2)	500	10 Minutes
	125	24 Hours
	50	1 Year
Nitrogen dioxide (NO_2)	200	1 Hour
	40	1 Year
Carbon monoxide (CO)	100,000	15 Minutes
	60,000	30 Minutes
	30,000	1 Hour
	10,000	8 Hour
Ozone (O_3)	120	8 Hour
PM_{10}	50	1 Year
	150	24 Hour
$\text{PM}_{2.5}$	15	1 Year
	65	24 Hour
Lead (Pb)	0.5	1 Year

2.3 The sources of criteria ambient air pollutants

The World Health Organization (2013) identified industries, households, cars and trucks as the main contributing factors of ambient air pollution. The Western Cape neighborhoods' sources of air pollution originating from a variety of both combustion and non-combustion sources (Walton, 2015). However, transportation, mining activities, agriculture, domestic fuel burn, fires and industrialization are considered significant sources of the air pollutants in the province (Singh et al., 2013).

The Global Burden of Disease estimates for 2017 show that more than 4.9 million people die prematurely every year due to the adverse health impacts of air pollution from all causes, while ambient particulate matter pollution exposure alone accounts for 4.6 million deaths. Airborne particulate matter, size-fractionated as PM_{10} and $\text{PM}_{2.5}$, with aerodynamic diameter less than 10 μm and 2.5 μm , respectively, is a widely used indicator of ambient air pollution. Particles in the

PM_{2.5} size range reach the deepest portions of the respiratory system, including the small airways and alveoli (Shrestha et al., 2019).

Sub-Saharan Africa (SSA), in general, and the Eastern Africa region in particular, where Ethiopia is located, lack urban air quality monitors for PM_{2.5} and other pollutants. Hence, there is a paucity of evidence on fine particulate air pollution and other. Including Addis Ababa the one of the largest metropolitan cities in SSA all city in Ethiopia, characterized by a rapidly growing population with a high rate of urbanization. The number of motorized vehicles per capita is still fairly low, at 3 vehicles per 1000 persons; however, the majority of the vehicles are quite old, with most operating in the absence of an emission standard, modern emission control technology, or an effective smog controlling enforcement system, the pollution from factories and other a lots of pollution source (Saadi et al., 2021).

According to DEADP (2013), 42% of industrial emission contributes 58% of the particulate matter followed by domestic emissions on 42%. Sulfur dioxide emissions industry emits 96% of the concern gas and households only contribute 4%. Lastly, the industrial sector in the Cape Town metropolitan area contributes 82% of NO_x emissions and 18% is produced by domestic emissions. Sources of combustion generally include pollution from a point source such as burning of tyre, industrial emissions, households fuel burning, transportation, wildfires and fuel burning appliances. Sources of non-combustion generally include pollution from non-point sources such as fugitive emissions from wind erosion, agriculture, evaporative losses, landfill operations and wastewater treatment (Scorgie, 2003).

2.3.1 Industrial emissions

Industrialization is associated with emissions of particulate matter and gases (Roy et al., 2018). In Ethiopia Province and specifically in the city of metropolitan area, fuel combustion within industrial, production factories, road transport, residential and other sectors, accounts for the major sources of air pollution. A study conducted in Bishoftu/Ethiopia of spatial and temporal disaggregation of anthropogenic CO emissions established that the largest per pixel emission of CO under listed industrial activities was 41 Mg per hour.

Industrial listed activities included cement production process, chemical production process, ceramic processes, hydrocarbon refining processes, iron and steel processes, road processes for asphalt production, and waste incineration processes and electricity generation at gas turbine

power plants (Nickless et al., 2015). Particulates and gases emissions such as SO₂, NO_x and CO are linked to industrial emissions (Thomas et al., 2011).

The caustic soda production sector and cement production sector in the Batu caustic soda and cement factory generate significant exposure pollutants of gases and PM. soda production sector, such as detergent, glass product and plastic production used in the production processes is associated with emissions of volatile organic compounds and aforementioned pollutants of concern. Furthermore, there is cement production sector in industries which produce a lot of particulate matter (PM). These industries emit fugitive dust during manufacturing activities. Manufacturing processes that include combustion release SO_x, NO_x, CO₂ and CO pollutants. The location of industrial sectors is a concern in many cities. Most industries are reported to be situated within residential areas, exposing residents to environmental health risks (Chipindu, 2019). Industrial air emissions consist of several environmental harmful pollutants discharged into the atmosphere through primary and secondary processes (Taylan et al., 2021).

Following traffic emissions, industrial emission is regarded as the second largest pollutant of the atmosphere (Gull et al., 2013). Industrial emission is one of the global challenges responsible for the majority of pollution. Walton (2006) cited that melting industry is responsible for the release of 99% total emissions of particles and gases. Industrial pollution consists of particles especially waste gases like carbon monoxide, sulfur oxides, and nitrogen oxides (Jerumeh et al., 2015).

These gases are classified as the waste products of industry which ultimately end up in the air (Jerumeh et al., 2015). Carbon dioxide is a predominant pollutant in the manufacturing industrial sector (Lu & Price, 2012). However, pollution generated by the industry is influenced by what product is manufactured in the process. Industries that manufacture glass and metals generate emissions such as CO, NO_x, SO₂ and PM. Furthermore, the combustion of various fuels used for production contributes to the level of VOCs produced and the release of heavy metals into the environment (NHDES, 2015).

2.3.2.1 Emission sources in Ethiopia country including the City of Batu

In the Batu caustic soda and cement factory specifically in the City of Batu area, flower industries, wine industries, road transport, residential and other sectors, accounts for the major sources of air pollution. Dust and wastes are the additional pollution sources within the City of Batu. In addition to the transport and industrial sources, Batu caustic soda and cement factory production accounts for further significant sources of emissions in the province.

2.3.2.2 Meteorological effects and temperature inversions on ambient air pollution

Meteorology plays a key role in the formation, transport, and dispersion of air pollutants. The state of the atmosphere in an area has a direct impact on the accumulation and dispersion of pollutants in the atmosphere (Cerbu et al., 2019). Meteorological conditions such as atmospheric wind speed, wind direction, relative humidity, rain cloud cover and temperature influence the concentration of air pollutants in ambient air.

Thus, the concentration of ambient air pollution discharged into the atmosphere from various sources is also influenced either by the stability or instability of the atmosphere to absorb or disperse pollutants (Jayamurugan et al., 2013). Rain cleanses the atmosphere. The anti-cyclones that ridge eastward over the southern Cape coast follows after the rainy days. Then the table mountain's shielding properties and the climatic factors in winter and Table Mountain's shielding properties circumstances influence the accumulation of pollution (Jury et al., 1990).

2.4 Pollution Modeling

There are many difficulties in measuring air pollution that compromises exposure assessment results (Han & Naehar, 2006). Sellier et al. (2014) suggest that measurement methods for estimating exposure to air pollution may have an influence on the outcome of results. Air pollution measurements usually include quantitative estimations such as direct-reading from devices, continuous monitoring analyzers, sampling and analytical methods including either pumped or passive sampling methods. In order for epidemiological studies to present valid, accurate and representative information, data from a large number of individuals should be obtained (COMEAP, 2018).

Investigating large sample size becomes convenient when sampling is conducted in a large area that allows for quantification of air pollution within the area. Failure to quantify large areas in air pollution assessment poses a potential for exposure misclassification considering that the

distribution of air pollutants is not constant throughout the environmental atmosphere. In most studies of air pollution and health outcomes, modelling of exposures have been the main method of assigning exposures, and such methods have different strengths and limitations.

2.4.1 The Proximity model

The Proximity model is considered to be the simplest approach to assessing air pollutants exposure within a geographical area (Abramo et al., 2020). Proximity modelling measures the proximity of the exposed subject in relation to the source of exposure. Proximity model studies make use of a restricted number of covariates that could be capable of confounding the relationship between air pollution and health. Proximity modelling is the most applicable approach in distinguishing air pollution exposure within a city because it measures how close the subject to a pollution source is. This method is good in discovering an existing association between health effects consequences and air pollution exposure on assumption that the closer the subject to the emission sources proxies for exposure in human populations. However, it has its limitations because it uses a restricted number of covariates that could possibly confound the relationship between air pollution and health. For instance studies of this nature has a potential to result to misclassification and biased risk estimation due to the fact that they do not take into consideration exposure from outside the place of residence, work or school (Lazzeretti & Capone, 2015).

2.4.2 Land use regression model (LUR)

Land use regression model predicts the concentrations of pollution in a specific place based on surrounding land use and traffic pollution (Guo et al., 2020). LUR has the ability to provide exposition for small scale variability in pollutant concentrations within the city (Boniardi et al., 2019). The limitation of LUR is associated with monitoring data from an extensive number of sites to present reliable information which is considered to be a challenge (Jin et al., 2019). However, having contrasted these models this research study adopted the LUR model because of its practicability and applicability in the research study.

- **Atmospheric Dispersion Models:** An air dispersion modeling exercise involving the AERMOD air dispersion model was conducted to predict the impact of the emissions on ambient air quality from the proposed manufacturing facility, The emission rates for particulate matter less than ten microns (PM10), sulphur dioxide (SO₂), nitrogen oxides

(NO_x) and carbon monoxide (CO) from the proposed cement manufacturing facility were calculated based on the design pollutant rate and the volumetric flow rate associated with the vent from the respective fabric filter (Kumar et al., 2021).

Building and terrain effects were included as part of the modeling analysis, and the meteorological data set was defined using wind speed and direction from a weather station that was located of the proposed plant property boundary, and the use of other surface data (temperature, pressure, rainfall and relative humidity, solar radiation, cloud height and cover) (Demirarslan et al., 2017). The general Gauss dispersion equation, for a continuous point source of pollutants as a cloud of smoke resulting from the chimney of evacuated pollutants in the atmosphere, is calculated with

$$C_{(x, y, z)} = \frac{Q}{u\sigma_z \sqrt{2\pi}} e^{-\frac{y^2}{2\sigma_y^2}} \left[e^{-\frac{(Hr-He)^2}{2\sigma_z^2}} + e^{-\frac{(Hr+He)^2}{2\sigma_z^2}} \right] \quad (2.1)$$

2.4.3 Interpolation model

The levels of the air pollutants are acquired from specified monitoring stations scattered all over the study area (Z. Liu et al., 2021). Interpolation model techniques have an advantage over the proximity models due to their use of real pollution measurements in the computation of exposure estimates (Yang & Xing, 2021). Multiple interpolation methods face challenges in obtaining available monitoring data, due to their requirement of a reasonably dense network of sampling sites.

2.5 Assessment of air pollution exposure in epidemiological studies

Air pollution is recognized as the largest global environmental risk to health contributing to an estimated 7 million premature deaths worldwide each year (WHO, 2018). Globally, an estimated 93% of children live in environments where outdoor air pollution levels exceeds World Health Organization health-based air quality limit values, and more than one in four deaths of children aged under 5 years is directly or indirectly related to environmental risks (WHO, 2018). The economic costs of these losses is also significant; the World Bank estimated that globally air pollution led to USD 5.1 trillion in welfare losses and USD 225 billion in lost labor income globally in 2013 (World Bank and IHME, 2016).

Modern epidemiological studies take place at the population level and use ambient air pollution levels as the exposure of interest (Jin et al., 2019). When conducting long term epidemiological studies involving personal exposure estimation for large populations, it is imperative to consider population density together with geographic information systems (Lehtomäki et al., 2020). In numerous studies, air pollution measurement is achieved by using stationary monitors within an area of interest. However, these stationary monitors in most cases are not accurate enough. Furthermore, stationary monitors are not optimized to detect spatial variability of pollutants within large areas, thus, this study conducts exposure assessment in households (Boniardi et al., 2019).

2.6 Batu city Air pollution

Previous studies conducted by WHO (2003) realized air pollution as an important contributor to the burden of disease in Ethiopia. Table 2.4 demonstrates exposure levels of air pollutants in Ethiopia and Batu city specially Batu caustic soda that have conducted an exposure assessment of air pollution within their respective regions. The table indicates the disparities in exposure levels and assessment methods for different air pollution gases and particulates. Exposure assessment of air pollution is important for determining health risks, health conscious, urban planning and developing strategies to reduce costs associated with air pollution and enforcing air quality standards (L B Merga et al., 2020). Regardless of air monitoring significance and associated benefits thereof, the majority of Ethiopian cities do not monitor exposure to air pollution.

Where there is exposure monitoring to air pollution, the monitoring system is usually regarded either flawed or working intermittently (Rodriguez et al., 2019). Moreover, monitored data is not readily available in scientific journals for evaluations which makes it difficult to determine the levels of exposure (Lemessa B Merga et al., 2020). Similarly, where air monitored data is available, it is either not easily accessible to the public or not communicated clearly, this challenge affects policy making and limits public knowledge (Kawo, 2018).

Table 2. 2: Summary of studies measuring air pollutants in Ethiopia/East shoa/Batu

Reference	Pollution	Cities	Aim of Study	Exposure Assessment Equipment	Exposure result	Air quality standard
(Kawo, 2018)	Environmental Policy in Ethiopia	Batu	To analyze the state of devolution in Batu.	Both purposive and random sampling.	-	-
(L B Merga et al., 2020)	Trends in chemical pollution and ecological status	Batu	Reviews the biological resources and spatiotemporal variation of water quality of the lake focusing on nutrients, metals and pesticides.		The reported values of some nutrients and trace metals of the lake (Ca ²⁺ , NO ₃ ⁻ , and alkalinity, NH ₄ ⁺ , Cu and Ni) and at shoreline spots (NO ₃ ⁻ and PO ₄ ³⁻) have shown an increasing temporal trend. S	Moreover, high sediment trace metal concentrations were observed at spots close to floriculture farming (Floriculture -1), at sites receiving urban effluents (Around lodges and Korekonch) and at the center of the lake.
(Kuto et al., 2018)	The influence of mechanized farming and industrialization on the Oromo people	Batu	Environment highly affected by mechanized farming and industrialization in the study area.	Interviews, observation and Descriptive and interpretative techniques to analyze the qualitative data.	Accordingly 94.6% of the respondents replied that their livelihood has been affected	Exceeds WHO guide line

Reference	Pollution	Cities	Aim of Study	Exposure Assessment Equipment	Exposure result	Air quality standard
(Haile & Seid, 2020)	The psycho-social context of Batu/ Dembel	Batu	To organize psycho-social information pertaining to the Air pollution issues	Interviews, questioner and interpretative techniques to analyze the qualitative data.	70% bio diversity change around Batu cities	Exceeds WHO guide line
(Huib et al., 2008)	Towards a sustainable future of the western shoreline Batu/ Ziway	Batu	To study the fragility of air pollution in great African rift valley.	Land using planning approach, Interviews and questioner		

2.7 Mechanical Ventilation system

Health has been an important concern to all of us these days. Because of that, having lots of biological pollutants and dust mite in the living place may cause a serious threat to our health. Thus, the improvement of the indoor air quality is an increasing concern to many people (Perumal et al., 2021). The Air Ventilation is the intentional introduction of ambient air into a space and is mainly used to control indoor air quality by diluting and displacing indoor pollutants (Saran et al., 2020).

Mechanical ventilation system provides sufficient amount of fresh air to the indoor environment. It has mentioned that there are standards for ventilation rate calculation. In a ventilation unit, there are supply and exhaust air fans, supply and exhaust air filters, control system, and heat recovery exchanger. The primary function is to supply fresh air to maintain indoor air quality and exhaust indoor air. Furthermore, energy recovery with ventilation is crucial in the point of view energy efficiency rules in building. Recently smart featured ventilation systems have been using to improve user comfort and energy efficiency of ventilation (Birtürk et al., 2020).

2.7.1 Human Comfort and Ventilation system

The purpose of a ventilation system is to provide an acceptable microclimate in the space being ventilated. In this context, microclimate refers to thermal environment as well as air quality. Increased levels of thermal comfort have become fait accompli through improved thermal insulation and more advanced air-conditioning and heating system design. On the other hand, a deterioration of indoor air quality has been experienced, particularly among air-conditioned buildings (Carli et al., 2020). For something so desirable, comfort can unfortunately be a nebulous concept. The room that makes one person put on a sweater can make another wish they were in shorts.

2.7.1.1 Thermal Comfort

Temperature is the most significant component to the experience of comfort in a space. Our bodies perform within an internal temperature range much narrower than external temperatures. In the process our bodies' metabolism generates heat, which must dissipate into the surrounding air or surfaces. When external temperatures are high, this process becomes more difficult and we may overheat or feel warm. When external temperatures are low, the rate of heat loss becomes more rapid, and we may feel uncomfortably cold (G. Liu et al., 2017).

- **Surfaces of a Room:** When assessing thermal comfort in a confined space, it must be examined both the general temperature in the room, as well as the uneven distribution of heat in the room. Typically when people refer to temperature, they mean the temperature of the air; however, our experience of thermal comfort depends on more than simply air temperature. The mean radiant temperature entails averaging the temperatures of each surface in the room. Combined with the air temperature this produces an overall measure, the mean operative temperature. However, even this measure has its limitations, as how close a person will be to a particular surface is usually variable, and in these cases different surfaces will dominate at different times (Guo et al., 2020).
- **Relative Humidity:** While temperature is the most important factors in generating a phenomenological sense of thermal comfort, many other climatic factors contribute. Relative humidity plays a large part in conjunction with temperature to provide a sense of discomfort. High levels of relative humidity can work against the evaporative cooling (ventilation) effects of sweating and leave the body prone to over-heating. Further, high levels of relative humidity in inclement winter weather produce a greater sense of cold (Li et al., 2020).

Human beings are sensitive to slight temperature changes, yet cannot perceive differences in relative humidity levels within the range of 25% and 60%, which is the primary reason that this range is often cited as the baseline (Palonen et al., 1993). If relative humidity falls outside this range, there are notable effects. When relative humidity gets too high, discomfort develops, either due to the feeling of the moisture itself, which is unable to evaporate from the skin, or due to increased friction between skin and clothing with skin moisture (Woloszyn et al., 2009). When relative humidity gets too low, skin and mucous surfaces become drier, leading to complaints about dry nose, throat, eyes, and skin. In particular, discomfort in working environments, which are prone to significant eyestrain, such as an office with a computer, is exacerbated (Kwag et al., 2019).

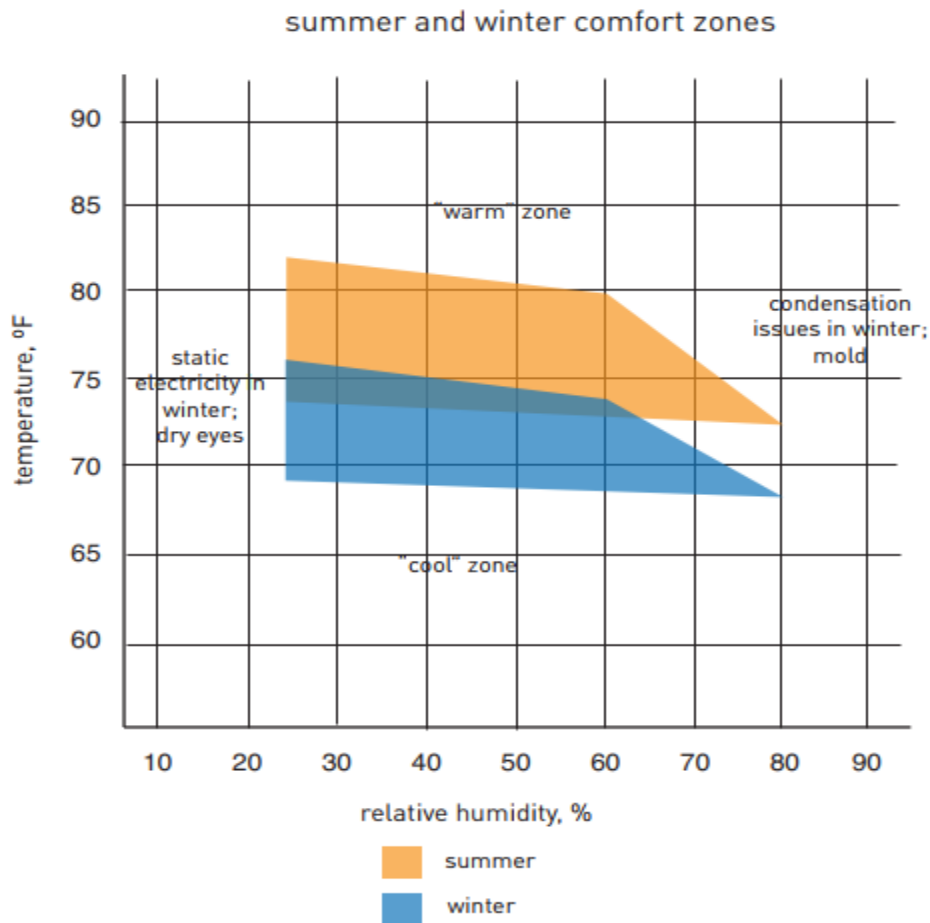


Figure2. 1 Comfort zones for summer and winter (Bradshaw, 2015).

- **Velocity of Air:** Air velocity plays a role in the perception of thermal comfort. In hot weather, as the body tries to cool itself, the flow of air across the body will assist evaporative cooling from sweating. When air has a high relative humidity, the air next to the sweating body may become saturated with moisture, but by moving the air next to the body away and bringing in fresh, lower-humidity air, the evaporation of sweat can continue (Ahmad Shafie et al., 2015). Mechanisms of convection can further move the heat generated by metabolic processes from the skin and into the surrounding air. All this leads to continued cooling, and the higher the velocity of air, the more effective is the process (Wu et al., 2017).

2.7.2 Ventilation Requirements

Nowadays, ventilation occupies an important position in the building design process since building occupants expect good standards of indoor air quality and comfort. People have become more aware of the effect of the indoor environment on health as a result of media publicity surrounding building related sickness. Coupled to this, it is argued that problems have been intensified as a result of a global reduction in air infiltration and general ventilation rates in order to conserve energy. In urban areas, most air pollution comes from human-made sources. Such sources can be classified as either mobile (cars, trucks, air planes, marine engines, *etc.*) or point source (factories, electric power plants, *etc.*) (Guo et al., 2020).

- **Carbon monoxide (CO):** CO is a gas that comes from incomplete combustion. The manufacturing industries and the residential/tertiary sector both come in second position, contributing 26% of emissions. Urban, collective or individual heating constitute the main part of the residential/tertiary sector contribution (Saadi et al., 2021).
- **Sulphur dioxide (SO₂):** In France, 85% of SO₂ emissions come from the use of sulphur containing manufacturing process emissions. These emissions are mainly released into the atmosphere by petroleum refining production of electricity (16%) and heating systems. The diesel motor vehicle sector is also responsible for a minor part of SO₂ emissions (Riordan & Adeeb, 2004).
- **Nitrogen oxides (NO_x):** In world (WHO, 2021), road traffic was responsible for releasing 48% (16% for diesel truck traffic) of NO_x emissions, followed by agriculture/forestry, the manufacturing industry and energy transformation (Munir et al., 2020).
- **Particulate matter (PM):** PM is a complex mixture of solid particles and liquid droplets suspended in the air that vary in size, composition and concentration. PM is usefully classified by size. PM with a mass median aerodynamic diameter <10 µm (PM₁₀; big particles) can stay in the air for minutes or hours and can travel as little as 100 yards or as much as 30 miles, while PM_{2.5} (fine particles) are lighter and stay in the air for days or weeks longer and travel farther. The sources of airborne PM comprise of the manufacturing industries, public buildings and the work sector, specifically agriculture/forestry, domestic heating systems and the incineration of waste. However, airborne particles are also released by motor vehicles, mainly diesel engines (Wang et al., 2019).

- **Volatile organic compounds (VOCs):** The number of VOCs is huge and a list is available on the web. Emissions from burning coal, oil and gasoline, and evaporation at gasoline service stations, and from solvents, cleaners and paints, all contribute to the baseline level of different VOCs found in outdoor air. Some VOCs are also released from tobacco smoke. The three main sources of VOC are road traffic, domestic or industrial use of painting, varnish or glue, and evaporation of hydrocarbons. VOCs levels in urban areas fluctuate widely; large differences occur because of both location and time (rush hour *versus* middle of the night) (Hay et al., 2015).

2.7.3 Air Infiltration

Since the mid-1970s there has been an urgent need to reduce the use of energy in the heating and cooling of buildings worldwide. In developing countries, the energy consumption for maintaining an acceptable environment in buildings constitutes by far the largest part of the total energy demand for that countries. In the developed countries, the proportion of energy usage in buildings is larger than that used in industrial processes or transport. It is not surprising, therefore, that numerous measures have been taken by governments, groups and individuals to reduce the usage of energy for heating and cooling buildings. This has recently been manifested by international agreements between countries (Shrestha et al., 2019).

A major initiative, undertaken by industrial and developing countries alike, was the reduction of heat transmission through the building fabric by increasing the insulation standard. As the insulation qualities of buildings have improved, the proportion of energy lost by air infiltration through the building fabric has steadily increased. In some cases air change now accounts for half the total energy requirement for the building. It is for this reason that air infiltration has taken on such significance (Hayati, 2017).

Infiltration air change rate within a building (if considered as single zone) can be calculated provided that the following quantities are known or can be determined: wind speed and direction; indoor-outdoor temperatures; the size, position, and flow characteristics of all openings within the envelope. In practice however, it is almost impossible to have information on all these quantities and hence theoretical models developed for infiltration calculations normally make some simple assumptions of what happens in reality. These simplifying assumptions retain the main physical concepts but enable an acceptable mathematical interpretation of the airflow process to be formulated (Miszczuk & Heim, 2021).

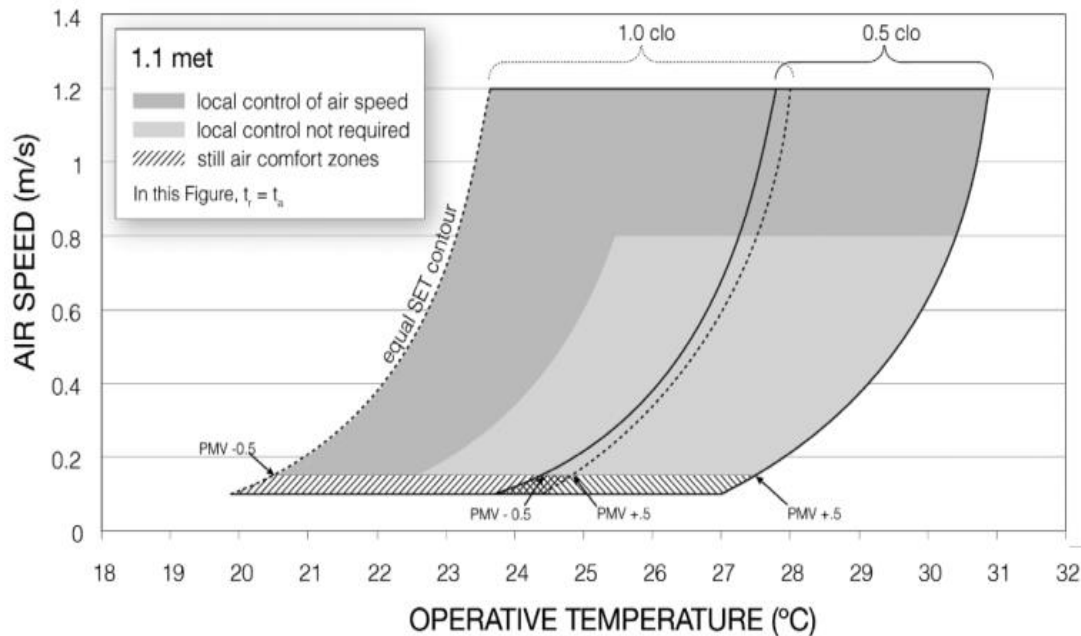


Figure2. 2 Acceptable range of operative temperature and air speeds for the comfort zone at humidity ratio 0.010Kg H₂O/Kg dry air (Sun & Smith, 2005).

2.7.3.1 Air flow characteristics

The air flow pattern and temperature distribution within a given room play important roles in determining the quality of the indoor environment and the amount of energy needed to condition the air for the room. By using a CFD simulation, the reliability and accuracy of results depend on several factors, such as the turbulence model, the numerical scheme, and the modeling of the boundary conditions. The supply air boundary condition is a crucial input because the momentum from the supply air diffuser dominates the room air flow (You et al., 2021).

- **Supply air diffuser:** the comfort of users of buildings must be at the highest level, regardless of the purpose of a given facility. Many factors directly affect the comfort of users. Besides the individual parameters related to users' activities, metabolism, and the work performed, the parameters of the air surrounding the user, including temperature, humidity, and air velocity, are crucial (Borowski et al., 2018).

In ventilating systems, the most popular group of supply air diffusers used are ceiling diffusers. The advantage of this type of supply components arises from the popularity of overhead mixing ventilation. Easy access to the ceiling space as well as available space and easy installation of ventilation increase the popularity of this type of solutions. Ceiling

diffusers can generate a limited or semi limited, attached to the ceiling, airflow. Depending on the type and structure of the supply air diffuser, the airflow can be swirling (with adjustable or nonadjustable blades), formed in a given direction (one-way, two-way, three-way, or four-way air discharge) (Borowski et al., 2021).

Slot diffusers, perforated ceilings or, in tall structures, long-range nozzles are often used in addition to typical supply air diffusers. The stability of the supply airflow moving in the room depends on the supply air diffuser's structure. From the point of view of proper air distribution in the room, the range of influence of the supply airflow and its parameters (temperature differential, velocity) at the input into the occupied zone are of great importance (Balaras, n.d.).

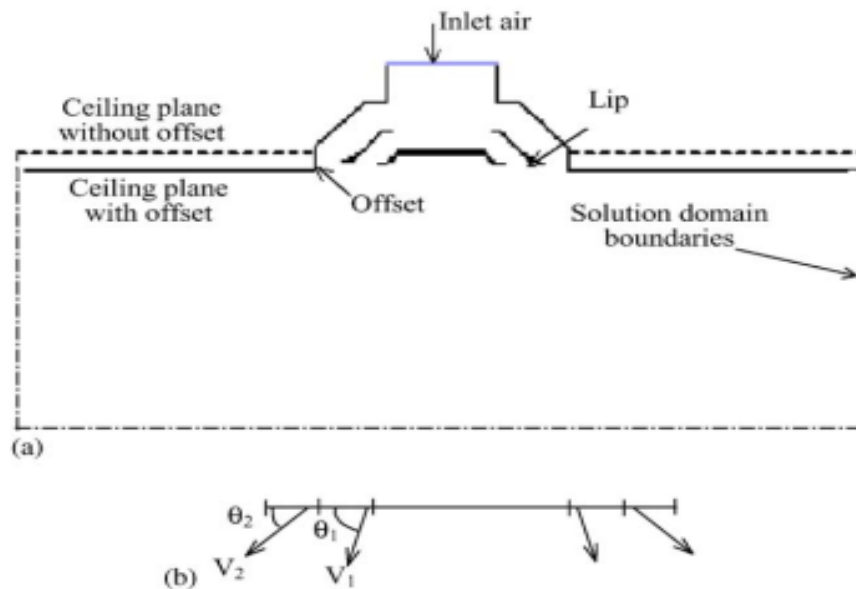


Figure2. 3: Diffuser model: (a) 2D diffuser model; (b) sectional view of the diffuser (Borowski et al., 2021).

- **Quantity of air infiltration:** The determination of the quantity of air infiltration in buildings is an important task complicated by the changing nature of wind speeds and direction, coupled with the buoyancy effect. Infiltration air change rate within a building (if considered as single zone) can be calculated provided that the following quantities are known or can be determined: wind speed and direction; indoor-outdoor temperatures; the size, position, and flow characteristics of all openings within the envelope (Lis & Lis, 2021).

In practice however, it is almost impossible to have information on all these quantities and hence theoretical models developed for infiltration calculations normally make some simple assumptions of what happens in reality. These simplifying assumptions retain the main physical concepts but enable an acceptable mathematical interpretation of the airflow process to be formulated (Etheridge and Sandberg, 1996). They proposed that the chosen equation for prediction of airflows in buildings satisfy two requirements (Sherman and Grimsrud 1980, CEN 1997, Kronvall 1978, and ASHRAE 1989).

- ✓ First, it should closely describe the flow characteristics of the range of pressure differences normally encountered with ventilation (low pressures).
- ✓ Second, it should satisfy the range of pressure differences used for experimental measurements of leakage (high pressures).

The power law equation assumes that air flow through adventitious openings is a developing flow, which is a combination of laminar and turbulent flows due to the complex nature of the driving forces (Shrestha et al., 2019).

- **Air tightness measurements:** The air tightness of the test buildings was measured by using fan pressurization method at a 50 Pa pressure difference. The ASHRAE 1989 gives recommendations regarding the level of air tightness. For detached single-unit dwellings and linked building, air tightness equivalent to an hourly air leakage of three times the building's volume (3 ach), at a pressure difference of 50 Pa is considered reasonable (Gusten 1989).
- **Air infiltration:** Air infiltration contributes to a heat loss typically representing up to one third of the heating demand of a building. The building airtightness, also quantified as air leakage, is the fundamental building property that impacts infiltration. The steady (de)pressurization method (blower door) is the widely accepted standard process for measuring building air leakage. However, this method requires the enclosure to be pressurized to a typical range of 10-60 Pa, which is not physically representative of the pressures experienced by buildings under natural conditions. The Pulse technique is a novel alternative method, which measures air leakage at low pressures; quoting it at a reference pressure of 4 Pa (Pasos et al., n.d.).

2.7.3.2 Models for air infiltration

Directive 2012/27/EU of the European Parliament and of the Council on energy efficiency, great attention is paid to the need to increase the energy efficiency of buildings. It is commonly recognized that airtightness has a large impact on the energy efficiency of buildings, which directly affects the level of CO₂ emissions. More and more recent studies and publications pay attention to the impact of air infiltration on the level of energy losses. Some analyzes show that infiltration is responsible for even 13%–30% of energy losses during the heating season and 4%–14% in the cooling season, while others found that the impact was more significant (even up to 50%). Moreover, airtightness is an important indicator of the building quality, as it affects not only the thermal parameters of the building but also the indoor air quality, the level of thermal comfort in new and modernized buildings, and the long-term durability of buildings (Miszczuk & Heim, 2021).

Regarding the number of airflow network models found in building publications, it appears that the calculation of airflow is of great importance for detailed building airflow network modelling computer codes. In fact, these airflows most often constitute an important thermal coupling between the building and the outside and the different thermal zones. It is thus, that during the period when heating a residential building, approximately 30% of the energy loss is due to air renewal. Furthermore, the airflow exchanged between two different zones, separated by a standard doorway with a temperature difference of 0.1 K is approximately 120 m³ /h. These airflows also play a major role in humidity displacement in the building (comfort, localized condensation risks), the dispersion of pollutants (smoke, gas flows ...), or in industrial and farming buildings (Hayati, 2017).

Uncontrolled air flows through the building enclosure can have a significant impact on energy consumption for heating and cooling. Moreover, the infiltrating air can transport outdoor contaminants indoors with varying efficiency: particulate pollutants are captured depending on their particle size, whereas gaseous pollutants are generally not removed during infiltration through the building envelope. Infiltration is also important from the viewpoint of security because accidental or intentional releases of toxic or hazardous materials can penetrate indoors with the infiltrating outdoor air (Yuan 2000). Although it is likely that concentrations of threat agents would be diluted in outdoor releases due to dispersion to the atmosphere, the number of potentially

exposed persons could be very high in densely occupied spaces such as transport hubs (Ng et al., 2020).

- **Theoretical model:** The presented method for determining infiltration is based on a relatively simple indoor particle mass balance model that is solved numerically using a forward time-marching method. The inputs for the model are obtained from existing data and from measurements. Under well-mixed conditions, the mass balance of particulate contaminant concentrations in a mechanically ventilated building without indoor sources can be expressed as.

$$Vdc = [q_{inf} p(d)C_{OA}(t, d) + qC_{OA}(t, d)(1 - E(d)) - qC(t, d) - q_{EXF}C(t, d) - kvC(t, d)].dt \quad (2.2)$$

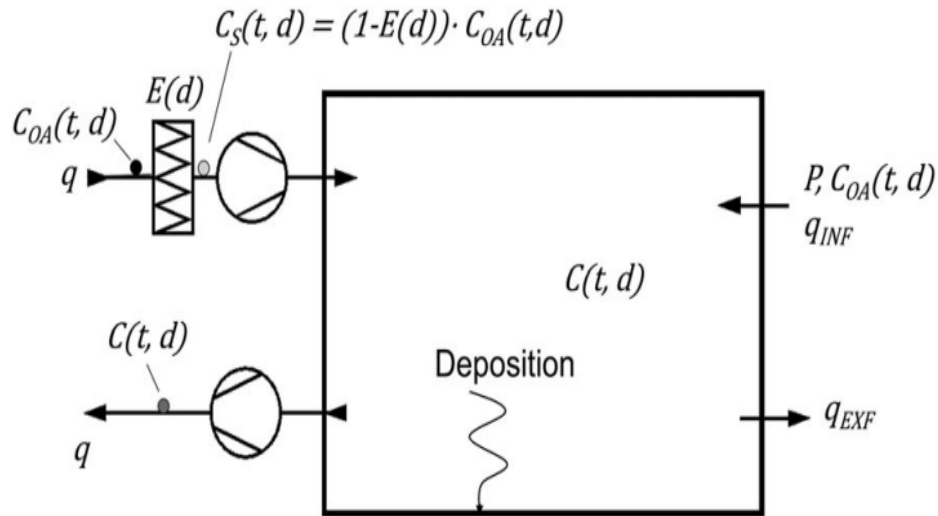


Figure2. 4: Schematic of the mechanically ventilated space and sampling points for the particle concentrations (no recirculation of room air) (Kulmala et al., 2020).

Where V is the volume of the building (m^3), $C(t, d)$ is the indoor and $COA(t, d)$ the outdoor air particle size-resolved, time-dependent concentration, and q is the mechanical ventilation flow rate (m^3/h). $P(d)$ is the contaminant penetration defined as the fraction of particles that enters through the building envelope with the infiltrating air, q_{inf} is the infiltration and q_{EXF} the exfiltration flow rate, and $E(d)$ is the removal efficiency of the supply air filter as a function of particle size d . The term k is the first-order indoor particle deposition loss-rate coefficient ($1/h$) and takes into consideration both gravitational settling on upwards-facing horizontal surfaces as well as deposition on other surfaces. In general,

outdoor concentration varies with time such that an analytical solution to Equation (1) is not possible. Therefore, the indoor concentration was solved numerically using a forward time-marching method with a time step of change in time (Hayati, 2017).

$$C(t_{i+1}) = C(t_i) + \Delta C_i \quad (2.3)$$

$$\Delta C = [q_{inf} p C_{OA}(t_i, d) + q C_{OA}(t_i, d)(1 - E(d)) - q C(t_i, d) - q_{ExF} C(t_i, d) - k_v C(t_i, d)] \cdot \frac{\Delta t}{V} \quad (2.4)$$

Table 2. 3: Estimation of parameters in Equation.

Parameter	Symbol	Value	Based on
Building volume	V	4000 m ³	Floor plan, room height
Mechanical ventilation flow rate	q	1 m ³ /s	Pressure drop characteristics of supply air filter
Infiltration/exfiltration flow rate	$q_{INF} = q_{ExF}$	Calibrated to data from unoccupied periods	Balanced ventilation assumption (Hurel et al. 2016)
Outdoor air concentration	$C_{OA}(t, d)$	varies	Measurements (optical particle counter)
Penetration factor	$p(d)$	0.97 (range 0.87–1.0)	Mosley et al. 2001
Deposition loss	K	0.04 1/h (range 0.01–0.07 1/h)	Riley et al. 2002; Lai and Nazaroff 2000
Filtration efficiency	$E(d)$	varies	Measurements (optical particle counter)

- **airflow network-building model:** The basic theory of this approach can be found among others, the buildings selected for the analyses were developed in the form of a computational model in the ESP-r (Environmental Systems Performance—Research) simulation tool. The building's space with controlled parameters as well as unheated zones was divided into a control volume represented each specific room (Antretter & Pallin, 2019).

All possible air exchanges between individual zones as well as between zones and external environment were defined, including the type of flow mechanism and its intensity. On the building models, nodes of the airflow network were plotted, and the air flow mechanism and its intensity were determined using the network method (Gurung et al., 2017).

2.7.3.3 Infiltration driving forces

Infiltration airflow is driven by a pressure gradient across the building envelope. The driving pressure is composed of two primary components: wind pressure and stack pressure/stack effect (a present ventilation system further contributes to the driving pressure gradient). The stack pressure is a function of the building height and ambient air temperatures (indoor and outdoor), while the wind pressure is mainly affected by the wind velocity, wind direction, local terrain and topography, and building shape characteristics (Shrestha et al., 2019).

- **Stack effect:** The roots of the stack effect lie in the temperature differences across the building envelope. As expressed by the ideal gas law Equation below, air density is a function of temperature. Therefore, the different temperatures on the inside and outside of a building result in a difference between the inside and outside air densities (Antretter & Pallin, 2019).

$$\rho = \frac{P}{RT} \quad (2.5)$$

Where ρ represents air density (kg/m^3), P represents air pressure (Pa), R represents the specific gas constant, and T represents temperature ($^{\circ}\text{C}$).

$$P = \rho gh \quad (2.6)$$

Where g represents gravitational acceleration constant (m/s^2) and h represents height of the air column.

$$P_{in}(y) = p_{oin} - \rho_{in}gy \quad (2.7)$$

$$P_{out}(y) = p_{out} - \rho_{out}gy \quad (2.8)$$

$$\Delta P_s(y) = (p_{oin} - p_{oot}) - gy(\rho_{in} - \rho_{out}) \quad (2.9)$$

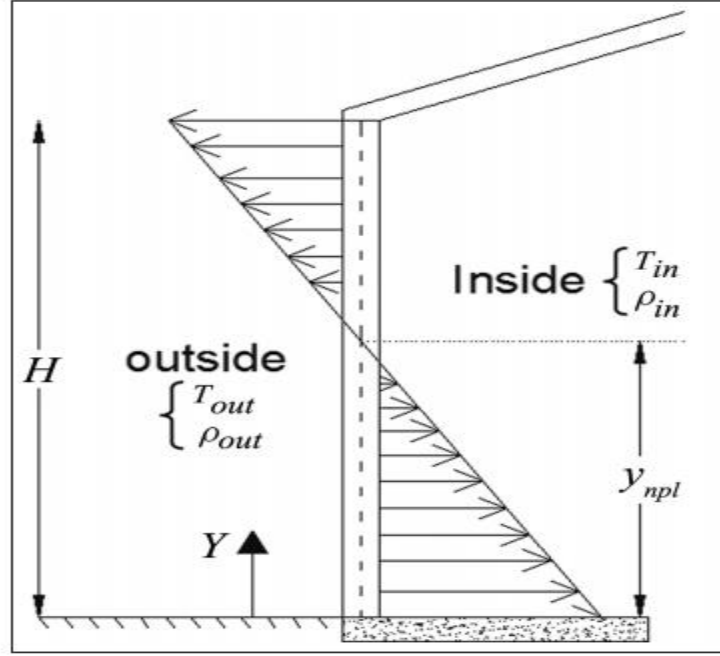


Figure2. 5: Stack effect and NPL on a building wall (Antretter & Pallin, 2019).

- **Wind pressure:** The flow of air around a building creates a wind pressure on the building. The pressure intensity, distribution, and nature of the building envelope vary by location. The wind pressure intensity depends on the air density, wind speed, and building shape characteristics expressed as a pressure coefficient, C_p . The general expression of the wind pressure intensity is given by (Happle et al., 2017).

$$P_w = \frac{1}{2} \rho_{air} C_p v^2 \quad (2.10)$$

Where P_w represents wind pressure (Pa), ρ_{air} represents outside air density (kg/m^3), and v represents wind speed (m/s).

Wind pressure coefficients, C_p , are no dimensional coefficients that represent the wind pressure distribution over a body and from which the wind pressure at a specific location over the building envelope can be calculated based on the local wind velocity (Gowri et al., 2009).

Pressure coefficients are generally calculated from data obtained through experimental testing, wind tunnel, or full-scale tests, on various building shapes and heights (Hayati, 2017).

$$C_p = \frac{p - p_o}{\frac{1}{2} \rho_{air} C_p V^2} \quad (2.11)$$

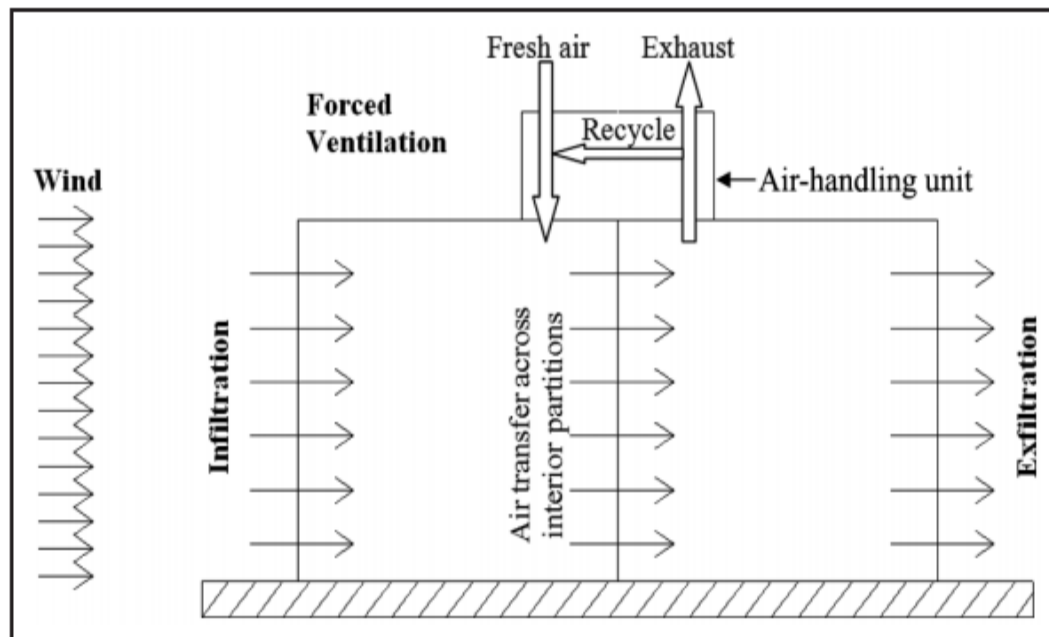


Figure2. 6: Typical infiltration and ventilation airflow (Hayati, 2017).

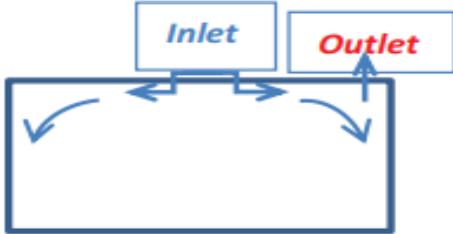
2.7.4 Building Office Air Distribution Systems

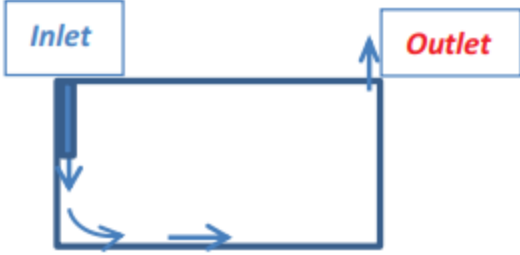
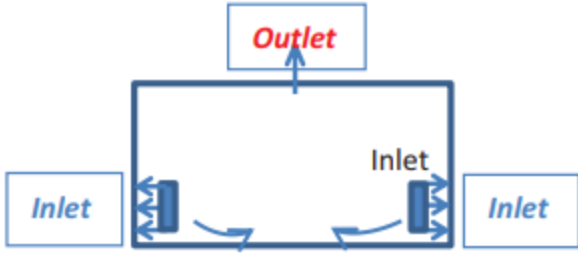
There was various types of mechanical ventilation and air distribution systems that used for buildings; by choosing those systems that are capable of providing better IAQ and energy efficiency for low energy buildings that can provide the necessary levels of IAQ for the occupants.

- **Status of Mechanical Ventilation and Air Distribution Systems:** Ventilation is the process of replacing contaminated indoor air with fresh air from outside the building. This can be fortuitous in the form of air leakage through cracks and openings in the building envelop (air infiltration) or purposely provided ventilation in the form of natural, mechanical, or a combination of the two (hybrid or mixed-mode). In mechanical ventilation, the airflow is distributed by means of fans and ductwork arrangement throughout the building and then distributed in the room via air terminal devices or diffusers.

The focus of this article is on the current status of mechanical room air distribution systems with a particular emphasis on recently developed methods of air distribution.

Table 2. 4: A summary of room air distribution types.

Type	Locations of air inlet and outlet	Main characteristics
Mixing ventilation (wall supply and extract)		➤ High momentum flow ➤ Uniform mixing through the room ➤ Limited IAQ and energy efficiency
Mixing ventilation (ceiling supply and extract)		➤ High momentum flow ➤ Uniform mixing through the room ➤ Suitable for large spaces
Displacement ventilation (wall supply and extract)		➤ Buoyancy-driven stratified flow, High ventilation effectiveness

Type	Locations of air inlet and outlet	Main characteristics
Impinging jet ventilation (ceiling supply and extract)		➤ Medium momentum flow ➤ Deep airflow penetration ➤ Can be used for cooling and heating ➤ High IAQ and energy efficiency
Confluent jets ventilation (wall supply and ceiling extract)		➤ Medium momentum flow ➤ Deep airflow penetration ➤ Can be used for cooling and heating ➤ High IAQ and energy efficiency

2.7.5 Duct systems

The duct systems convey the purified air from the air conditioning equipment to the proper air distribution point and also carry the return air from the room back to the air conditioning equipment for reconditioning and recirculation. Duct systems are also classified as high pressure or high velocity ductwork and low pressure or low velocity ductwork (*Air Duct Systems 2004–2007*, 2007).

The term high-pressure or high-velocity ductwork includes ductwork systems and plenums from the fan discharge to the final high-velocity mixing boxes, or other final pressure-reducing devices or any air supply system served by a fan operating with a static pressure range of 3 inches to 7 inches of water column (WC) (Nasr & Nasr, 2019).

High-velocity or high-pressure systems with fan static pressures of 3 inches WC or greater are defined as high pressure. Usually the static pressure is limited to a maximum of 7 inches WC, and duct velocities are limited to 4,000 feet per minute (fpm). Systems requiring pressures more than 7 inches WC are normally unwarranted and could result in very high operating costs. Systems with velocities more than 4,000 fpm performs satisfactorily when all duct fittings are carefully designed and installed. However, velocity pressure losses are excessive and velocities more than 4,000 fpm are not recommended (Bhatia, 2012).

2.7.5.1 Types of Duct Systems

Since there are many possibilities for an adequate duct system, one such system is modified to fit the needs of two different residential configurations.

- **Radial system:** The radial duct system in its simplest form consists of a central supply plenum that feeds a number of individual branch ducts arranged in a generally radial pattern. It also can be designed and sized so that each individual run leaving the plenum can feed two or more supply outlets. This is frequently the case because of the number of supply outlets required to condition the structure successfully and the amount of space at the plenum available for take offs. The radial system commonly is applied in attics, crawl spaces, and in slab on grade installations. It can be used with up flow, down flow, or horizontal air handlers and furnaces (Yang & Xing, 2021).
- **Extended plenum system:** The extended plenum duct system generally consists of one or two boxlike pieces of ductwork extending from the main plenum at the indoor unit. This

extended plenum has the same dimensions (height and width) from the starting collar to the end of the run. Branch runs to feed the supply outlets are tapped into the extended plenum(s). The best results are achieved when the maximum length of the extended plenum is not greater than 24 ft from the air handler or furnace. If two plenums are used, this total length can be extended to 48 ft. If longer distances are required based on the physical layout of the structure, consideration should be given to using one of the other designs discussed below (such as the reducing plenum or the reducing trunk duct system). There is another area of concern with the extended plenum system because of the higher velocities in the plenum, it is possible that the branches closest to the indoor blower may not feed the desired amount of air (cfm) (Nomaam, 2019).

- **Reducing plenum system:** The reducing plenum duct system can be used when the physical size or layout of the structure calls for greater distances than the length constraints imposed on the extended plenum (24 ft). The concept of the reducing plenum system is simple when the air velocity lost to the branch runs reaches approximately 50%, the plenum size is reduced to regain the velocity in the remaining portion of the plenum. This reduction also improves the air flow characteristics at the branch ducts that are closest to the air-handling unit (Kuto et al., 2018).
- **Perimeter loop system:** The perimeter loop duct system is well-suited for buildings that are constructed using concrete slab on grade. It generally performs better than the radial system in such applications, especially in cold climates. However, the perimeter loop system does have the disadvantage of being a little more difficult to design and more expensive to install. It is basically laid out around the perimeter of the structure next to the edge of the slab. The entire perimeter loop is the same size duct. The loop is fed by four or more ducts radiating out from the central plenum. They are usually the same size as the loop duct. The boot boxes are sized to deliver the proper cfm to each room of the structure (Basavaraja et al., 2019).

2.7.5.2 Supply Duct System Locations

Decisions regarding the location of a supply air distribution system should be made based on the winter design temperature for the structure's geographic location. This information should be consulted to ensure that the proper type and location of duct system is selected for the structure in

question. The general guidelines state that if the winter design temperature for the location of the structure is above 35°F, then both perimeter floor and ceiling distribution systems will provide satisfactory results. If the winter design temperature for the location of the structure is below 35°F, the ceiling distribution system is not recommended and the floor distribution system should be considered. A modified type of ceiling distribution system can be used if the registers are moved closer to the outside walls and the primary air is directed out of the occupied zone and toward the window and door openings (Jin et al., 2019).

Supply Air Outlet Locations

The designer must select locations that will deliver the conditioned air into the space to be conditioned in such a way that the high heat gain/heat loss load areas (doors and windows) are covered by the primary airstream. In addition, the secondary air (room air) must mix with the primary airstream to ensure even temperatures throughout the room. The design should accomplish these objectives while keeping the primary airstream from entering the occupied zone within the room (Kulmala et al., 2020).

By delivering the primary air into the unoccupied zone of the room and mixing it with the secondary air, the designer ensures that the primary airstream does not come in direct contact with the room occupants. If the primary airstream is allowed to come in contact with the room occupants, the result will be complaints of drafts and the overall comfort of the occupants will suffer (Sanchez et al., 2018). In most cases (depending on the location of the duct system and the location of the outlet), this problem is next to impossible to solve without the investment of major time and expense. The best time to prevent this potential problem is when the system is being designed and installed. There are several specifications listed in the engineering tables for outlet devices that are very important to the designer (Borowski et al., 2021).

2.7.5.3 Return Air Duct Systems

The return air duct system is just as important as the supply system, and too often it is largely ignored. In many cases, the return system consists of one or two improperly sized openings connected back to the air handler through improperly sized return ducts or through wall, floor, and ceiling framing cavities. Although panned framing members can be used for return systems, they

generally should be avoided. These types of installations are next to impossible to seal (Borowski et al., 2018).

- **Central Return Systems:** Central return systems can be applied in smaller structures with success. These systems consist of one or more return inlets, strategically located within the structure and sized to return the total amount of the supplied air volume to the indoor blower. Both the inlet box and the return connecting ductwork should be sized where the room air will flow back to the blower at a relatively low velocity. As emphasized above, the return duct system must be free of leaks (Borowski et al., 2018).

It is sometimes desirable to have some turns or other fittings in the duct run in order to prevent blower noise from being transmitted down the duct and exiting the return air grille. The return air box should be lined with sound attenuation insulation and insulated on the outside to prevent or retard heat gain and heat loss (Lazzeretti & Capone, 2015).

- **Individual room return systems:** In some structures it may be desirable to install individual room returns. Of course, this means that the return system is just as elaborate as the supply system. Each individual room return is sized to return the same amount of air that is delivered to the room by the supply system. The results are excellent if the layout and sizing are done properly in fact, there is no other return system that performs as well as the individual room return system. The major drawback to this approach is the initial cost (Awbi, 2015).

2.8 Evaporative cooling system

(Shakerin and Contreras, 1998) an instrumented evaporative cooler was developed for use in an undergraduate laboratory in psychometrics. The air temperature and relative humidity at the inlet and outlet, and the air flow and water consumption rates were measured. Given the experimental data for the inlet air and water, the outlet air properties were predicted using psychometric equations, and a good comparison was found with the experimental result.

(Qureshi and Zubair, 2007) the accurate prediction of various aspects of thermal behaviour of evaporative fluid coolers is very important for both design and rating calculations. Exactly predicting evaporation losses is significant since the process fluid is cooled primarily by evaporation of a portion of the recirculating water that causes the concentration of dissolved solids and other impurities to increase. An empirical relation to predict evaporation losses is developed

on the basis of the rule of thumb recommended by manufacturers, which is simple and accurate with a wide range of applicability. The predicted values are in good agreement with the numerical values obtained from the calibrated model where the maximum error was found to be approximately 4% but often s of heat and mass transfer in primary and secondary air and water flows

(Heiderinejad and Bozorgmehr, 2007) indirect evaporative cooling in an air cooler has been modelled. This model has been obtained from the governing equation of heat and mass transfer in primary and secondary air and water flows. Factors affecting on evaporative cooling performance such as mass flow rates, geometry and flow configuration has been investigated. Results shows that cooling efficiency considerably depends on mass flow rates ratio of primary and secondary air flows and spacing between plates of wet and dry passages. The performance of this cooling system for typical conditions of some cities has been investigated. Using a first stage indirect evaporative cooling prior to conventional direct evaporative cooling systems in most regions of Iran will provide cooling comfort conditions as a low energy consumer alternative.

(Mizushina, 2008) developed two different rating methods for evaporative coolers; one, a numerical procedure and the other, a straightforward analytical model.

(Finlay and Grant, 2009) simplified the equations describing the mass transfer in an evaporative cooler by assuming that the vapour pressure of saturated moist air is a linear function of temperature. The model can be expected to be very accurate, as this is the only major assumption made in the analytical formulation. The final design equations are somewhat complicated and therefore require a numerical solution procedure.

(Effatenjad and Salehian, 2009) a new method for energy labelling of evaporative air coolers in Iran with national standard number of 4910-2. Procedure of test has been developed typically in Iranian standard IS3315-1974 and Australian Standard AS 2913-1978. The criteria are calculate based on energy consumption and evaporative air cooler test results in the past three years and have developed experimentally and analytically. EER (Energy Efficiency Ratio) is defined for energy labelling in Iran. Index of the previous standard of energy consumption and cooler label is used sensible cooling capacity of air power of the total input. EER is the sensible cooling capacity of air power of the total input power as well.

(Mohammed, 2012) Evaporative air cooler are an efficient and effective machine for cooling. As a direct placement for air conditioning in dry climates like Iraq, it is an example of how man can work with nature. Being so much less expensive than air conditioning, Re circulated spray of water is used to provide cool air. In order to perpetuate the process it is necessary to use a water pump to continuously re circulated spray of water. After the adiabatic saturation has achieved a steady state condition, certain combinations of air will result in a given wet bulb temperature and can be defined by writing an energy balance about the saturation. This energy balance, written on the basis of unit mass flow of air.

$$h_1 = h_2 - (w_o - w_i) * h_f \quad (2.12)$$

If the heat exchange contact between the air and water spray is 100%, the dry bulb temperature of the entering air should drop down to its wet bulb temperature, the air should become fully saturated. This is however never achieved because of the bypass factor, a small portion of the air will not come in contact with spray and so it comes out of the saturated at the same condition as it entered. Hence the total air quantity at the outlet of the saturated is a mixture of the by passed and the saturated air. Thus the evaporative cooler efficiency (or saturation efficiency) of the saturation is expressed as %.

$$\eta = \frac{t_{dbi} - t_{dbo}}{t_{dbi} - t_{wbi}} * 100\% \quad (2.13)$$

The mass flow rate of air that exit from evaporative cooler can be calculated:

$$m_o = \rho_a * Q_a \quad (2.14)$$

The air that enter the cooler include quantity of water vapour calculated by equation:

$$m_i = m_{wi} + m_a \quad (2.15)$$

$$m_a = \frac{m_i}{1 + w_i} \quad (2.16)$$

$$w_i = \frac{m_{wi}}{m_a} \quad (2.17)$$

By the same method that apply on air that exit from air cooler to get:

$$m_a = \frac{m_o}{1 + w_o} \quad (2.18)$$

$$w_o = \frac{m_{wo}}{m_a} \quad (2.19)$$

The optimum time that needed to reach for the suitable temperature of air that exit from air cooler calculated from this relationship:

$$\tau = \rho_w * \frac{V}{m_v * 60} \quad (2.20)$$

The electrical power required to move a volume rate of air through the Packing (P_s) is define the power that consumption air cooler engine to decrease the one kilogram temperature of air one siliceous degree.

$$P_s = \frac{P}{m_i * (t_{dbi} - t_{dbo})} \quad (2.21)$$

The relative density of packing is the ratio between the densities of packing to the material density that manufacturing from her packing.

$$\rho_s = \frac{\rho_a}{\rho_p} \quad (2.22)$$

$$\rho_a = \frac{m_p}{v_t} \quad (2.23)$$

$$v_t = A_t * t \quad (2.24)$$

The total volume of packing (V_p)

$$v_p = \frac{m_p}{\rho_p} \quad (2.25)$$

The mean velocity of air that passes the packing calculate from relationship:

$$U_{ap} = \frac{Q_a}{A_a} \quad (2.26)$$

$$A_a = A_t - A_p \quad (2.27)$$

To find the (A_p) equals the ratio between total volume of packing to total volume of window and ratio between section areas of packing material to section area of window orifice.

$$\frac{v_p}{v_t} = \frac{A_p}{A_t} \quad (2.28)$$

(Abaranji et al., 2020) Adaptive cooling system was also a conditioning of air filter out contaminates in the air. As shown in Fig, in an air filter air comes in direct contact with a spray of water and there will be an exchange of heat and mass (water vapour) between air and water, the outlet condition of air depends upon the temperature of water sprayed in the air washer. Hence, by controlling the water temperature externally, it is possible to control the outlet conditions of air, which then can be used for air conditioning purposes.

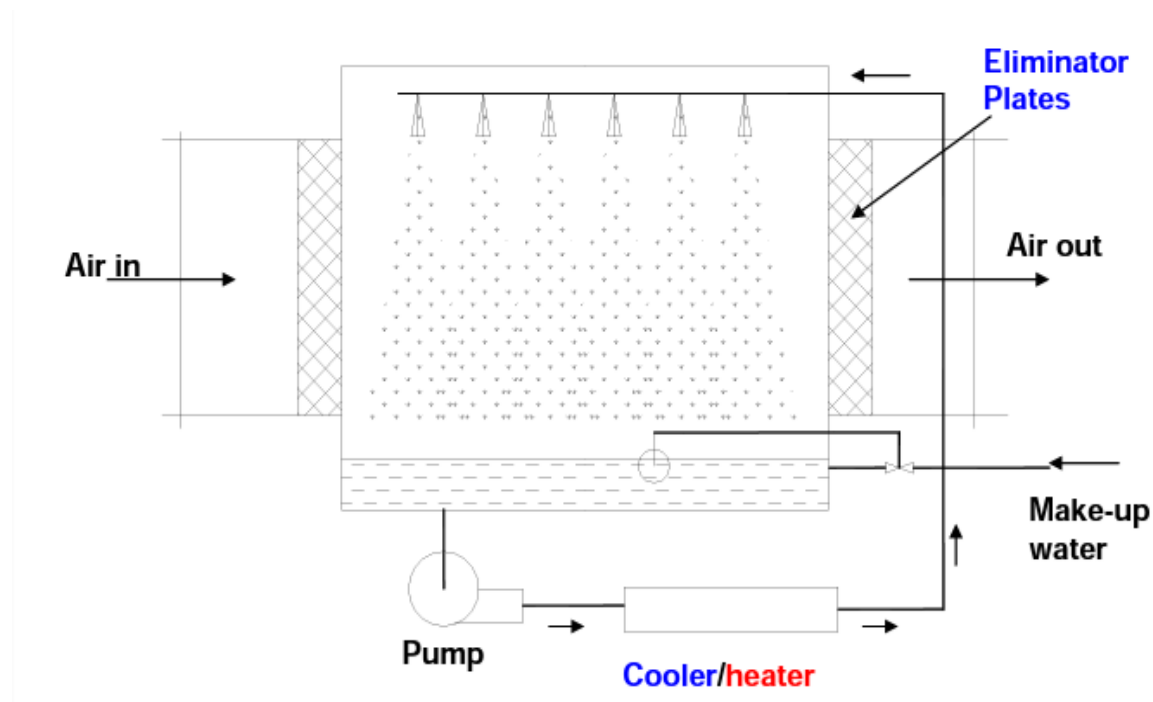


Figure 2. 7: air washer (air filter) (Klein & Reindl, 2017).

Working principle of evaporative cooling air washer system

In the air washer, the mean temperature of water droplets in contact with air decides the direction of heat and mass transfer. As a consequence of the 2nd law, the heat transfer between air and water droplets will be in the direction of decreasing temperature gradient. Similarly, the mass transfer will be in the direction of decreasing vapour pressure gradient.

- ***Cooling and dehumidification: $t_w < t_{DPT}$*** : Since the exit enthalpy of air is less than its inlet value, from energy balance it can be shown that there is a transfer of total energy from air to water. Hence to continue the process, water has to be externally cooled. Here both latent and sensible heat transfers are from air to water. This is shown by Process O to A. If unsaturated moist air is cooled at constant pressure, then the temperature at which the moisture in the air begins to condense is known as *dew-point temperature (DPT)* of air.
- ***Adiabatic saturation: $t_w = t_{WBT}$*** : Here the sensible heat transfer from air to water is exactly equal to latent heat transfer from water to air. Hence, no external cooling or heating of water is required. That is this is a case of pure water recirculation. This is shown process O-B in figure. This is the process that takes place in a perfectly insulated evaporative cooler. WBT- wet bulb temperature is defined as that temperature at which water, by evaporating into air, can bring the air to saturation at the same temperature adiabatically.
- ***Cooling and humidification: $t_{DPT} < t_w < t_{WBT}$*** : Here the sensible heat transfer is from air to water and latent heat transfer is from water to air, but the total heat transfer is from air to water, hence, water has to be cooled externally. This is shown by Process O-C in Fig.

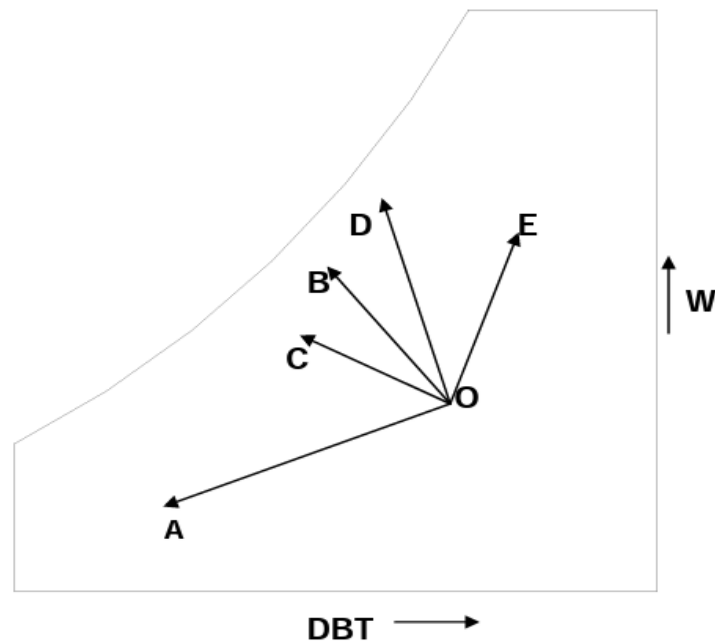


Figure 2. 8: Psychrometric chart

- **Cooling and humidification:** $t_{WBT} < t_w < t_{DBT}$: Here the sensible heat transfer is from air to water and latent heat transfer is from water to air, but the total heat transfer is from water to air, hence, water has to be heated externally. This is shown by Process O-D in Fig. This is the process that takes place in a cooling tower. The air stream extracts heat from the hot water coming from the condenser, and the cooled water is sent back to the condenser. Dry bulb temperature (DBT) is the temperature of the moist air as measured by a standard thermometer or other temperature measuring instruments.
- **Heating and humidification:** $t_w > t_{DBT}$: Here both sensible and latent heat transfers are from water to air, hence, water has to be heated externally. This is shown by Process O-E in Fig. Thus, it can be seen that an air washer works as a year-round air conditioning system. Though air washer is a and extremely useful simple device, it is not commonly used for comfort air conditioning applications due to concerns about health resulting from bacterial or fungal growth on the wetted surfaces. However, it can be used in industrial applications.

Evaporating cooling system using air washer

Evaporative cooling has been in use for cooling water and for providing thermal comfort in hot and dry regions. This system is based on the principle that when moist but unsaturated air comes in contact with a wetted surface whose temperature is higher than the dew point temperature of air, some water from the wetted surface evaporates into air. The latent heat of evaporation is taken from water, air or both of them. In this process, the air loses sensible heat but gains latent heat due to transfer of water vapour. Thus the air gets cooled and humidified. The cooled and humidified air can be used for providing thermal comfort.

From those three type of evaporative cooling method direct evaporative cooling method where applied in this research study. In direct evaporative cooling, the process or conditioned air comes in direct contact with the wetted surface in this case atomized water, and gets cooled and humidified. Figure shows the schematic of an elementary direct, evaporative cooling system and the process on a psychrometric chart. As shown in the figure, hot and dry outdoor air is brought in contact with the wetted surface or spray of water droplets in the air washer. The air gets cooled and dehumidified due to simultaneous transfer of sensible and latent heats between air and water (process o-s). The cooled and humidified air is supplied to the conditioned space, where it extracts the sensible and latent heat from the conditioned space (process s-i). Finally the air is exhausted at state i. In an ideal case when the air washer is perfectly insulated and an infinite amount of contact area is available between air and the wetted surface, then the cooling and humidification process follows the constant wet bulb temperature line and the temperature at the exit of the air washer is equal to the wet bulb temperature of the entering air (t_{owbt}), the process becomes an adiabatic saturation process. However, in an actual system the temperature at the exit of the air washer will be higher than the inlet wet bulb temperature due to heat leaks from the surroundings and also due to finite contact area.

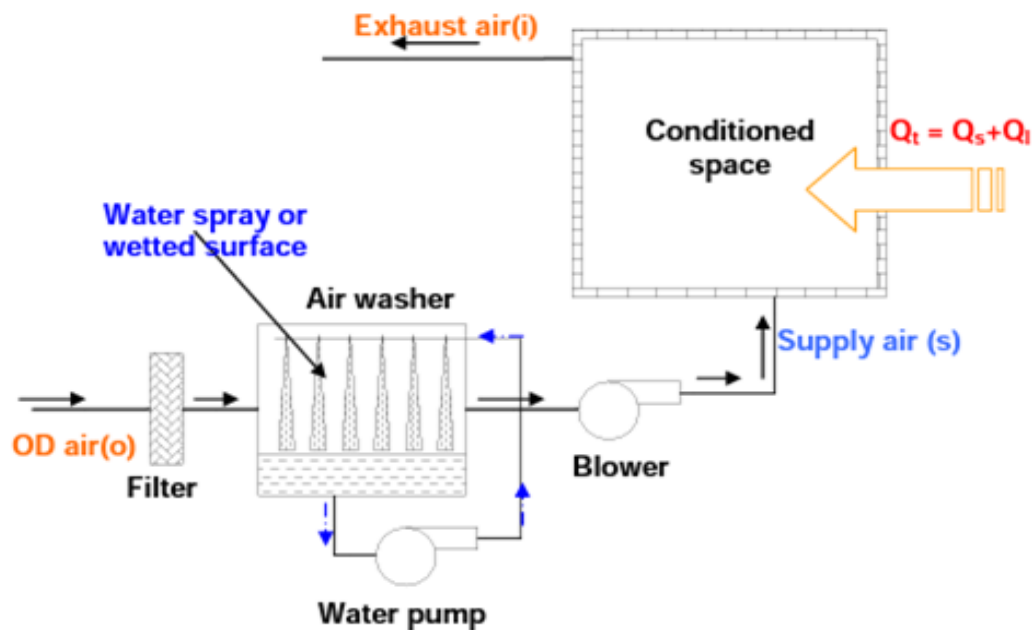


Figure 2. 9: direct evaporative cooling system

Evaporation is the change in state of a material from a liquid to a gas (vapour). The laws of physics require that in order for evaporation to occur, heat energy, called “heat of evaporation”, must be supplied. In evaporative cooling processes, where water is evaporated in an air stream, the heat needed to supply the “heat of vaporization” for the water is removed from the air. This results in a sensible cooling effect.

When air passes through Air Washer, it is brought into intimate contact with water which is recirculated from a reservoir. When the process reaches equilibrium, the temperature of the water in the reservoir is approximately equal to the wet bulb temperature of the entering air. The transfer of the heat from air to water may be assumed to occur at the wet bulb temperature of the entering air. The removal of sensible heat energy from the air stream results in lowering the dry bulb temperature. The resulting air-water mixture has a lower dry bulb temperature and a higher moisture content than the entering air.

The design can include extra features like filters to trap harmful particulates in order to increase the purity of the air. Like other devices that clean air, an air washer can eat up a lot energy. Some are designed to be more efficient and may have features that are intended to reduce energy usage.

Air conditioning may be considered to be the simultaneous control of all or at least the first three of the following factors which affect the physical and chemical conditions of the atmosphere within any structure (Walsem et al., 2018). These factors most of which affect human health or comfort include:

- **Dust:** Indoor Air Quality (IAQ) is comprised by pollutants such as dust, pollen, mould, mildew and volatile organic compounds. These pollutants can come from furnishings, laboratories, cleaning solutions and building occupants, health and problem can begin shortly after exposure to poor Indoor Air quality. Occupants may be experienced itchy eyes, headache and fatigue. Long term exposure to poor IAQ can contribute to chronic respiratory illness such as asthma (Rodriguez et al., 2019).
- **Toxic gases:** The elements in home or office secretly affect health of our environment. They cause respiratory problems or allergies, they include Toxic trio (Toluene – solvent part) in nail polish, Chemicals in cleaning products and Smoke of cigarette or other smoking (Wang et al., 2019).
- **Odors:** Odor is also associated with human comfort. Bad odor is irrelevant to human we can't survive in environment with bad odor. It is mostly generated indoor by following sources: Human sweat, cigarette, medicines, human breathe, clothing etc. and outdoor also from different chemical production factory plants etc. (Shrestha et al., 2019).
- **Temperature:** Thermal comfort is the condition of mind that expresses satisfaction with the thermal environment and is assessed by subjective evaluation. Maintaining the standards of thermal comfort for occupants of buildings or other enclosures is one of the important goals of HVAC design engineers. Thermal neutrality is maintained when the heat generated by human metabolism is allowed to dissipate, thus maintaining the thermal equilibrium with the surroundings. Temperatures of places vary with change in altitude, weather, climate and horizons (Hayati, 2017).

Room temperature is colloquial expression for the typical or preferred indoor temperature to which people are generally accustomed. It represents the small range of temperature at which the air feels neither hot nor cold, approximately 70° F (21° c) (Narayana Rao et al., 2019).

- **Humidity:** Relative humidity levels below 25% are associated with increased discomfort and drying of mucous membranes and skin, which can lead to chapping and irritation. Low relative humidity also increases static electricity which causes discomfort and can hinder the operation of computers and paper processing equipment's. High humidity levels can result in condensation within building structure and on interior and exterior surfaces and the subsequent development of mould and fungi (Guo et al., 2020).

2.8.1.2 Psychrometric condition

The psychrometric graph gives a helpful methods for speaking to the distinctive procedures which are conceivable in the operation of an air washer and can be utilized to take care of the issues of aerating and cooling which are included. The psychrometric outline indicates graphically the properties of air on the premise of one pound of dry (air free of water vapour) at standard climatic weight, 29.92 creeps of mercury (Jagemar, 1989). Psychrometric relates to psychrometric or the condition of the air with reference to dampness (Ng et al., 2020).

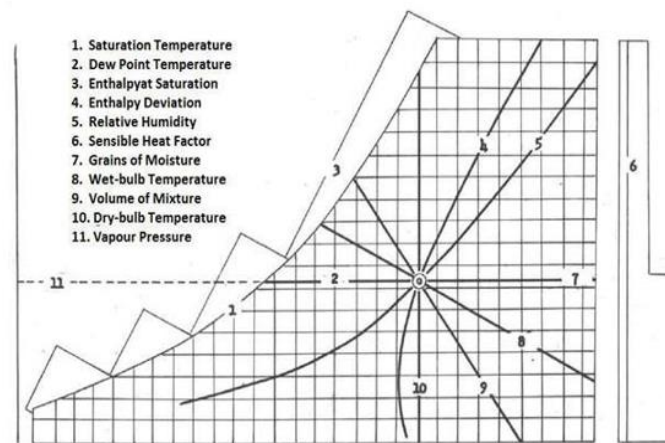


Figure 2. 10: psychrometric graph

- **Adiabatic Saturation:** When air below saturation is brought into contact with water, some or the water will vaporize, adding to the moisture content of the air. If no heat is added, moisture will be supplied entirely at the expense of the heat of the air and of the superheat or the original quantity of water vapor. Evaporation will continue and the temperature of the air will be lowered until the air is saturated with the water vapor. This process, taking place without heat transfer to or from an outside source, is called adiabatic, and the final

temperature that is approached is termed the temperature of adiabatic saturation or the wet-bulb temperature.

Actually, the mixing of water vapor and air at the wetted surface of a wet-bulb thermometer is not adiabatic because the bulb “sees” objects at the dry bulb temperature. The result is a radiation error and a higher reading than the true adiabatic saturation temperature. Also, because of the difference of rate of diffusion between the air and water vapor, a wet-bulb thermometer will tend to read lower than the true temperature of adiabatic saturation. Hence, the errors are largely compensated and account for the largely close agreement between the observed wet-bulb temperature and the true temperature of adiabatic saturation.

- ***Heat Exchange between Air and Water:*** When air is brought into contact with water at a temperature different than the wet-bulb temperature of the air, there will be an exchange of heat, as well as moisture, between the air and water. In any exchange of heat between air and water, the temperature of the water can never fall below the initial wet-bulb temperature of the air if the initial water temperature is greater than the initial wet-bulb temperature. Also, the converse is true; if the initial water temperature is below the initial wet-bulb temperature of the air, the final water temperature will never rise above the initial wet bulb temperature.

This relationship is shown above, the diagram is shown as parallel flow because the test air washer was of this type. There is only one exception to the rule that whether there is an exchange of heat occurring when air is brought into contact with water, the temperatures of both must change. This occurs in an air washer when the spray water is continually re-circulated without external heating or cooling. The temperature of the spray water will soon assume the initial wet-bulb temperature and will not change thereafter.

Aside from this, there is always a change in the temperature of the spray water when it is brought in contact with air. In order to humidify air with spray water, the temperature of the spray water must be higher than the required final dew-point temperature of the air. Such an amount of water must be used that, as the water cools from its initial temperature, its final temperature will still be above the required final dew-point temperature. When dehumidifying, the exact converse is true; the final water temperature must be below the required final dew-point of the air.

2.9 Literature summary

In the literatures, air pollution, air pollution source, indoor air pollution modeling systems, outdoor air pollution modeling systems, ventilation system, supply duct system, return duct system and evaporative cooling systems have been discussed. In most of the literatures associated with the air pollution, as air pollution are health risks and primary source of global warming. Even though an HVAC is mainly used by people, HVAC system itself noticed as the primary source of air pollution and global warming, the dispersion model was used as outdoor air pollution modelling system and box modeling where used as indoor air pollution modeling system. Comparatively, literature showed that round duct have an advantage of increasing air flow, decreasing head loose, reduce friction in building ventilation system design. Surveys of literatures indicated that considerable efforts have been invested in research and development of direct evaporative cooling system. However, limited research works were reported on performance improvement of direct evaporative cooling system. Besides, those limited works in literature focuses mostly on energy efficiency of the systems even if pollution control is the true performance measure of direct evaporative cooling system which indicate the quality of supply air to building office. Generally, in this study experimental investigating on liquid based air filtration system was studded which integrated with office ventilation system a case of Batu caustic soda and cement factory.

2.10 Research Gap

Several studies have been examined on direct evaporative cooling or passive evaporative cooling system have been found to have higher efficiency in terms of saturation effectiveness and producing humidified air for building. However, the efficiency (saturation effectiveness) of system is low (65% - 72%) and the humidity of the air from the system some time extremely high out of range (65%-90%). To overcome this problem, a cooler pad with (4-8) mm thickness provided performance enhancement and increase saturation effectiveness, techniques such as using different packing mass, increasing gap (distance) between cooler pad and supply air exit and using variable thickness of cooler pad will help to manage relative humidity and temperature in range. On the other hand, to minimize the air pollution level various researchers have used pre filter before evaporation cooling process have minimized the pollution level and improve quality of the supply air better because pre filter can remove unnecessary pollution matter. But by increasing the time of evaporative cooling process in a system and by increasing the gap between cooler pad and supply air exit it is possible to provide enough time for air to humidify and to remove pollution inside it. In addition to this by varying mostly decreasing the speed of blower it is possible to increase the process time this also will create a possibility to remove pollution from the air. It was also observed, that there is a little experimental study on the direct evaporative cooling system, therefore, this study presents an experimental evaluation of liquid based air filtration system (direct evaporative cooling) integrated with ventilation system to enhance the efficiency and quality of the air for Batu caustic soda and cement factory office building.

CHAPTER THREE

METHODOLOGY and MATERIALS

3.1 Materials

the raw materials used in this thesis includes GI Sheet metal 24 gauge, pump, cooling pad, fan, fan motor, pipe and fitting, chamber, measuring equipment. For measuring equipment, for temperature measurement of inlet and outlet water digital thermometer is used, for measuring the outlet velocity of the air Pitot tube is used, air quality detector is used to measure the humidity of the air, digital thermometer and air quality detector used for air temperature measurement. GI Sheet 24 gauge sheet is the selected as material for chamber to its higher corrosion resistance, good surface appearance, good formability for meeting the deep drawing requirements, good paint ability provided proper pre-treatment is performed, and good weld ability.

3.2 Methodology

Theoretical as well as experimental approaches used as a methodology in this study. The numerical (theoretical) approach use governing equation in each analysis of outdoor air pollution by dispersion model and indoor by using box modeling, for ventilation system design and sizing HAP 4.9 software for Load Calculation, MCQUAY software for duct design and sizing. The software's used in this thesis were SOLID work for part drawing, HAP 4.9 for ventilation system calculation, Mcquay for Duct calculation, and some analysis validation and An excel program has been used to determine cooling and heating load at different points of building office and for graph. Literature studies were performed to gain an insight to the area of interest, background, influencing regulations and research. The literature was obtained through statistics databases, academic literature, articles, internet forums, websites etc.

The experimental approach used in this thesis is to test and validation with numerically developed model results at the same working condition. Therefore, in order to achieve the required objective, this study will use both numerical and experimental method. The overall design and fabrication of central liquid base air filtration integrated with ventilation is described in flow diagram below.

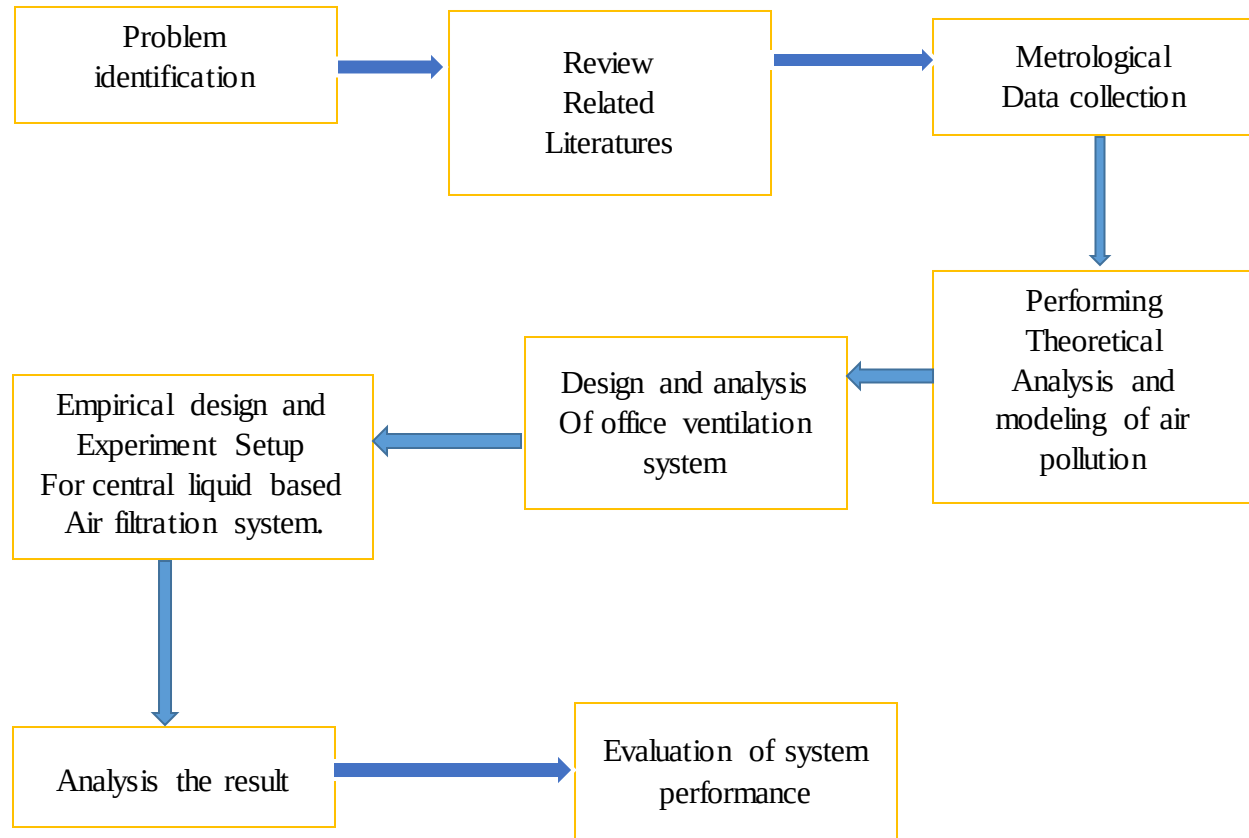


Figure 3. 1: Work flow diagram of evaporative cooling system integrated with ventilation system.

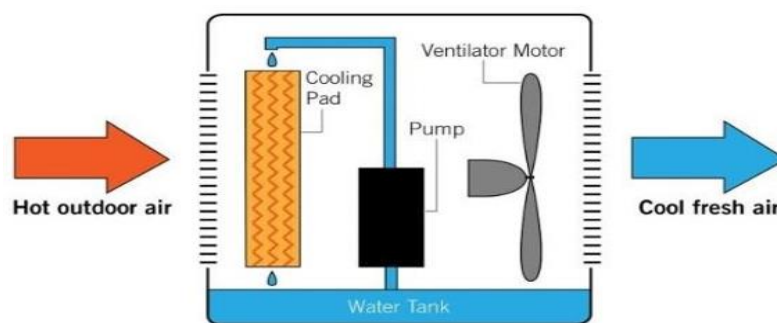


Figure 3. 2: Direct evaporative cooling.

3.2.1 Data sources and collection techniques

The data are collected from both Batu caustic soda and cement factory, metrological data were collected from Hawasa national metrology agency service and Addis Ababa national metrology agency service center. As primary sources of data are direct measure it include all data from Batu caustic soda and cement factory was primary data and secondary data sources are from above both metrology agency service, lastly saturation efficiency from Reference books and international journal.

3.2.2 Design parameters

To ensure that the system will operate at outdoor average dry bulb temperature 18 °C - 31 °C, wet bulb temperature 16 °C and indoor dry bulb temperature 21 °C, wet bulb temperature range 12 °C- 13 °C to satisfy in door humidity 59 % - 60 % and deliver air of volume 5.615 m³/s to office building each second.

3.2.3 Design Data

Table 3. 1: Emission data of pollutant (2010-2021)

Year	Compounds(μ g/m ³)					
	SO ₂	NO ₂	CO	O ₃	PM ₁₀	PM _{2.5}
2010	282.2	22.67	36648	43.07	149.8	4472
2011	281.8	22.81	36950	43.16	278.4	4509
2012	279.5	21.72	36975	43.3	331.7	4495
2013	277.9	21.24	36192	43.19	307.9	5121
2014	273.1	22.67	36162	43.26	313.6	5213
2015	281.5	21.25	35184	43.1	312.1	5220
2016	286.9	22.67	35211	43.19	317.1	5202
2017	281.5	22.13	36081	43.28	315.1	5219
2018	286.3	22.67	36743	43.33	353.7	5253
2019	284.4	21.96	36723	43.02	394	5290
2020	285.1	22.67	36489	43.21	392.2	5298
2021	289.2	22.75	36769	45.22	395.3	5327
Annual	282.5	22.3	36343.9	43.63	321.7	5030

Table 3. 2: Metrology data

Year	Parameter			
	Pressure (KPa)	Temperature (K)	Surface wind speed (m/s)	Wind speed at 10 m (m/s)
2010	81.61	298.79	4.831	7.447
2011	81.56	300	3.2	5.473
2012	81.59	299.64	4.1	5.495
2013	81.62	298.54	4.861	6.076
2014	81.64	299.15	5.118	7.522
2015	81.67	300.25	3.111	5.635
2016	81.69	299.12	2.944	5.054
2017	81.65	299.18	5.245	5.413
2018	81.64	299.09	5.222	6.407
2019	81.65	299.47	2.793	5.454
2020	81.65	298.41	5.087	5.375
2021	81.6	301.8	5.824	6.879
Annual	81.63	299.45	5.01	6.108

Table 3. 3: Outdoor dry bulb and wet bulb temperature

Year	2010	2011	2012	2013	2014	2015	2016	2017	2018	2019	2020	2021
T _{db}	25.8	27	26.1	25.5	26.1	27.2	26.1	26.2	26.1	26.44	25.4	27.1
T _{wb}	15.6	15	14.9	15.4	15.7	15.4	15.8	15	15.5	15.8	15.8	15.9

CHAPTER 4

THEORETICAL FORMULATION OF AIR QUALITY, VENTILATION SYSTEM AND CENTRAL LIQUID BASED AIR FILTRATION SYSTEM

4.1 Outdoor Air Quality

The atmospheric dispersion characterizes the time and space evolution of a set of particles (aerosols, gases, dust) emitted into the atmosphere. The atmospheric dispersion phenomenon is influenced by the atmospheric conditions, the terrain parameters and emission values. The mathematical simulation of pollutants' dispersion into the atmosphere is used for estimating the concentration of pollutants emitted from the factory activity or automobile traffic downwind (Niepsch et al., 2021).

4.1.1 Atmospheric Dispersion Models

This section gives the mathematical formulations and analytical solutions of the Gaussian model for the most general case that includes transport, diffusion, deposition, sedimentation, and first-order chemical transformation of pollutants. By considering two chemically-coupled gaseous or particulate pollutant species; the primary (species-1 or reactant) pollutant is assumed to transform into the secondary (species-2 or reaction product) pollutant at a known constant rate. For generality, the two species are assumed to have known non-equal deposition and settling velocities.

First, we derive analytical solutions for concentrations of the two pollutant species emitted from an elevated continuous point source. Then we express the dispersion coefficients in these solutions in terms of the widely used Gaussian plume dispersion parameters. The resulting expressions are parameterized, simplified, and presented as extensions of the Gaussian plume algorithms (AERMOD) currently used in EPA air quality models for various atmospheric stability and mixing conditions. Further simplifications of the new algorithms are indicated for gaseous or fine suspended particulate pollutants with negligible settling, and for ground-level sources and/or receptors. Limiting expressions-of the algorithms are derived for large particles, when gravitational settling is the dominant deposition mechanism. Finally, we utilize these new point source Concentration algorithms to derive expressions for the concentrations of the two species due to emissions from area sources, using an innovative approach based on mass balance considerations.

The inputs for the atmospheric dispersion models are.

- the meteorological conditions (wind speed and direction, atmospheric turbulence, air temperature)
- the emission parameters (source location and height, chimney diameter, mass flow rate and velocity, and exhaust temperature)
- The geographic data of the source and receiver.
- The location and sizes of obstructed objects (buildings or other structures).

4.1.1.1 Mathematical Formulations by Statistical Approach

Consider a number of particles released from a point. The distribution of the particles at time T, due to turbulent fluctuations denoted by:

$$\sigma^2(T) = \overline{X^2} \quad (4.1)$$

Where X is the distance from the center.

The rate change of $\overline{X^2}$ is given by

$$\frac{d\overline{X^2}}{dt} = 2\overline{X \frac{dX}{dt}} = 2\overline{Xu'} \quad (4.2)$$

Where u' is the turbulent fluctuation of wind acting on the particles which take up the fluid speed immediately. This rate of spread may be expressed in terms of the wind speed at successive time intervals

$$\frac{d\overline{X^2}}{dt} = 2 \int_0^t \overline{u'(t)u'(t+\xi)} d\xi \quad (4.3)$$

The concentration X is obtained assuming a triple Gaussian expression for a puff release and a double Gaussian expression for a plume.

$$X(x, y, z, t) = \frac{Q}{(2\pi)^{3/2} \sigma_x \sigma_y \sigma_z} \exp \left\{ -\frac{1}{2} \left(\frac{x^2}{\sigma_x^2} + \frac{y^2}{\sigma_y^2} + \frac{z^2}{\sigma_z^2} \right) \right\} \quad (4.4)$$

Where $\sigma_x, \sigma_y, \sigma_z$ are the standard deviations and x, y, z are measured with respect to a moving coordinate system with origin at the cloud center. The x -axis is in the down wind direction, y -axis in the horizontal cross wind direction and z -axis in the vertical. Q is the source strength.

By using plume model the above equation of concentration will be rewritten in the following form:

$$X(x, y, z) = \frac{Q}{2\pi u \sigma_x \sigma_y} \exp \left\{ -\frac{1}{2} \left(\frac{y^2}{\sigma_y^2} + \frac{z^2}{\sigma_z^2} \right) \right\} \quad (4.5)$$

The Gaussian model can be obtained as a solution of the diffusion for the situation when the wind velocity and the diffusion coefficient are independent of distance or from statistical theory. The model can be expressed in two ways, namely the Gaussian Puff model which gives the instantaneous concentration from a single release or the Gaussian Plume model which gives the time integral of concentration from a release or the standing concentration from a continuous release. Both models relate to an inert pollutant released from a point source under idealized conditions of horizontal uniformity and terrain. Practical situations however differ from these ideals, creating a need to modify the basic formulae for application to the situation considered.

The Gaussian models for a point source

The concentration distribution from a single release is given by the Gaussian Puff model as

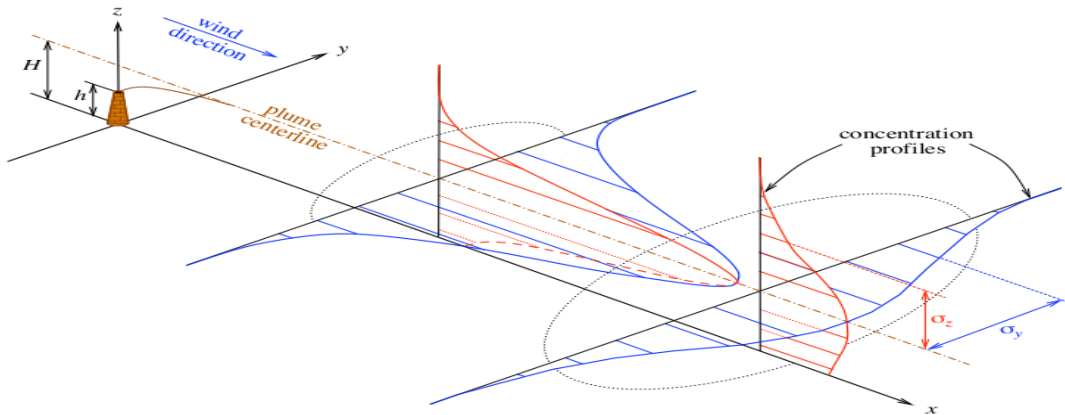


Figure 4. 1: A contaminant plume emitted from a point source (Laxmipriya & Narayanan, 2019).

$$X(x, y, z, t) = \frac{Q}{(2\pi)^{3/2} \sigma_x \sigma_y \sigma_z} \exp \left[-\frac{(x - \bar{u}t)^2}{2\sigma_x^2} \right] \cdot \exp \left[-\frac{y^2}{2\sigma_y^2} \right] \cdot \left\{ \exp \left[-\frac{(z - H)^2}{2\sigma_z^2} \right] + \exp \left[-\frac{(z + H)^2}{2\sigma_z^2} \right] \right\} \quad (4.6)$$

Where X is concentration of pollutant, x, y, z are rectilinear axis, t is travel time, Q is rate of emission, $\sigma_x, \sigma_y, \sigma_z$ are the standard deviations along x, y, z , $\bar{u}$ is the mean wind speed, H is effective release height, h is stack height, ΔH plume rise height. $H = h + \Delta H$.

The time integrated concentration is given by the Gaussian Plume model as

$$X(x, y, z) = \frac{Q}{2\pi u \sigma_y \sigma_z} \exp\left[-\frac{y^2}{2\sigma_y^2}\right] \cdot \left\{ \exp\left[-\frac{(z-H)^2}{2\sigma_z^2}\right] + \exp\left[-\frac{(z+H)^2}{2\sigma_z^2}\right] \right\} \quad (4.7)$$

In both cases the mean wind direction is taken to be the x -axis and the origin of the co-ordinate system is assumed to be at the release point or the base of the release point for releases from a stack. The second exponential term within the curly brackets is the term due to reflection at the ground. From the expression for concentration above, we get the ground level concentration (GLC) by setting $Z=0$ is given by:

$$X(x, y, 0) = \frac{Q}{\pi u \sigma_y \sigma_z} \exp\left[-\frac{1}{2}\left(\frac{y^2}{\sigma_y^2} + \frac{H^2}{\sigma_z^2}\right)\right] \quad (4.8)$$

Centre line GLC at $y=0, Z=0$ is given by:

$$X(x, 0, 0) = \frac{Q}{\pi u \sigma_y \sigma_z} \exp\left[-\frac{1}{2}\left(\frac{H^2}{\sigma_z^2}\right)\right] \quad (4.9)$$

The maximum GLC is given by:

$$X_{MAX} = \frac{2Q}{e\pi u H^2} \left(\frac{\sigma_z}{\sigma_y}\right) \quad (4.10)$$

Both standard deviation given by:

$$\begin{aligned} \sigma_y &= ax^{0.894} \\ \sigma_z &= cx^d + f \end{aligned} \quad (4.11)$$

Where x is distance from the stack, a, c, d and f is constant parameter of standard deviation.

Batu Caustic Soda Air Pollution Modeling

Air pollution modeling begin with identifying the stability character of the atmospheric air, Stability is determined using both the change of temperature with height (which can be termed the adiabatic lapse rate) and wind speed. Stability classes are determined for different meteorological conditions, which are dependent on wind speed and solar insolation during the day and cloud cover during the night. A method of categorizing the stability of a region of the atmosphere in terms of the horizontal surface wind, the amount of solar radiation and the fractional cloud cover was summarized on pasquill stability class table. It categorized the atmospheric turbulence into six stability classes named A, B, C, D, E and F with class A being the most unstable or most turbulent class, and class F the most stable or least turbulent class.

Table 4. 1: Pasquill Stability Classes (Awasthi et al., 2006).

Surface wind Speed (m/s)	Day time incoming Solar radiation			Day / night time Cloud cover	Night cloud cover	
	strong	moderate	Slight	8-10	Upper sky layer 5-10, middle / lower sky layer 5-7	0-4
<2	A	A-B	B	D	G	G
2-3	A-B	B	C	D	E	F
3-4	B	B-C	C	D	D	E
4-6	C	C-D	D	D	D	D
>6	C	D	D	D	D	D

The stability class the average wind speed of the given year at specified point and daytime incoming solar radiation strength. Metrological air quality data of surface wind speed (m/s) and day time incoming solar radiation of Batu Caustic Soda (Latitude 7.9381 and Longitude 38.7058) was summarized as follows. For the record metrological air quality data used in this research was collected from Hawasa national metrology agency service and Addis Ababa national metrology agency service center.

Table 4. 2: Wind speed annual average of Batu city (2010-2021)

Year	2010	2011	2012	2013	2014	2015	2016	2017	2018	2019	2020	2021
Speed (m/s)	7.45	3.2	4.12	6.07	7.52	3.11	2.94	5.41	6.40	2.79	5.37	6.88

Depending on the each year annual wind speed average the overall average wind speed of each 12 year was 5.108 m/s. depending on this result and strong solar radiation of Batu cities from Pasquill Stability Classes table Batu caustic soda air quality will categorized under class (C) with speed (4-6) m/s which mean slightly unstable air quality. For this the plume standard deviation coefficient factor was essential and it was given in the following table.

Table 4. 3: Plume standard deviation coefficient (Offermann et al., 1985).

Stability class	X < 1 Km				X > 1 Km		
	a	c	d	f	c	d	f
A	13	440.8	1.941	9.27	459.7	2.094	-9.6
B	156	100.6	1.149	3.3	108.2	1.098	2
C	104	61	0.911	0	61	0.911	0
D	68	33.2	0.725	-1.7	44.5	0.516	-13.0
E	50.5	22.8	0.678	-1.3	55.4	0.305	-34.0
F	34	14.35	0.74	-0.35	62.6	0.18	-48.6

Plume standard deviation is given by

$$\begin{aligned}\sigma_y &= ax^{0.894} \\ \sigma_z &= cx^d + f\end{aligned}\quad (4.12)$$

Where σ_y , σ_z is horizontal source dispersion deviation and vertical standard deviation and a, c, d and f are plume standard deviation factor constant from the table above and x was the distance considered for pollution dispersion.

As the pollution is coming from the three plant I mentioned before CaO₂ plant, NaOH plant and boiler plant, the plume distribution to the atmosphere begin within stuck height (h), there is also

plume height after stuck height, which denoted by (ΔH). The height at which the pollutant start to distribute to the atmospheric air was denoted by (H), total height is given by:

$$H = h + \Delta H \quad (4.13)$$

$$\Delta H = \frac{V_s d}{u} \left[1.5 + \left(0.0268 P \left(\frac{T_s - T_a}{T_s} \right) d \right) \right] \quad (4.14)$$

Where V_s is speed of a plume, d is stack diameter, u is wind speed at 10m height, P is pressure, T_s was stack temperature and T_a was an ambient temperature.

The concentration of pollutant where calculated by the following formula but in some boundary condition case we may use different formula to calculate the concentration of the pollutant element on specified location or point.

- I. General Gaussian plume dispersion model describe the pollutant distribution to all direction with:

$$X_{(x, y, z, H)} = \frac{E}{2\pi u \sigma_y \sigma_z} \exp \left[- \left(\frac{y^2}{2\sigma_y^2} \right) \right] \cdot \left\{ \exp \left[- \frac{(Z - H)^2}{2\sigma_z^2} \right] + \exp \left[- \frac{(Z + H)^2}{2\sigma_z^2} \right] \right\} \quad (4.15)$$

- II. For Ground level concentration pollutant distribution to atmosphere from stuck at height H and the horizontal dispersion was not considered, the concentration of the pollutant from the stuck was given by using Gaussian plume dispersion model as: ($Z=0$)

$$X_{(x, y, 0, H)} = \frac{E}{\pi u \sigma_y \sigma_z} \exp \left[- \frac{y^2}{2\sigma_y^2} \right] \cdot \exp \left[- \frac{H^2}{2\sigma_z^2} \right] \quad (4.16)$$

- III. For ground level at the center of pollution source, where ($y=0, Z=0$) the concentration was given by:

$$X_{(x, 0, 0, H)} = \frac{E}{\pi u \sigma_y \sigma_z} \exp \left[- \frac{H^2}{2\sigma_z^2} \right] \quad (4.17)$$

- IV. Ground level center line concentration when source of pollution is also at the ground level, ($y=0, Z=0, H=0$) the concentration was given by:

$$X_{(x, 0, 0, 0)} = \frac{E}{\pi u \sigma_y \sigma_z} \quad (4.18)$$

- V. The maximum ground level center line concentration elevated source ($y = 0, Z = 0$) is given by:

$$X_{(x, 0, 0, H)} = \frac{E}{\pi u e H^2} \cdot \frac{\sigma_z}{\sigma_y} \quad (4.19)$$

- VI. For infinite line source located at the ground level with some elevation H ($Z=0$) concentration was given by:

$$X_{(x, y, 0, H)} = \frac{2E_L}{(2\pi)^{1/2} u \sigma_z} \exp \left[-\frac{H^2}{2\sigma_z^2} \right] \quad (4.20)$$

- VII. For infinite line source located at ground level ($Z=0, H=0$) concentration was given by:

$$X_{(x, y, 0, 0)} = \frac{2E_L}{(2\pi)^{1/2} u \sigma_z} \quad (4.21)$$

- VIII. The maximum ground level concentration is given by:

$$X_{\max(x, 0, 0, 0)} = \frac{E}{\pi u \sigma_y \sigma_z} \quad (4.22)$$

Where E is pollutant emission rate and E_L was pollutant emission rate per unit length.

A plume travels down wind, it disperses horizontally and vertical, dispersion model are used to estimate the concentration of pollutant downwind as function of:

1. **Effective stack height:** the plume rise together with the physical stack height and it increase as the exit velocity increase and stack exit temperature increase.
2. **Mass rate of Emission pollutant:** there is direct relationship between Emission rate and concentration of the pollutant.
3. **Wind speed and direction:** increasing wind speed will decrease the plume rise but increase the dilution.
4. **Atmospheric stability:** the more unstable the atmosphere the greater the diluting factor, inversion about the stack height restrict vertical dilution.

The basic assumptions of this model.

- Emission rate is constant.
- Stability can be adequately represented by metrological data.
- Cross winds are minimal/ negligible.
- Terrain is flat near the source.

- The plume where reflected back to the atmosphere when it reach the ground.
- Plume from different source do not interact.
- The plume is buoyant.
- Statically normal distribution are followed.

Batu Caustic soda office building where found around center of the factory and there are four pollution source which affects the building office and selected from all other source due to their severity. They are calcium oxide (CaO_2) plant which found at 200m distance to North hand from the office building, Caustic Soda (NaOH) plant which found at 157m on the East hand, boiler plant at 230 m on the east side and Dust accumulation at 190 m away from the building office.

The Emission rate of Batu Caustic soda ambient atmospheric pollutants where collected from air quality data and presented as follows.

Table 4. 4: The following Emission rate data was for 10 year on each pollutant.

Year	Compounds(mg/m ³) = 1 x 10 ⁴ μ g/m ³					
	SO ₂	NO ₂	CO	O ₃	PM ₁₀	PM _{2.5}
2010	282.2	22.67	36648	43.07	149.8	4472
2011	281.8	22.81	36950	43.16	278.4	4509
2012	279.5	21.72	36975	43.3	331.7	4495
2013	277.9	21.24	36192	43.19	307.9	5121
2014	273.1	22.67	36162	43.26	313.6	5213
2015	281.5	21.25	35184	43.1	312.1	5220
2016	286.9	22.67	35211	43.19	317.1	5202
2017	281.5	22.13	36081	43.28	315.1	5219
2018	286.3	22.67	36743	43.33	353.7	5253
2019	284.4	21.96	36723	43.02	394	5290
2020	285.1	22.67	36489	43.21	392.2	5298
2021	289.2	22.75	36769	45.22	395.3	5327
Annual	282.5	22.3	36343.9	43.63	321.7	5030

It can be can be concluded from the above two table that the emission rate of Batu Caustic Soda air quality emission level was greater than the emission rate given by the Ethiopian Guideline standards for priority ambient atmospheric pollutants, from this result it's possible to conclude the environment of study was polluted.

4.2 Theoretical Modeling of Outdoor Air Pollution

To characterize the air quality by using Gaussian dispersion mathematical modeling which known by dispersion model the surface pressure of the wind, wind speed at surface and at 10 m height, temperature of the area, pollution data where required.

Table 4. 5: Outdoor air quality modeling parameter

Year	Parameter			
	Pressure (KPa)	Temperature (K)	Surface wind speed (m/s)	Wind speed at 10 m (m/s)
2010	81.61	298.79	4.831	7.447
2011	81.56	300	3.2	5.473
2012	81.59	299.64	4.1	5.495
2013	81.62	298.54	4.861	6.076
2014	81.64	299.15	5.118	7.522
2015	81.67	300.25	3.111	5.635
2016	81.69	299.12	2.944	5.054
2017	81.65	299.18	5.245	5.413
2018	81.64	299.09	5.222	6.407
2019	81.65	299.47	2.793	5.454
2020	81.65	298.41	5.087	5.375
2021	81.6	301.8	5.824	6.879
Annual	81.63	299.45	5.01	6.108

the stack height of CaO₂ lime stone production plant was 45m from the ground, the stack of NaOH plant was 27 m and that of boiler was 32m from the ground, the rest source of pollution was at ground level. In case of this research study the source where categorized in to four man poin source and all concentration and other parameter where analysed and modeld at those four points from office building of the factory. Their distance from office building was The following figure shows Batu Causic soda factory and the point where considered as a source for the pollution.

4.2.1 Calcium oxide (CaO₂) Plant Emission

Calcium oxide plant in Batu Caustic soda used for production of lime which called by CaO₂ it used for sugar factory to make sugar white, in poultry feed production, in detergent production and for Nora production. Height of stack 45m and 200m away from office building, plume temperature was 1100 °C, the plume speed was 16 m/s, air stability class was C, the concentration of pollutant from 2010-2021 was:

SO₂ Emission at January 2010

$$\sigma_y = ax^{0.894}$$

$$\sigma_z = cx^d + f$$

Table 4. 6: Calcium oxide (CaO₂) plant theoretical modeling analysis parameter.

X = 1 Km	a=104	c = 61	d= 0.911	f = 0	P = 81.61 KPa	h = 45m
E = 2.822 g/s	Vs = 16 m/s	d = 0.5 m	U = 4.831 m/s	T _s = 1373 K	Ta = 298.79 K	y = 0.2 Km = 200 m

$$\sigma_y = 104(1)^{0.894} = 104 \text{ m}$$

$$\sigma_z = 61(1)^{0.911} + 0 = 61 \text{ m}$$

$$\begin{aligned} \Delta H &= \frac{Vsd}{u} \left[1.5 + \left(0.0268P \left(\frac{T_s - T_a}{T_s} \right) d \right) \right] \\ &= \frac{16 * 0.5}{7.447} \left[1.5 + \left(0.0268 * 81.61 \left(\frac{1373 - 298.79}{1373} \right) * 0.5 \right) \right] \\ &= 1.074 \left[1.5 + (2.187(0.146) * 0.5) \right] \\ &= 1.782 \text{ m} \end{aligned}$$

$$\begin{aligned} H &= h + \Delta H \\ &= 45 + 1.782 \text{ m} \\ &= 46.782 \text{ m} \end{aligned}$$

$$X_{(x, y, 0, H)} = \frac{E}{\pi u \sigma_y \sigma_z} \exp\left[-\frac{y^2}{2\sigma_y^2}\right] \cdot \exp\left[-\frac{H^2}{2\sigma_z^2}\right]$$

$$X_{(x, y, 0, H)} = \frac{2.822}{3.14 * 4.831 * 104 * 61} \exp\left[-\frac{(0.2)^2}{2(104)^2}\right] \cdot \exp\left[-\frac{(46.782)^2}{2(61)^2}\right]$$

$$X_{(x, y, 0, H)} = 2.9324 \times 10^{-5} \exp[-1.85 \times 10^{-6}] \cdot \exp[-0.2941]$$

$$= 2.9324 \times 10^{-5} * 0.99999 * 0.7452$$

$$= 2.18522 \times 10^{-5} \text{ g / m}^3$$

$$= 21.8522 \mu\text{g / m}^3$$

The concentration of SO₂ on the Building office, in this case office building was summarized as follows.

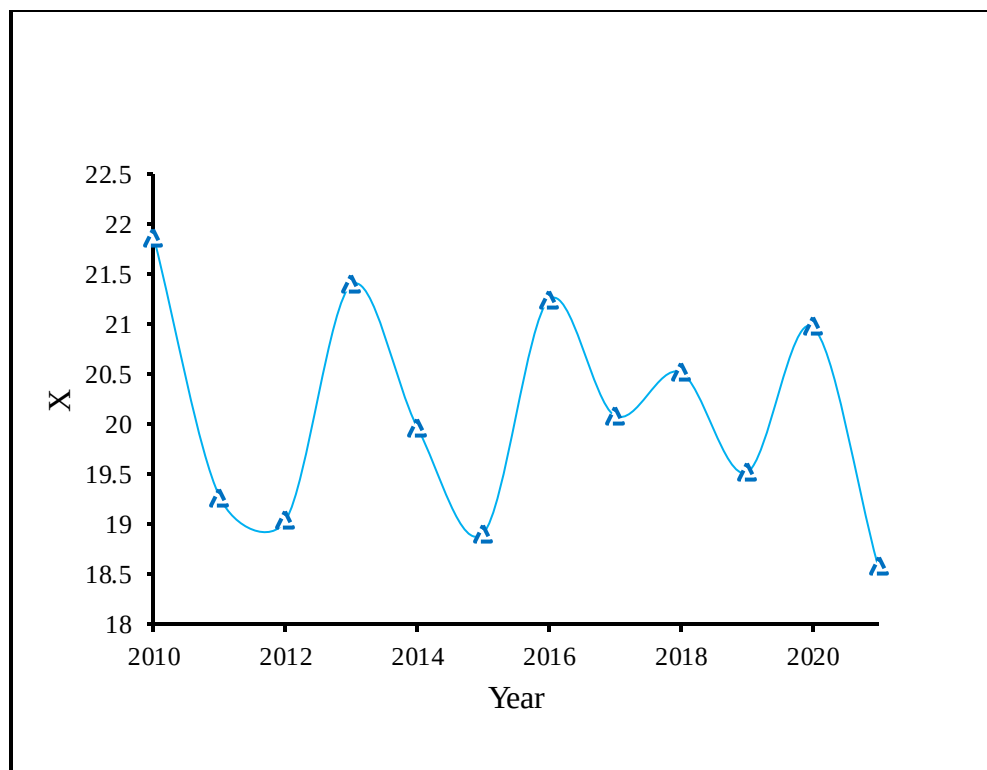


Figure 4. 2: Calcium oxide (CaO₂) plant annual sulfur dioxide Emission concentration,
For Numerical Result Data for graph (1) Reference (Appendix A1).

NO₂ Emission at January 2010

$$\sigma_y = ax^{0.894}$$

$$\sigma_z = cx^d + f$$

$$\sigma_y = 104(1)^{0.894} = 104 \text{ m}$$

$$\sigma_z = 61(1)^{0.911} + 0 = 61 \text{ m}$$

$$\begin{aligned}\Delta H &= \frac{Vsd}{u} \left[1.5 + \left(0.0268P \left(\frac{T_s - T_a}{T_s} \right) d \right) \right] \\ &= \frac{16 * 0.5}{7.447} \left[1.5 + \left(0.0268 * 81.61 \left(\frac{1373 - 298.79}{1373} \right) * 0.5 \right) \right] \\ &= 1.074 \left[1.5 + (2.187(0.146) * 0.5) \right] \\ &= 3.90079 \text{ m}\end{aligned}$$

$$H = \Delta H + h$$

$$h = 45 \text{ m}$$

$$H = \Delta H + h$$

$$H = 3.90079 \text{ m} + 45 \text{ m}$$

$$H = 48.90079 \text{ m}$$

When h is height of the stack and change in H was height of the plume on atmosphere, H was over all maximum height at which plume stop raising and start distributing in atmosphere.

$$\begin{aligned}X_{(x, y, 0, H)} &= \frac{E}{\pi u \sigma_y \sigma_z} \exp \left[-\frac{y^2}{2\sigma_y^2} \right] \cdot \exp \left[-\frac{H^2}{2\sigma_z^2} \right] \\ X_{(x, y, 0, H)} &= \frac{366.48}{3.14 * 4.831 * 104 * 61} \exp \left[-\frac{(0.2)^2}{2(104)^2} \right] \cdot \exp \left[-\frac{(46.782)^2}{2(61)^2} \right] \\ X_{(x, y, 0, H)} &= 2.9324 \times 10^{-5} \exp \left[-1.85 \times 10^{-6} \right] \cdot \exp \left[-0.2941 \right] \\ &= 0.003808205 * 0.99999 * 0.7452 \\ &= 0.002761663 \text{ g} / \text{m}^3 \\ &= 2761.663 \mu\text{g} / \text{m}^3\end{aligned}$$

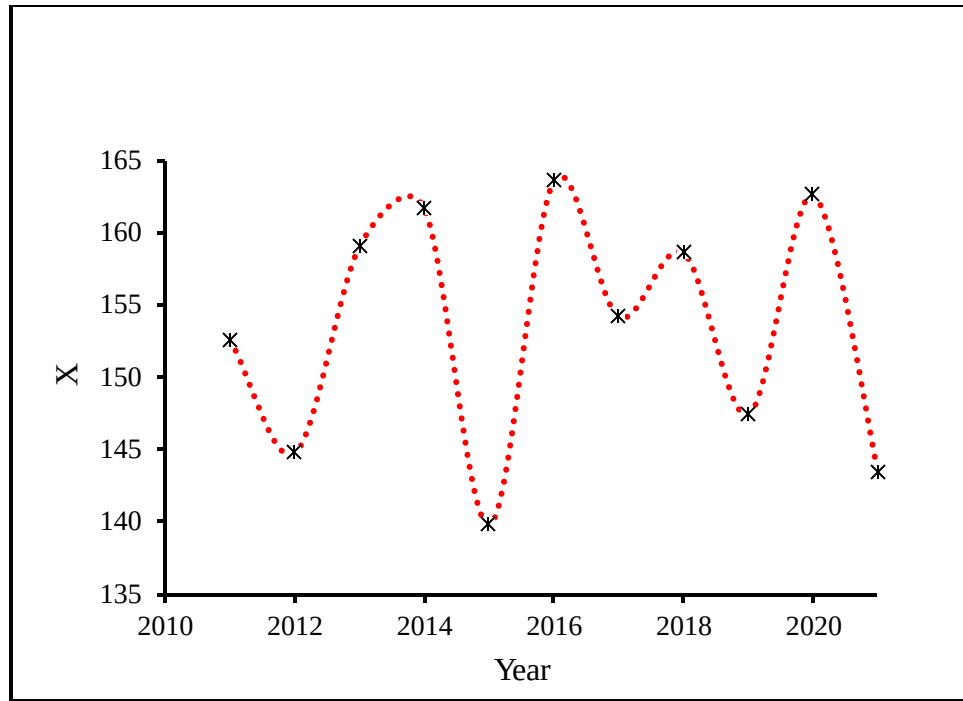


Figure 4. 3: Calcium oxide (CaO₂) plant annual Nitrogen dioxide Emission concentration,
For Numerical Result Data for graph (2) Reference (Appendix A1).

CO Emission at January 2010

$$\sigma_y = ax^{0.894}$$

$$\sigma_z = cx^d + f$$

$$\sigma_y = 104(1)^{0.894} = 104 \text{ m}$$

$$\sigma_z = 61(1)^{0.911} + 0 = 61 \text{ m}$$

$$\begin{aligned} \Delta H &= \frac{V_s d}{u} \left[1.5 + \left(0.0268 P \left(\frac{T_s - T_a}{T_s} \right) d \right) \right] \\ &= \frac{16 * 0.5}{7.447} \left[1.5 + \left(0.0268 * 81.61 \left(\frac{1373 - 298.79}{1373} \right) * 0.5 \right) \right] \\ &= 1.074 \left[1.5 + (2.187(0.146) * 0.5) \right] \\ &= 3.90079 \text{ m} \end{aligned}$$

$$H = \Delta H + h$$

$$H = 3.90079m + 45m$$

$$H = 48.90079m$$

$$X_{(x,y,0,H)} = \frac{E}{\pi u \sigma_y \sigma_z} \exp\left[-\frac{y^2}{2\sigma_y^2}\right] \cdot \exp\left[-\frac{H^2}{2\sigma_z^2}\right]$$

$$X_{(x,y,0,H)} = \frac{366.48}{3.14 * 4.831 * 104 * 61} \exp\left[-\frac{(0.2)^2}{2(104)^2}\right] \cdot \exp\left[-\frac{(46.782)^2}{2(61)^2}\right]$$

$$X_{(x,y,0,H)} = 2.9324 \times 10^{-5} \exp[-1.85 \times 10^{-6}] \cdot \exp[-0.2941]$$

$$= 0.003808205 * 0.99999 * 0.7452$$

$$= 0.002761663g / m^3$$

$$= 2761.663\mu g / m^3$$

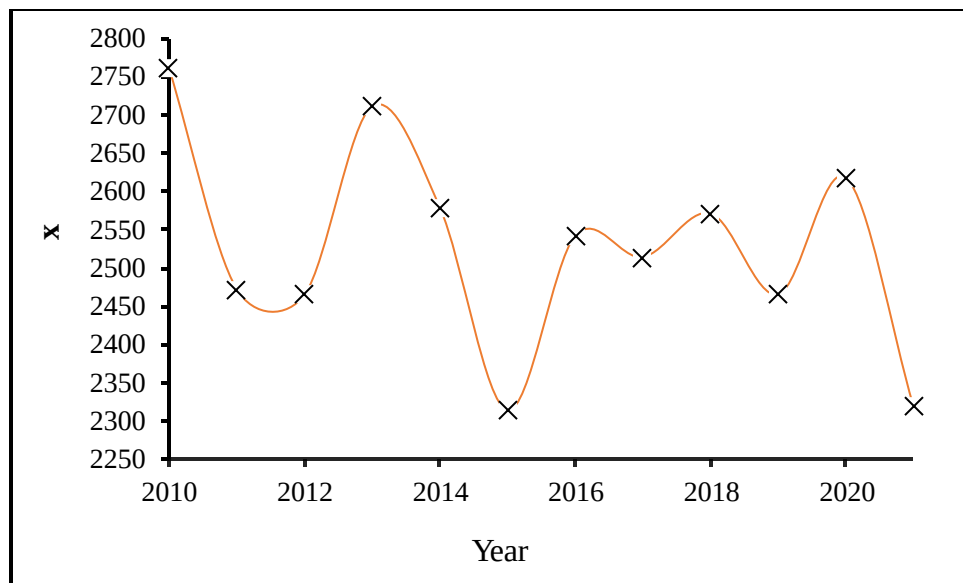


Figure 4. 4: Calcium oxide (CaO₂) plant annual Carbon monoxide Emission concentration,
For Numerical Result Data for graph (3) Reference (Appendix A1).

Ozone Emission at January 2010

$$\sigma_y = ax^{0.894}$$

$$\sigma_z = cx^d + f$$

$$\sigma_y = 104(1)^{0.894} = 104 \text{ m}$$

$$\sigma_z = 61(1)^{0.911} + 0 = 61 \text{ m}$$

$$\begin{aligned}\Delta H &= \frac{V_s d}{u} \left[1.5 + \left(0.0268 P \left(\frac{T_s - T_a}{T_s} \right) d \right) \right] \\ &= \frac{16 * 0.5}{7.447} \left[1.5 + \left(0.0268 * 81.61 \left(\frac{1373 - 298.79}{1373} \right) * 0.5 \right) \right] \\ &= 1.074 \left[1.5 + (2.187(0.146) * 0.5) \right] \\ &= 3.90079 \text{ m}\end{aligned}$$

$$H = \Delta H + h$$

$$H = 3.90079 \text{ m} + 45 \text{ m}$$

$$H = 48.90079 \text{ m}$$

$$\begin{aligned}X_{(x, y, 0, H)} &= \frac{E}{\pi u \sigma_y \sigma_z} \exp \left[-\frac{y^2}{2\sigma_y^2} \right] \cdot \exp \left[-\frac{H^2}{2\sigma_z^2} \right] \\ X_{(x, y, 0, H)} &= \frac{43.07}{3.14 * 4.831 * 104 * 61} \exp \left[-\frac{(0.2)^2}{2(104)^2} \right] \cdot \exp \left[-\frac{(46.782)^2}{2(61)^2} \right] \\ X_{(x, y, 0, H)} &= 2.9324 \times 10^{-5} \exp \left[-1.85 \times 10^{-6} \right] \cdot \exp \left[-0.2941 \right] \\ &= 0.000447554 * 0.99999 * 0.7452 \\ &= 0.00032456 \text{ g} / \text{m}^3 \\ &= 324.5602 \mu\text{g} / \text{m}^3\end{aligned}$$

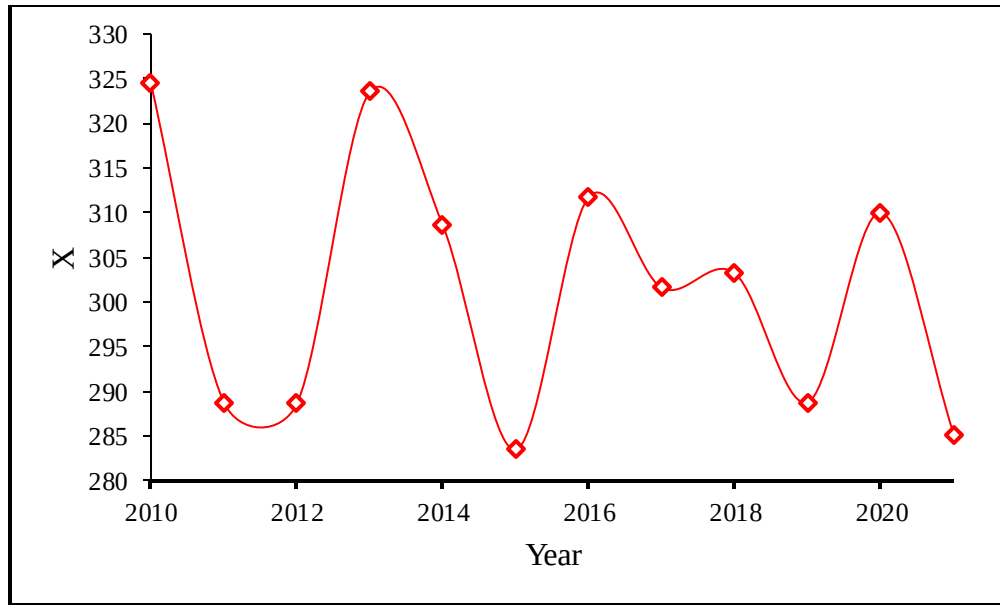


Figure 4. 5: Calcium oxide (CaO₂) plant Annual Ozone Emission concentration,
For Numerical Result Data for graph (4) Reference (Appendix A1).

PM₁₀ Emission at January 2010

$$\sigma_y = ax^{0.894}$$

$$\sigma_z = cx^d + f$$

$$\sigma_y = 104(1)^{0.894} = 104 \text{ m}$$

$$\sigma_z = 61(1)^{0.911} + 0 = 61 \text{ m}$$

$$\begin{aligned} \Delta H &= \frac{V_s d}{u} \left[1.5 + \left(0.0268 P \left(\frac{T_s - T_a}{T_s} \right) d \right) \right] \\ &= \frac{16 * 0.5}{7.447} \left[1.5 + \left(0.0268 * 81.61 \left(\frac{1373 - 298.79}{1373} \right) * 0.5 \right) \right] \\ &= 1.074 \left[1.5 + (2.187(0.146) * 0.5) \right] \\ &= 3.90079 \text{ m} \end{aligned}$$

$$H = \Delta H + h$$

$$H = 3.90079m + 45m$$

$$H = 48.90079m$$

$$X_{(x, y, 0, H)} = \frac{E}{\pi u \sigma_y \sigma_z} \exp\left[-\frac{y^2}{2\sigma_y^2}\right] \cdot \exp\left[-\frac{H^2}{2\sigma_z^2}\right]$$

$$X_{(x, y, 0, H)} = \frac{1.498}{3.14 * 4.831 * 104 * 61} \exp\left[-\frac{(0.2)^2}{2(104)^2}\right] \cdot \exp\left[-\frac{(46.782)^2}{2(61)^2}\right]$$

$$X_{(x, y, 0, H)} = 1.55662 \times 10^{-5} \exp[-1.85 \times 10^{-6}] \cdot \exp[-0.2941]$$

$$= 1.55662 \times 10^{-5} * 0.99999 * 0.7452$$

$$= 1.12884 \times 10^{-5} g / m^3$$

$$= 11.12884 \mu g / m^3$$



Figure 4. 6: Calcium oxide (CaO₂) plant Annual Particulate Matter (PM₁₀) Emission concentration, For Numerical Result Data for graph (5) Reference (Appendix A1).

PM_{2.5} Emission at January 2010

$$\sigma_y = ax^{0.894}$$

$$\sigma_z = cx^d + f$$

$$\sigma_y = 104(1)^{0.894} = 104 \text{ m}$$

$$\sigma_z = 61(1)^{0.911} + 0 = 61 \text{ m}$$

$$\begin{aligned}\Delta H &= \frac{V_s d}{u} \left[1.5 + \left(0.0268 P \left(\frac{T_s - T_a}{T_s} \right) d \right) \right] \\ &= \frac{16 * 0.5}{7.447} \left[1.5 + \left(0.0268 * 81.61 \left(\frac{1373 - 298.79}{1373} \right) * 0.5 \right) \right] \\ &= 1.074 \left[1.5 + (2.187(0.146) * 0.5) \right] \\ &= 3.90079 \text{ m}\end{aligned}$$

$$H = \Delta H + h$$

$$H = 3.90079 \text{ m} + 45 \text{ m}$$

$$H = 48.90079 \text{ m}$$

$$\begin{aligned}X_{(x, y, 0, H)} &= \frac{E}{\pi u \sigma_y \sigma_z} \exp \left[-\frac{y^2}{2\sigma_y^2} \right] \cdot \exp \left[-\frac{H^2}{2\sigma_z^2} \right] \\ X_{(x, y, 0, H)} &= \frac{44.72}{3.14 * 4.831 * 104 * 61} \exp \left[-\frac{(0.2)^2}{2(104)^2} \right] \cdot \exp \left[-\frac{(46.782)^2}{2(61)^2} \right] \\ X_{(x, y, 0, H)} &= 1.55662 \times 10^{-5} \exp \left[-1.85 \times 10^{-6} \right] \cdot \exp \left[-0.2941 \right] \\ &= 0.000464699 * 0.99999 * 0.7452 \\ &= 0.000336994 \text{ g} / \text{m}^3 \\ &= 336.994035 \mu\text{g} / \text{m}^3\end{aligned}$$

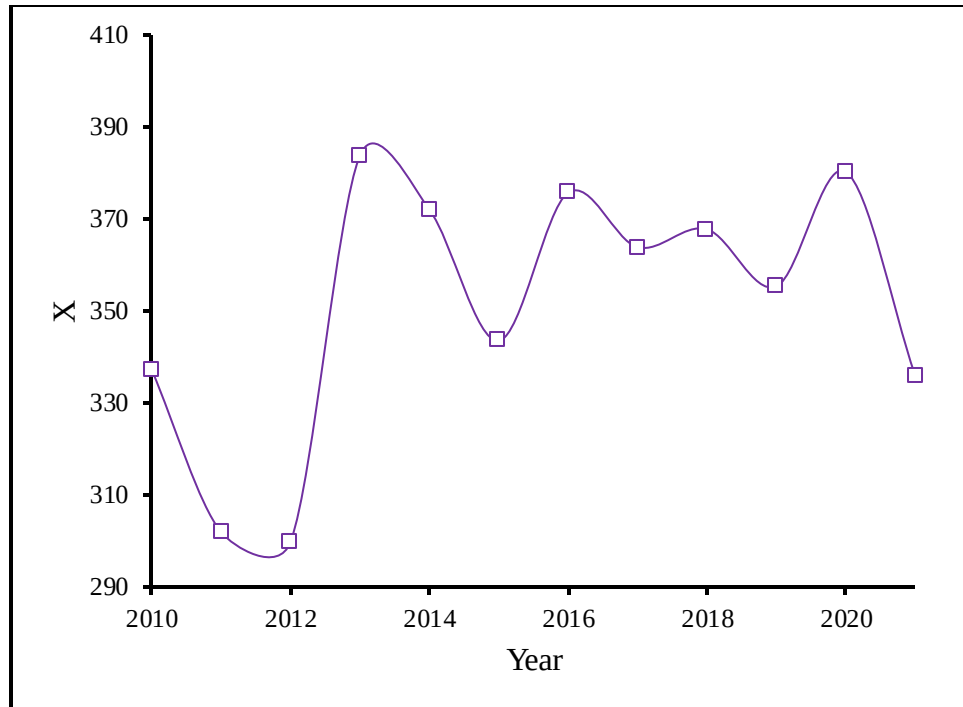


Figure 4. 7: Calcium oxide (CaO₂) plant Annual Particulate Matter (PM_{2.5}) Emission concentration, For Numerical Result Data for graph (6) Reference (Appendix A1).

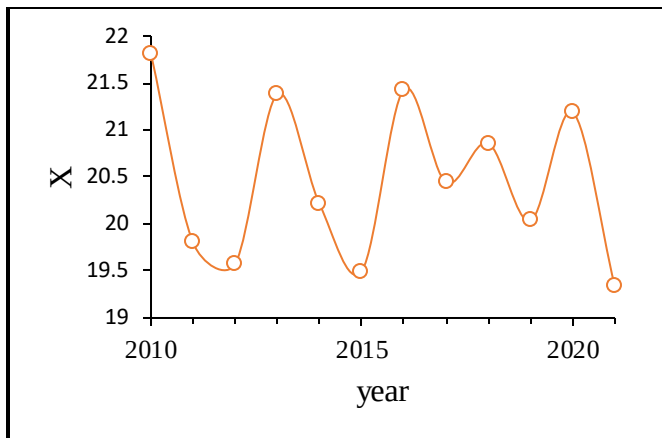
4.2.2 Caustic Soda (NaOH) Plant Emission

Caustic Soda (NaOH) plant in Batu Caustic soda and cement factory used for production of sodium Hydroxide for glass production, for plastic production and for a lot of application. Height of stack 27m and 157m away from office building, plume temperature was 500 °C, the plume speed was 17 m/s, air stability class was C, the concentration of pollutant from 2010-2021 was:

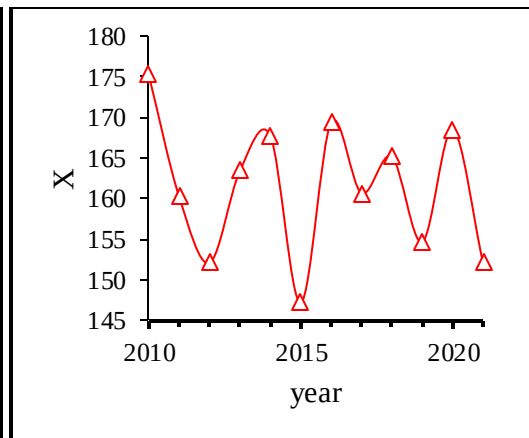
SO₂ Emission at January 2010

Table 4. 7: Caustic Soda (NaOH) Plant theoretical modeling analysis parameter.

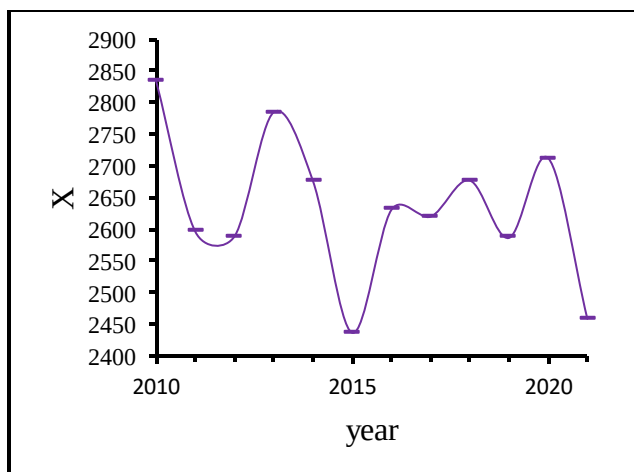
X = 1 Km	a=104	c = 61	d= 0.911	f = 0	P = 81.61 KPa	h = 27m
E = 2.822 g/s	Vs = 17 m/s	d = 1.5 m	U = 4.831 m/s	T _s = 773 K	Ta = 298.79 K	y = 157 m



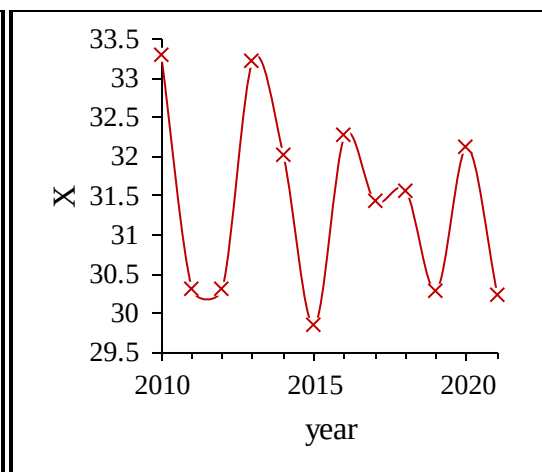
(a) - SO₂



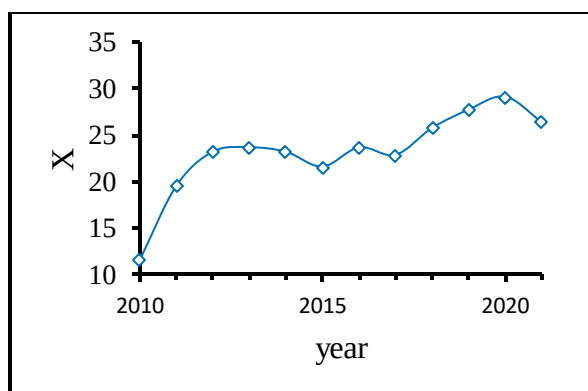
(b) - NO₂



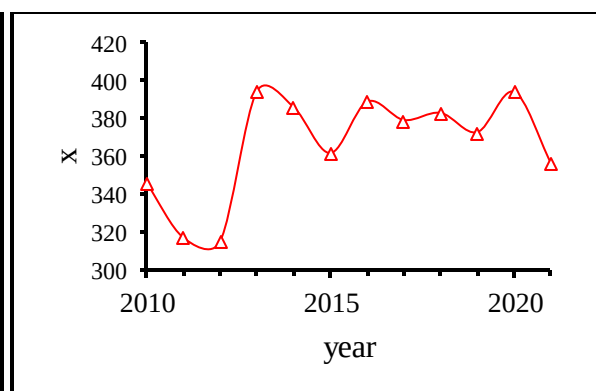
(c) - CO



(d) - O₃



(e) - PM₁₀



(f) - PM_{2.5}

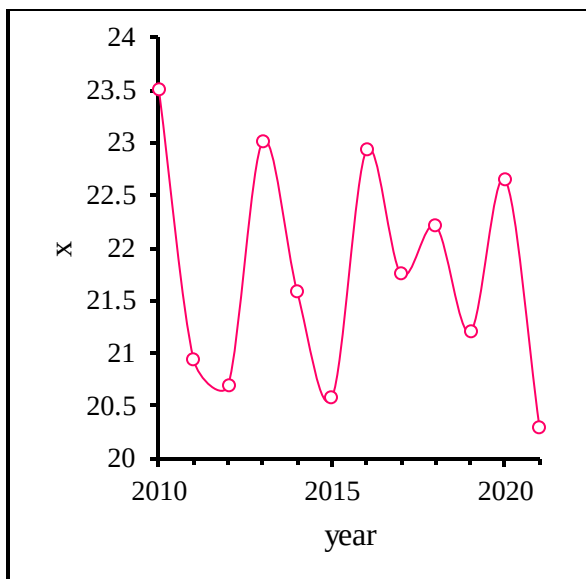
Figure 4. 8: Caustic Soda Plant Emission concentration, for Numerical Result Data for graph (7) Reference (Appendix A2).

4.2.3 Boiler Plant Emission

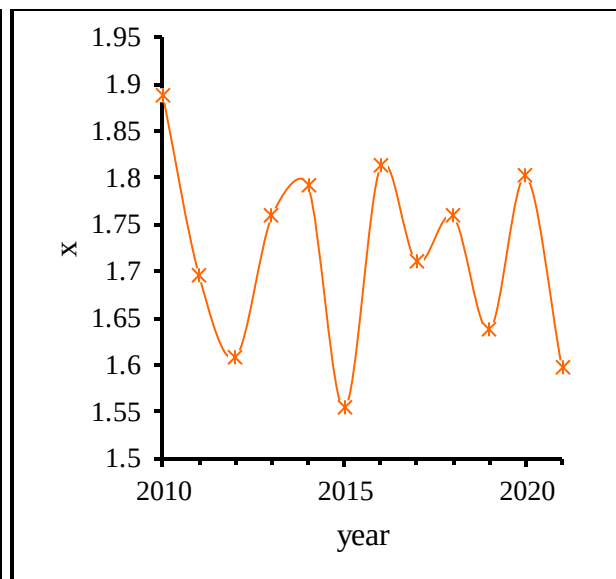
Boiler plant stack height was 32 m and 250m away from office building, plume temperature was 710 K, the plume speed was 15 m/s, air stability class was C, the concentration of pollutant from 2010-2021 was:

Table 4. 8: Boiler Plant theoretical modeling analysis parameter.

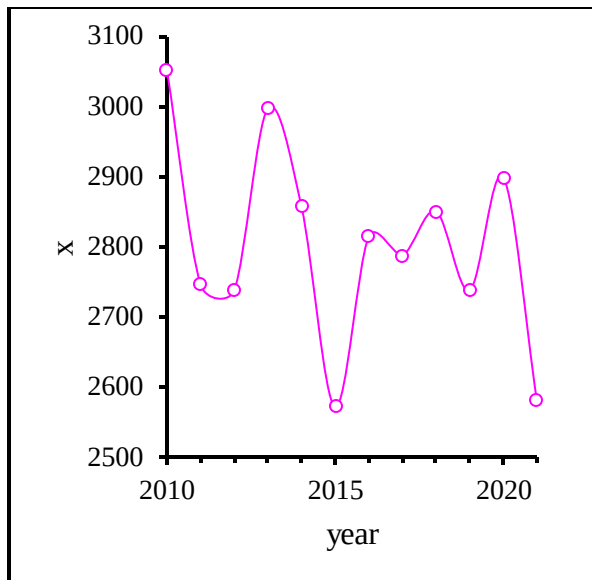
X = 1 Km	a=104	c = 61	d= 0.911	f = 0	P = 81.61 KPa	h = 32m
E = 2.822 g/s	Vs = 15 m/s	d = 1 m	U = 4.831 m/s	T _s = 710 K	Ta = 298.79 K	y = 250 m



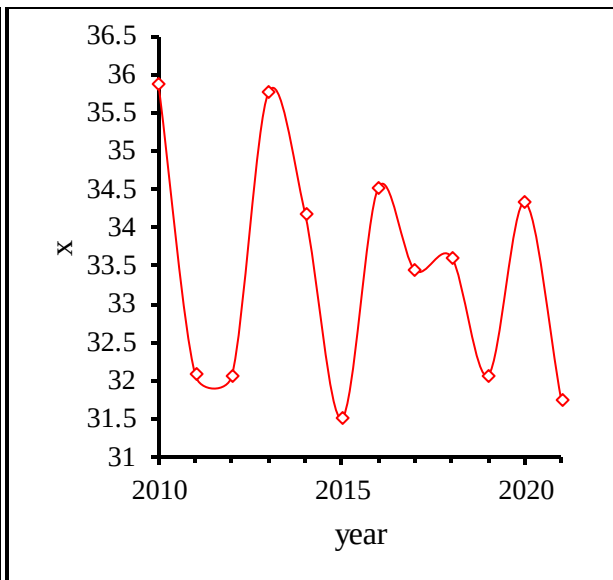
(a) - SO₂



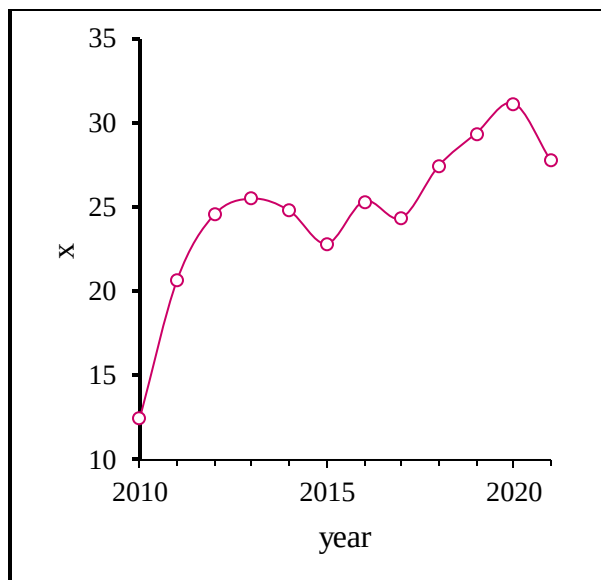
(b) - NO₂



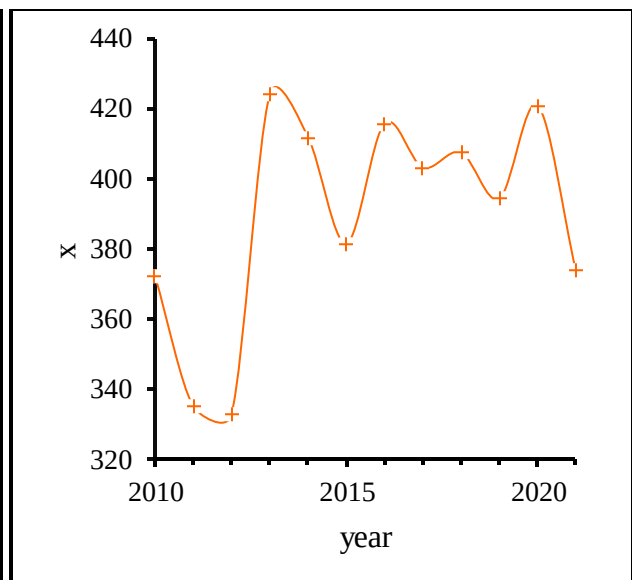
(C) - CO



(d) - O₃



(e) - PM₁₀



(f) - PM_{2.5}

Figure 4. 9: Boiler Plant Emission concentration, for Numerical Result Data for graph (8) Reference (Appendix A3).

4.2.4 Ground Level Emission Concentration

The sever Ground Level pollutant was the deposition of different material found on the north side of the factory and 300m away from office building, air stability class was C, the modeling of the concentration of pollutant from 2010-2021 should consider the following assumption:

- The source of pollution was at the ground level, ($y=0, Z=0, H=0$).
- The source of pollution was infinite source at the ground level with no elevation.

For infinite source located at ground level ($Z=0, H=0$) concentration was given by:

$$X_{(x, y, 0, 0)} = \frac{2E_L}{(2\pi)^{1/2} u \sigma_z} \quad (4.23)$$

The maximum ground level concentration is given by:

$$X_{\max(x, 0, 0, 0)} = \frac{E}{\pi u \sigma_y \sigma_z} \quad (4.24)$$

Infinite source concentration located at ground level ($Z=0, H=0$)

$X = 1 \text{ Km}$ $a = 104$ $c = 61$ $d = 0.911$ $f = 0$ $U = 4.831 \text{ m/s}$

$y = 300 \text{ m} = 0.35 \text{ Km}$

SO₂ Emission at January 2010

The maximum SO₂ Emission from the ground level was:

$$\begin{aligned} X_{\max(x, 0, 0, 0)} &= \frac{E}{\pi u \sigma_y \sigma_z} \\ &= \frac{2.822}{3.14 * 7.447 * 104 * 61} \\ &= 1.902317 \times 10^{-5} \text{ g / m}^3 \\ &= 19.02317 \mu\text{g / m}^3 \end{aligned}$$

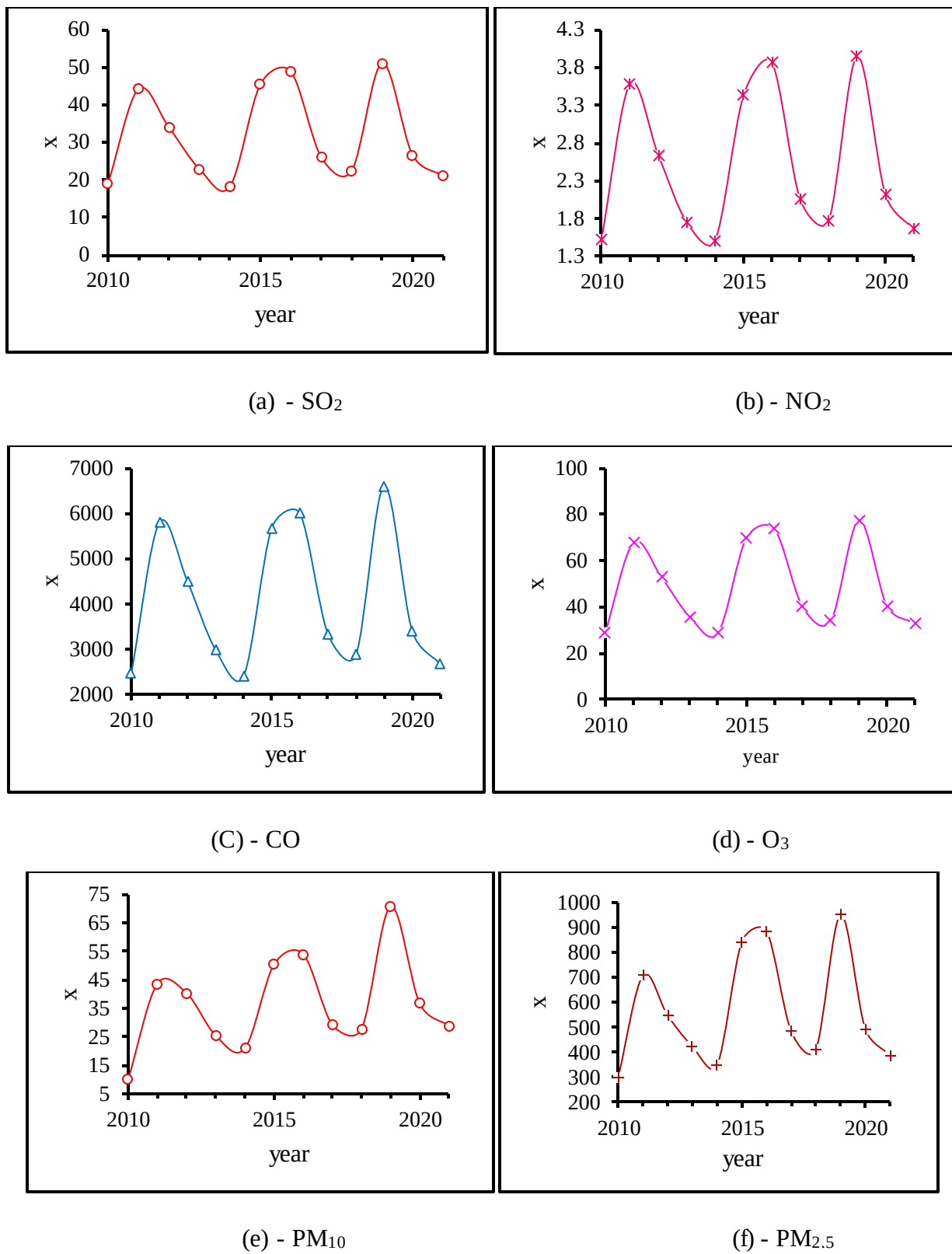


Figure 4. 10: Ground level Annual Emission rate for Numerical Result Data for graph (9)
Reference (Appendix A4).

From above analysis it's concluded that in Calcium carbonate production plant emission rate of pollutant such as NO_2 , O_3 and $\text{PM}_{2.5}$ where above the international limit as well as Ethiopian air quality standard limit given per year.

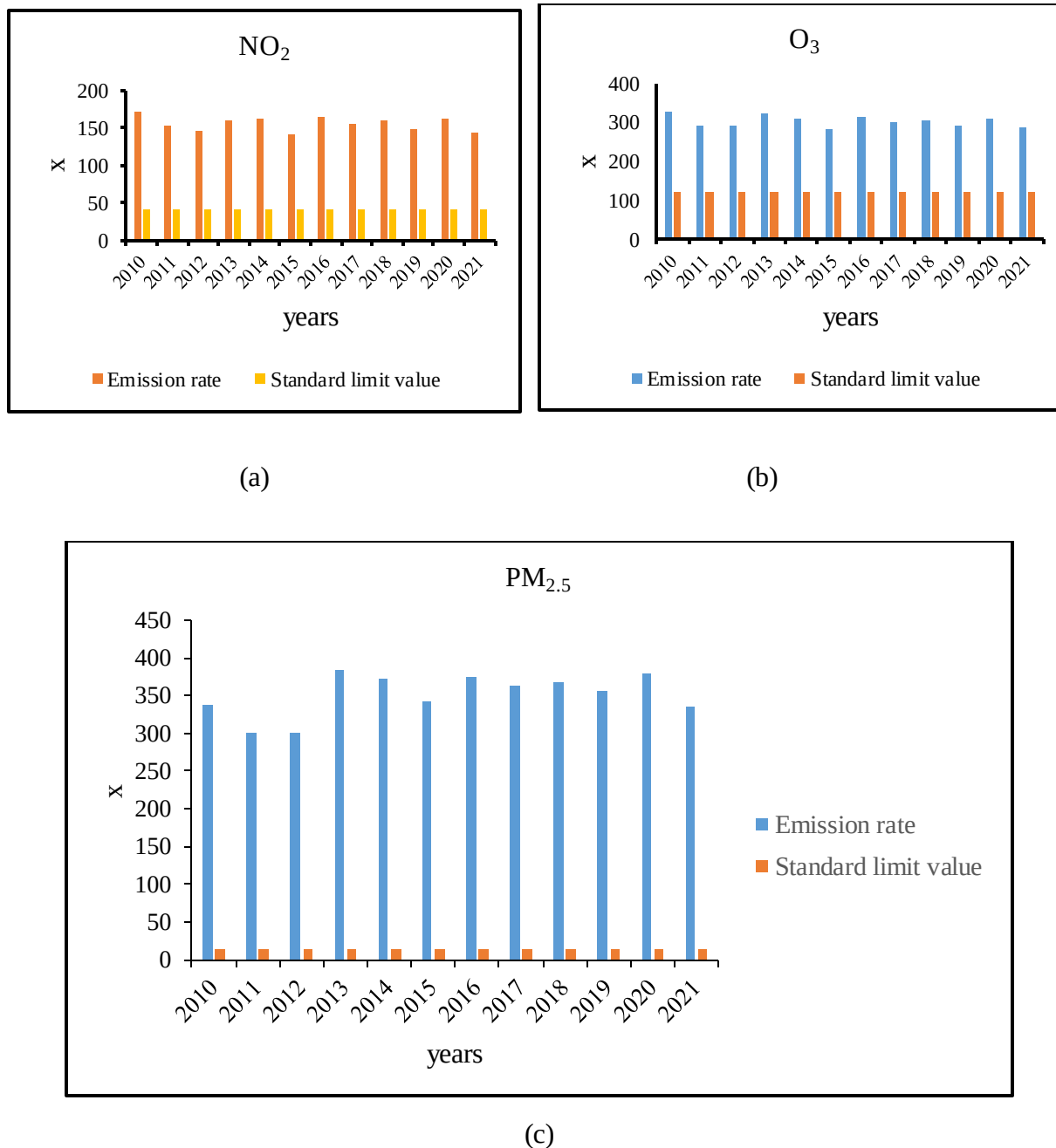


Figure 4. 11: CaCO_3 plant Emission compound Versis Ethiopia air quality standard.

In NaOH plant NO_2 and $\text{PM}_{2.5}$ was extent the limit of both ethiopian and international air quality standard.

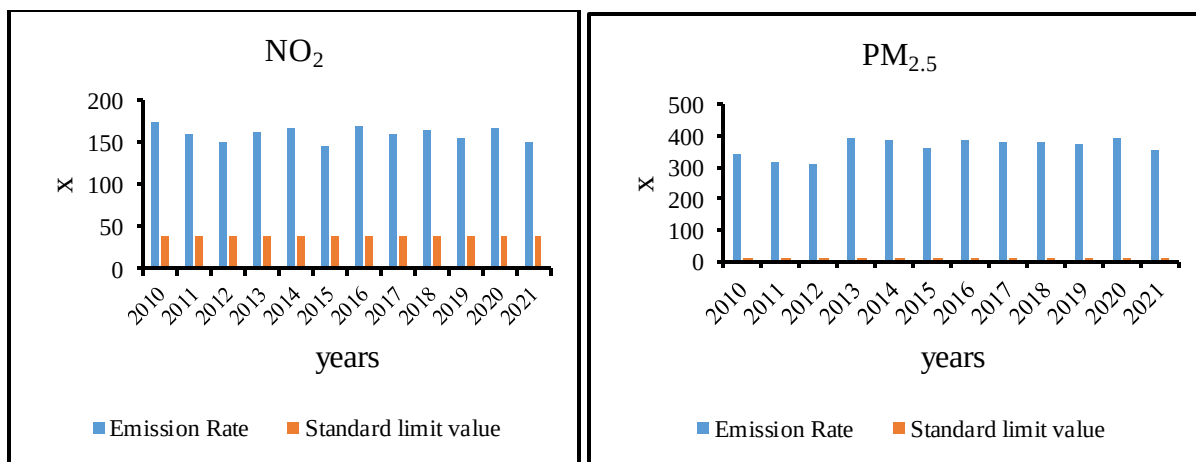


Figure 4. 12: NaOH plant Emission compound Versis Ethiopia air quality standard.

In Boiler plant $\text{PM}_{2.5}$ was extent the limit of both ethiopian and international air quality standard.

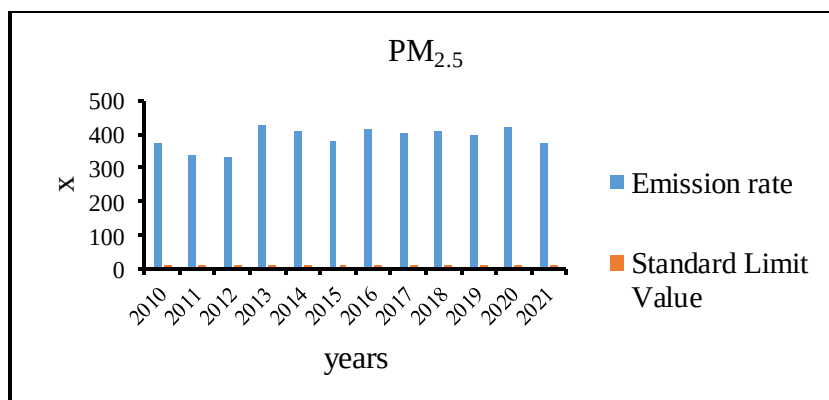


Figure 4. 13: Boiler plant Emission compound Versis Ethiopia air quality standard.

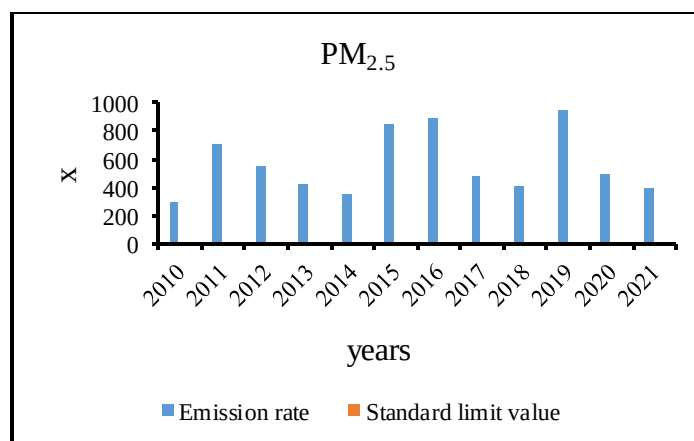


Figure 4. 14: Ground Level Emission compound Versis Ethiopia air quality standard.

4.3 Indoor Air Quality Modeling

Indoor air quality modeling carried out for estimation of indoor air polluting compound concentration, as referred in literature indoor air was 2-5 times more polluted than outdoor air. There is a lot of source of indoor air pollution emission from them outdoor air pollution infiltrated by natural circulation where play a great role in indoor air pollution the other was indoor activity which will cause air pollution. In case Batu Caustic Soda and Cement factory the outdoor air where completely polluted by four point source and even more as tried to be modeled in the outdoor air quality modeling above, in addition to this in Batu Caustic Soda and Cement factory office building different activity will go on, which cause emission and different appliance in the building and office will cause indoor air pollution emission.

There are a lot of methods used for modeling indoor air quality level and emission, in case of this research study Box modeling was used with some boundary condition case. For this modeling the building considered as boxes and the outdoor pollution concentration C_o , outdoor surface wind speed U , emission in the box Q_A , concentration of emission in the box C , length, width and height of the box $\Delta X, \Delta Y, Z$. The box will divided in to different small boxes because the building contain a lot of office inside it, after emission of each office calculated it summed up and the overall concentration of the indoor air pollution will be determined. In case of this modeling the air pollution dispersion was considered as steady state.

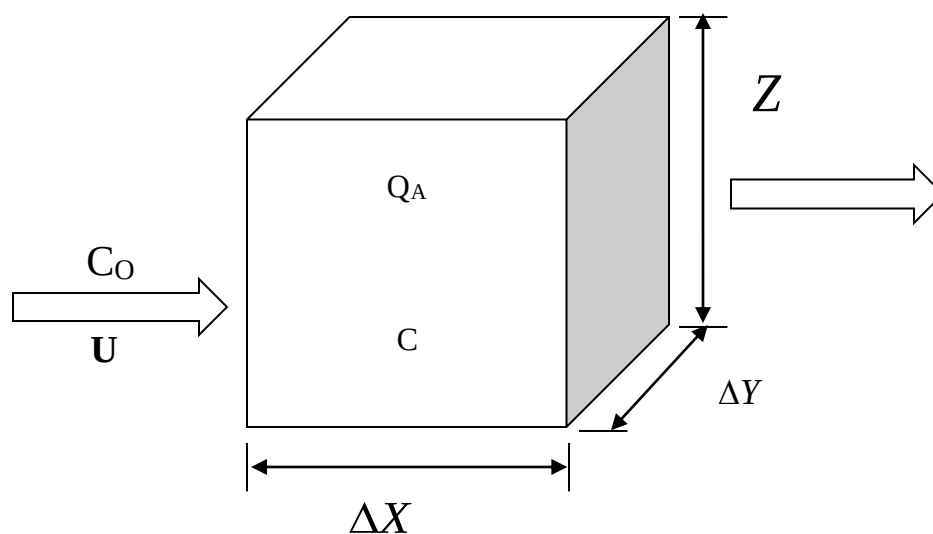


Figure 4. 15: General Box modeling out line

The general Governing equation of the box model was given by:

Accumulation(mass / time) = input to the box(mass / time) – output of the box(mass / time)

$$Q_A = \frac{\text{mass of emission in a box}}{\text{Area in unit time}} \quad (4.25)$$

$$\frac{\partial C}{\partial t} \Delta X . \Delta Y . Z = C_o . \Delta Y . Z . U + Q_A . \Delta X . \Delta Y - C . \Delta Y . Z . U \quad (4.26)$$

When we consider steady state air flow the above equation will become:

$$\frac{\partial C}{\partial t} = 0 \quad \text{and} \quad \Delta Y = 0$$

$$C = C_o + \frac{Q_A . \Delta X}{UZ} \quad (4.27)$$

As there is decay through time in indoor.

$$C = C_o + \frac{Q_A . \Delta X}{UZ} - KC \Delta X . \Delta Y . Z \quad (4.28)$$

From Batu caustic soda indoor air quality data store from (2017-2021) the indoor emission rate of indoor air pollutant was:

Table 4. 9: Batu Caustic Soda and Cement Factory Indoor pollutant emission rate

Compound	Annual averaged Concentration
CO ₂	4070000 mg/m ³
TVOC	1.938 mg/m ³
PM _{2.5}	0.972 mg/m ³
PM ₁₀	0.542 mg/m ³
Formaldehyde (HCHO)	0.517 mg/m ³

Before calculating any indoor pollutant concentration the emission rate of those pollutant from office to office where considered the same. Some indoor air pollution parameter where ignored during this analysis because they are too small and below than standard value for this they eradicated during this calculation.

Batu Caustic Soda and Cement Factory office building where ground plus one building (G+1) on the ground level it contain 17 office, 2 male and female toilet and 1 clinic room. On the top or first floor there are 16 office. Height of the building was 10m from the ground, length 95m and 12m width. Each office in the building have the same height, width and length except clinic, first floor finance office and toilet room.

Table 4. 10: Emission rate of each indoor portion of the building.

Floor	Name	Number		Dimension (m)			Area	Volume
				X	Y	Z		
Ground	Office	17	8	4	4	3.5	16	56
			2	5	4	3.5	20	70
			7	4.5	4	3.5	18	63
	Clinic	1		6	12	4	72	288
1 st floor	Office	16	1	12	4	3.5	60	168
			15	5	4	3.5	20	70
	Meeting room	1	1	12	4	4	48	192

By using above each indoor polluting parameter concentration of the each polluting compounds in each building office, lastly it is possible to calculate emission rate for each office and the overall concentration of pollutant in building by using box modeling.

$$C = C_o + \frac{Q_A \Delta X}{UZ} \quad (4.29)$$

From this above equation it's possible to calculate the emission rate

$$Q_A = \frac{C_o}{\Delta X . \Delta Y . Z}$$

The emission rate of CO for ground floor single office was given by:

$$Q_A = \frac{C_o}{\Delta X . \Delta Y . Z}$$

$$Q_A = \frac{407000 \text{ mg} / \text{m}^3 \text{s}}{4 * 4 * 3.5}$$

$$Q_A = \frac{407000 \text{ mg} / \text{m}^3 \text{s}}{56 \text{m}^3}$$

$$Q_A = 72678.6 \text{ mg} / \text{s}$$

Table 4. 11: Emission from each office of the building

Floor	Name	Number		Dimension (m)			Emission rate Per room (mg / s)
				X	Y	Z	
Ground	Office	17	1	4	4	3.5	72678.6
			1	5	4	3.5	58142.9
			1	4.5	4	3.5	64603.2
	Clinic	1		6	12	4	14132
1 st floor	Office	16	1	12	4	3.5	24226.2
			1	5	4	3.5	58142.9
	Meeting room	1		12	4	4	21197.9

In above table the emission from toilet room where considered as a zero. In case of this study the ventilation system used for toilet room was natural ventilation due to this it considered as zero.

$$C = C_o + \frac{Q_A . \Delta X}{UZ}$$

Table 4. 12: Total indoor concentration of the pollutant.

Floor	Name	Number		Dimension (m)			Emission rate Per total uniform room (mg / s)
				X	Y	Z	
Ground	Office	17	8	4	4	3.5	581428.8
			2	5	4	3.5	116285.8
			7	4.5	4	3.5	452222.4
	Clinic	1		6	12	4	14132
1 st floor	Office	16	1	12	4	3.5	24226.2
			15	5	4	3.5	871243.5
	Meeting room	1		12	4	4	21197.9

4.4 Ventilation System

From above outdoor and indoor numerical analysis result, it is clear that the outdoor and indoor air quality of Batu caustic soda and cement factory where polluted. The air quality indicator value from analysis was more than that given by international as well as Ethiopian air quality standard. For this case it is necessary to design the proper air filtration system as well as air ventilation system. In order to filter and deliver the purified air in to office building it should be known the amount of air needed for the building office. In the following section below designing of ventilation system and its analysis will take place.

4.4.1 Heating and Cooling Loads

Heating load calculations are used to evaluate the building's heat loss in the winter and determine the required heating capabilities. During the winter, the peak heating load usually comes before daybreak, and exterior temperatures do not vary greatly during the season. Internal heat sources, such as people or appliances, are also advantageous since they compensate for some of the heat losses. As a result, heat load calculations are typically performed under steady state settings (no solar radiation and constant external temperatures) while internal heat sources are ignored.

This is a simple but conservative approach that leads to slight overestimation of the heating capacity. For more accurate estimation of heating loads, taking into account the thermal capacity of the walls and internal heat sources, which makes the problem more complicated. For estimating cooling loads, one has to consider the unsteady state processes, as the peak cooling load occurs during the day time and the outside conditions also vary significantly throughout the day due to solar radiation.

In addition, all internal sources add on to the cooling loads and neglecting them would lead to underestimation of the required cooling capacity and the possibility of not being able to maintain the required indoor conditions. Thus cooling load calculations are inherently more complicated. In determining the heating load, credit for solar heat gain or internal heat gains is usually not included and the thermal storage effects of building structure are generally ignored. Whereas in cooling load calculations, the thermal storage characteristics of the building play a vital role because the time at which the space may realize the heat gain as a cooling load will be considerably offset from the time the heat started to flow.

4.4.1.1 Components of Cooling Load and Heating Load

The total building cooling load consists of heat transferred through the building envelope (walls, roof, floor, windows, doors etc.) and heat generated by occupants, equipment and lights. The load due to heat transfer through the envelope is called as external load, while all other loads are called as internal loads. The percentage of external versus internal load varies with building type, site climate, and building design. The total cooling load on any building consists of both sensible as well as latent load components.

The sensible load affects the dry bulb temperature, while the latent load affects the moisture content of the conditioned space. Buildings may be classified as externally loaded and internally loaded. In externally loaded buildings the cooling load on the building is mainly due to heat transfer between the surroundings and the internal conditioned space. Since the surrounding conditions are highly variable in any given day, the cooling load of an externally loaded building varies widely.

In internally loaded buildings the cooling load is mainly due to internal heat generating sources such as occupants, lights or appliances. In general the heat generation due to internal heat sources may remain fairly constant, and since the heat transfer from the variable surroundings is much less compared to the internal heat sources, the cooling load of an internally loaded building remains fairly constant.

Cooling load calculation method

For a thorough calculation of the zones and whole-building loads, one of the following three methods should be employed:

- ***Transfer Function Method (TFM)***: This is the most complex of the methods proposed by ASHRAE and requires the use of a computer program or advanced spreadsheet.
- ***Cooling Load Temperature Differential/Cooling Load Factors (CLTD/CLF)***: This method is derived from the TFM method and uses tabulated data to simplify the calculation process. The method can be fairly easily transferred into simple spreadsheet programs but has some limitations due to the use of tabulated data.
- ***Total Equivalent Temperature Differential/Time-Averaging (TETD/TA)***: This was the preferred method for hand or simple spreadsheet calculation before the introduction of the CLTD/CLF method.

Outdoor Design Weather Conditions

ASHRAE Handbook 1993 Fundamentals (Chapter 26) list tables of climate conditions for International locations: The information provided in table 1a, 2a and 3a are for heating design conditions that include:

- Dry bulb temperatures corresponding to 99.6% and 99% annual cumulative frequency of occurrence,
- Wind speeds corresponding to 1%, 2.5% and 5% annual cumulative frequency of occurrence, c. Wind direction most frequently occurring with 99.6% and 0.4% dry-bulb temperatures and
- Average of annual extreme maximum and minimum dry-bulb temperatures and standard deviations.

For cooling and humidity control conditions that include:

- Dry bulb temperature corresponding to 0.4%, 1.0% and 2.0% annual cumulative frequency of occurrence and the mean coincident wet-bulb temperature (warm). These conditions appear in sets of dry bulb (DB) temperature and the mean coincident wet bulb (MWB) temperature since both values are needed to determine the sensible and latent (dehumidification) loads in the cooling mode.
- Wet-bulb temperature corresponding to 0.4%, 1.0% and 2.0% annual cumulative frequency of occurrence and the mean coincident dry-bulb temperature
- Dew-point temperature corresponding to 0.4%, 1.0% and 2.0% annual cumulative frequency of occurrence and the mean coincident dry-bulb temperature and humidity ratio (calculated for the dew-point temperature at the standard atmospheric pressure at the elevation of the station).
- Mean daily range (DR) of the dry bulb temperature, which is the mean of the temperature difference between daily maximum and minimum temperatures for the warmest month (highest average dry-bulb temperature). These are used to correct CLTD values.

Indoor Design Conditions and Thermal Comfort

The indoor design conditions are directly related to human comfort. Current comfort standards, ASHRAE Standard 55-1992 [4] and ISO Standard 7730 [5], specify a “comfort zone,” representing the optimal range and combinations of thermal factors (air temperature, radiant temperature, air

velocity, humidity) and personal factors (clothing and activity level) with which at least 80% of the building occupants are expected to express satisfaction. The environmental factors that affect the thermal comfort of the occupants in an air-conditioned space are mainly:

- Metabolic rate, expressed in met (1 met = 18.46 Btu/hr.ft²) determines the amount of heat that must be released from the human body and it depends mainly on the intensity of the physical activity.
- Indoor air temperature (T_r) and mean radiant temperature (T_{rad}), both in °F. T_r affects both the sensible heat exchange and evaporative losses, and T_{rad} affects only sensible heat exchange.
- Relative humidity of the indoor air in %, which is the primary factor that influences evaporative heat loss.
- Air velocity of the indoor air in fpm, which affects the heat transfer coefficients and therefore the sensible heat exchange and evaporative loss.
- Clothing insulation in clo (1 clo = 0.88 h.ft². °F/Btu), affects the sensible heat loss. Clothing insulation for occupants is typically 0.6 clo in summer and 0.8 to 1.2 clo in winter.

Indoor Air Quality and Outdoor Air Requirements

According to the National Institute for Occupational Safety and Health (NIOSH), 1989, the causes of indoor air quality complaints in buildings are inadequate outdoor ventilation air. There are three basic means of improving indoor air quality: (1) eliminate or reduce the source of air pollution, (2) enhance the efficiency of air filtration, and (3) increase the ventilation (outdoor) air intake. Abridged outdoor air requirements listed in ANSI/ASHRAE Standard 62-1989 are as follows:

Table 4. 13: Abridged outdoor air requirements listed in ANSI/ASHRAE Standard 62-1989

Applications	Cfm/person
Offices, conference rooms, offices	20
Retail Stores	0.2 – 0.3 cfm/ft ²
Classrooms, theaters, auditoriums	15
Hospitals patient rooms	25

4.4.1.1 Cooling and Heating Load Hap4.9 Numerical Analysis

The total cooling load on a building consists of external as well as internal loads was computed by using Hap 4.9 software and the result where summarized in following section.

Cooling Load

The external loads consist of heat transfer by conduction through the building walls, roof, floor, doors etc., heat transfer by radiation through fenestration such as windows and skylights. All these are sensible heat transfers. To calculate the external cooling load of the building office, let take the building as a space divided in to four zone, those are West, East, North and South. In this case west is the first Zone, North was the second, east was the thread and south is the fourth.

Table 4. 14: Dimension of office, wall, roof and window.

Floor	Zone	Number of External Wall	Wall Side	Number Of room	Area of the wall (m ²)	Area of the window (m ²)	Area of the roof (m ²)
Ground Floor	1	1	W	8	14	1.5	16
	1	1	W	1	24	3.75	72
	2	1	N	2	14	1.5	16
	3	1	E	7	16	1.5	18
	4	1	S	1	48	6	72
First floor	1	1	W	1	42	5.25	48
	1	1	W	8	17.5	3.75	20
	2	1	N	2	17.5	3.75	48
	3	1	E	6	17.5	3.75	20
	3	1	E	1	76	11.25	95
	4	1	S	1	76	11.25	95
	4	1	S	1	17.5	3.75	20

The cooling load and heating load of the building was computed by using Hap 4.9 software and the result were summarized as follows.

Ground floor Result

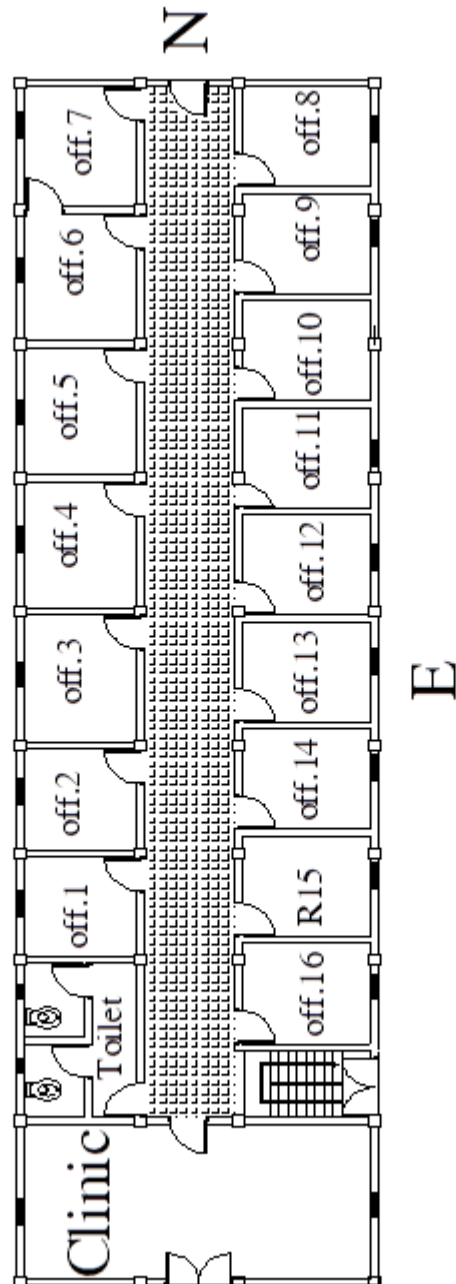


Figure 4. 16: Floor site of ground floor

Table 4. 15: Zone Sizing Data

Room Name	Maximum Cooling Sensible (kW)	Design Airflow (L/s)	Minimum Airflow (L/s)	Time of Peak Load	Zone Floor Area (m²)	Zone L/(s-m²)
Zone 1	3.0	268	52	Dec 21	72.0	3.73
Zone 2	0.7	65	25	Aug 22	24.0	2.72
Zone 3	0.7	65	7	Aug 22	16.0	4.09
Zone 4	0.7	65	7	Aug 22	16.0	4.09
Zone 5	0.7	65	7	Aug 22	16.0	4.09
Zone 6	0.7	65	7	Aug 22	16.0	4.09
Zone 7	0.7	65	7	Aug 22	16.0	4.09
Zone 8	0.7	65	7	Aug 22	16.0	4.09
Zone 9	0.7	65	7	Aug 22	16.0	4.09
Zone 10	1.2	109	14	Jun 19	20.0	5.47
Zone 11	1.1	98	7	Jun 22	16.0	6.14
Zone 12	1.0	90	19	Aug 17	20.0	4.50
Zone 13	0.8	75	14	Aug 17	20.0	3.75
Zone 14	0.8	75	14	Aug 17	20.0	3.75
Zone 15	0.8	75	14	Aug 17	20.0	3.75
Zone 16	0.8	75	14	Aug 17	20.0	3.75
Zone 17	0.8	75	14	Aug 17	20.0	3.75
Zone 18	0.8	75	14	Aug 17	20.0	3.75
Zone 19	0.8	75	14	Aug 17	20.0	3.75
Zone 20	0.7	61	32	Aug 23	90.0	0.67

Table 4. 16: Space Loads and Airflows

Space Name	Mult.	Cooling Sensible (kW)	Time of Load	Air Flow (L/s)	Floor Area (m²)	Space L/(s-m²)
Clinic	1	3.0	Dec 21	268	72.0	3.73
toilet	1	0.7	Aug 22	65	24.0	2.72
Office-1	1	0.7	Aug 22	65	16.0	4.09
Office-2	1	0.7	Aug 22	65	16.0	4.09
Office-3	1	0.7	Aug 22	65	16.0	4.09
Office-4	1	0.7	Aug 22	65	16.0	4.09
Office-5	1	0.7	Aug 22	65	16.0	4.09
Office-6	1	0.7	Aug 22	65	16.0	4.09
Office-7	1	0.7	Aug 22	65	16.0	4.09
Office-8	1	1.2	Jun 19	109	20.0	5.47
Office-9	1	1.1	Jun 22	98	16.0	6.14
Office-10	1	1.0	Aug 17	90	20.0	4.50
Office-11	1	0.8	Aug 17	75	20.0	3.75
Office-12	1	0.8	Aug 17	75	20.0	3.75
Office-13	1	0.8	Aug 17	75	20.0	3.75
Office-14	1	0.8	Aug 17	75	20.0	3.75
Office-15	1	0.8	Aug 17	75	20.0	3.75
Office-16	1	0.8	Aug 17	75	20.0	3.75
Corridor	1	0.7	Aug 23	61	90.0	0.67

Table 4. 17: Ventilation size summer Result.

Space Name	Mult.	Floor Area(m ²)	Maximum Occupants	Maximum Supply Air (L/s)	Required Outdoor Air (L/s/p)	Required Outdoor Air (L/(s-m ²))	Uncorrected Outdoor Air (L/s)
Clinic	1	72.0	3.0	268.4	10.00	0.30	51.6
toilet	1	24.0	1.0	65.2	3.80	0.90	25.4
Office-1	1	16.0	1.0	65.4	2.50	0.30	7.3
Office-2	1	16.0	1.0	65.4	2.50	0.30	7.3
Office-3	1	16.0	1.0	65.4	2.50	0.30	7.3
Office-4	1	16.0	1.0	65.4	2.50	0.30	7.3
Office-5	1	16.0	1.0	65.4	2.50	0.30	7.3
Office-6	1	16.0	1.0	65.4	2.50	0.30	7.3
Office-7	1	16.0	1.0	65.4	2.50	0.30	7.3
Office-8	1	20.0	3.0	109.5	2.50	0.30	13.5
Office-9	1	16.0	1.0	98.2	2.50	0.30	7.3
Office-10	1	20.0	5.0	90.1	2.50	0.30	18.5
Office-11	1	20.0	3.0	75.1	2.50	0.30	13.5
Office-12	1	20.0	3.0	75.1	2.50	0.30	13.5
Office-13	1	20.0	3.0	75.1	2.50	0.30	13.5
Office-14	1	20.0	3.0	75.1	2.50	0.30	13.5
Office-15	1	20.0	3.0	75.1	2.50	0.30	13.5
Office-16	1	20.0	3.0	75.1	2.50	0.30	13.5
Corridor	1	90.0	2.0	60.7	2.50	0.30	32.0
Totals				1675.5			293.9

Table 4. 18: Peak load detail summary

ZONE LOADS	DESIGN COOLING		
	COOLING DATA AT Aug 1800		
	COOLING OA DB / WB 23.5 °C / 14.6 °C		
	Details	Sensible (W)	Latent (W)
Window & Skylight Solar Loads	37 m²	4060	-
Wall Transmission	311 m²	2812	-
Roof Transmission	494 m²	912	-
Window Transmission	37 m²	-186	-
Skylight Transmission	0 m²	0	-
Door Loads	6 m²	-15	-
Floor Transmission	494 m²	0	-
Partitions	8 m²	0	-
Ceiling	8 m²	0	-
Overhead Lighting	593 W	593	-
Task Lighting	0 W	0	-
Electric Equipment	692 W	692	-
People	43	3087	2583
Infiltration	-	0	0
Miscellaneous	-	0	0
Safety Factor	15% / 10%	1793	258
>> Total Zone Loads	-	13749	2842
Zone Conditioning	-	12736	2842
Plenum Wall Load	0%	0	-
Plenum Roof Load	0%	0	-
Plenum Lighting Load	0%	0	-
Return Fan Load	1105 L/s	0	-
Ventilation Load	294 L/s	-139	-1122
Supply Fan Load	1105 L/s	0	-
Space Fan Coil Fans	-	0	-
Duct Heat Gain / Loss	0%	0	-
>> Total System Loads	-	12597	1720
Central Cooling Coil	-	3039	1720
Central Heating Coil	-	0	-
Precool Coil	-	9558	0
Preheat Coil	-	0	-
>> Total Conditioning	-	12597	1720
Key:	Positive values are clg loads		
	Negative values are htg loads		

Supply and Return Fan Sizing Data

Actual max L/s at Aug 22 1430 L/s
Standard L/s 1173 L/s
Actual max L/(s-m²) 2.89 L/(s-m²)

Outdoor Ventilation Air Data

Design Ventilation Airflow Rate 294 L/s
Design airflow L/s 294 L/s
L/(s-m²) 0.59 L/(s-m²)
L/s/person 6.83 L/s/person

Maximum cooling will take place on August 18:

Load occurs at Aug 18
OA DB / WB 23.5 / 14.6 °C
Entering DB / WB 24.3 / 16.9 °C
Leaving DB / WB 15.6 / 14.1 °C
Bypass Factor 0.100

Location: BATU, ETHIOPIA
Altitude: 1643.0 m.
Data for: August DESIGN COOLING DAY, 1800

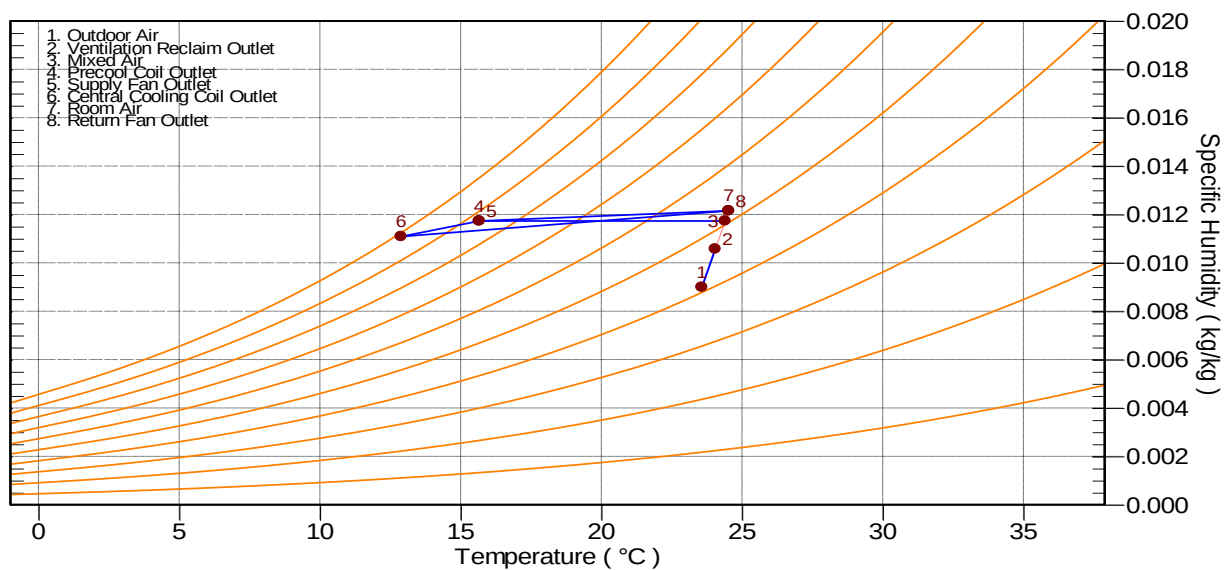


Figure 4. 17: Psychrometric analysis result for ventilation system of ground floor.

First floor Hap 4.9 Result

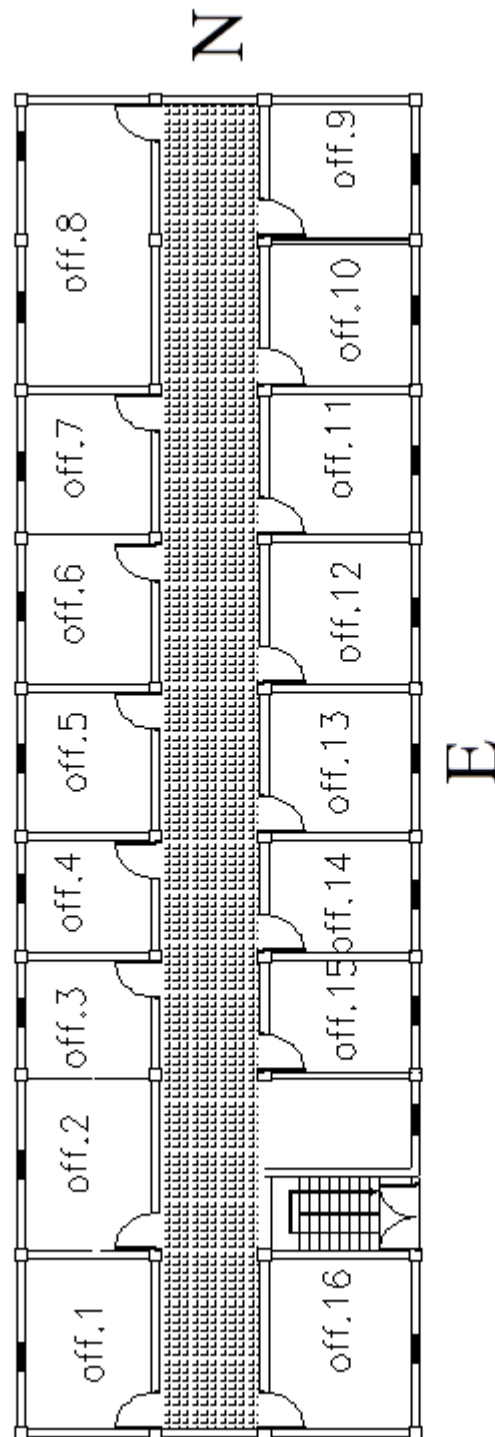


Figure 4. 18: Floor site of first floor.

Table 4. 19: Zone Sizing Data

Zone Name	Maximum Cooling Sensible (kW)	Design Airflow (L/s)	Minimum Airflow (L/s)	Time of Peak Load	Maximum Heating Load (kW)	Zone Floor Area (m ²)	Zone L/(s-m ²)
Zone 1	0.9	83	1	Aug 22	0.3	16.0	5.18
Zone 2	0.9	83	1	Aug 22	0.3	16.0	5.18
Zone 3	0.7	68	1	Aug 22	0.3	16.0	4.24
Zone 4	0.7	68	1	Aug 22	0.3	16.0	4.24
Zone 5	0.7	68	1	Aug 22	0.3	16.0	4.24
Zone 6	0.7	68	1	Aug 22	0.3	16.0	4.24
Zone 7	0.7	68	1	Aug 22	0.3	16.0	4.24
Zone 8	0.7	68	1	Aug 22	0.3	16.0	4.24
Zone 9	1.0	92	1	Jun 18	0.5	16.0	5.76
Zone 10	0.6	57	1	Aug 16	0.3	16.0	3.54
Zone 11	0.6	57	1	Aug 16	0.3	16.0	3.54
Zone 12	0.6	57	1	Aug 16	0.3	16.0	3.54
Zone 13	0.6	57	1	Aug 16	0.3	16.0	3.54
Zone 14	0.6	57	1	Aug 16	0.3	16.0	3.54
Zone 15	0.6	57	1	Aug 16	0.3	16.0	3.54
Zone 16	3.2	291	3	Oct 17	1.0	48.0	6.06
Zone 17	1.3	119	1	Jun 22	0.6	24.0	4.96
Zone 18	4.1	376	4	Aug 23	0.4	180.0	2.09
Zone 19	2.1	188	2	Aug 23	0.2	90.0	2.09

Table 4. 20: Space Loads and Airflows

Space Name	Mult.	Cooling Sensible (kW)	Time of Load	Air Flow (L/s)	Heating Load (kW)	Floor Area (m ²)	Space L/(s-m ²)
Office-15	1	0.9	Aug 22	83	0.3	16.0	5.18
Office-14	1	0.9	Aug 22	83	0.3	16.0	5.18
Office-13	1	0.7	Aug 22	68	0.3	16.0	4.24
Office-12	1	0.7	Aug 22	68	0.3	16.0	4.24
Office-11	1	0.7	Aug 22	68	0.3	16.0	4.24
Office-10	1	0.7	Aug 22	68	0.3	16.0	4.24
Office-9	1	0.7	Aug 22	68	0.3	16.0	4.24
Office-7	1	1.0	Jun 18	92	0.5	16.0	5.76
Office-6	1	0.6	Aug 16	57	0.3	16.0	3.54
Office-5	1	0.6	Aug 16	57	0.3	16.0	3.54
Office-4	1	0.6	Aug 16	57	0.3	16.0	3.54
Office-3	1	0.6	Aug 16	57	0.3	16.0	3.54
Office-2	1	0.6	Aug 16	57	0.3	16.0	3.54
Office-1	1	0.6	Aug 16	57	0.3	16.0	3.54
Meeting Room (office 16)	1	3.2	Oct 17	291	1.0	48.0	6.06
Finance Office (office 8)	1	1.3	Jun 22	119	0.6	24.0	4.96
corridor	2	2.1	Aug 23	188	0.2	90.0	2.09

Table 4. 21 Ventilation system design summery

Space Name	Mult.	Floor Area (m²)	Maximum Occupants	Maximum Supply Air (L/s)
Office-15	1	16.0	4.0	82.8
Office-14	1	16.0	4.0	82.8
Office-13	1	16.0	2.0	67.8
Office-12	1	16.0	2.0	67.8
Office-11	1	16.0	2.0	67.8
Office-10	1	16.0	2.0	67.8
Office-9	1	16.0	2.0	67.8
Office-7	1	16.0	2.0	92.2
Office-6	1	16.0	1.0	56.6
Office-5	1	16.0	1.0	56.6
Office-4	1	16.0	1.0	56.6
Office-3	1	16.0	1.0	56.6
Office-2	1	16.0	1.0	56.6
Office-1	1	16.0	1.0	56.6
Meeting Room (office 16)	1	48.0	20.0	290.7
Finance Office (office 8)	1	24.0	3.0	118.9
corridor	2	90.0	20.0	187.9
Totals				1977.8

Table 4. 22: Peak load detail summary

ZONE LOADS	DESIGN COOLING		
	COOLING DATA AT Aug 2200		
	COOLING OA DB / WB 17.1 °C / 12.4 °C		
		Sensible	Latent
	Details	(W)	(W)
Window & Skylight Solar Loads	29 m²	4500	-
Wall Transmission	286 m²	3356	-
Roof Transmission	582 m²	815	-
Window Transmission	29 m²	-566	-
Skylight Transmission	0 m²	0	-
Door Loads	0 m²	0	-
Floor Transmission	312 m²	-820	-
Partitions	0 m²	0	-
Ceiling	0 m²	0	-
Overhead Lighting	698 W	698	-
Task Lighting	0 W	0	-
Electric Equipment	815 W	815	-
People	111	7970	6669
Infiltration	-	0	0
Miscellaneous	-	0	0
Safety Factor	15% / 10%	2515	667
>> Total Zone Loads	-	19284	7336
Zone Conditioning	-	18433	7336
Plenum Wall Load	0%	0	-
Plenum Roof Load	0%	0	-
Plenum Lighting Load	0%	0	-
Return Fan Load	1580 L/s	0	-
Ventilation Load	0 L/s	0	0
Supply Fan Load	1580 L/s	0	-
Space Fan Coil Fans	-	0	-
Duct Heat Gain / Loss	0%	0	-
>> Total System Loads	-	18433	7336
Central Cooling Coil	-	18433	7341
Central Heating Coil	-	0	-
Preheat Coil	-	0	-
>> Total Conditioning	-	18433	7341
Key:	Positive values are clg loads		
	Negative values are htg loads		

Supply and Return Fan Sizing Data

Actual max L/s at Aug 22 1758 L/s
Standard L/s 1441 L/s
Actual max L/(s-m²) 3.02 L/(s-m²)
Load occurs at Aug 22
OA DB / WB 17.1 / 12.4 °C
Entering DB / WB 24.6 / 17.5 °C
Leaving DB / WB 12.8 / 12.1 °C
Bypass Factor 0.100
Resulting RH..... 53%
Design supply temperature 12.8°C

Location: Batu, ETHIOPIA
Altitude: 1643.0 m.
Data for: August DESIGN COOLING DAY, 2200

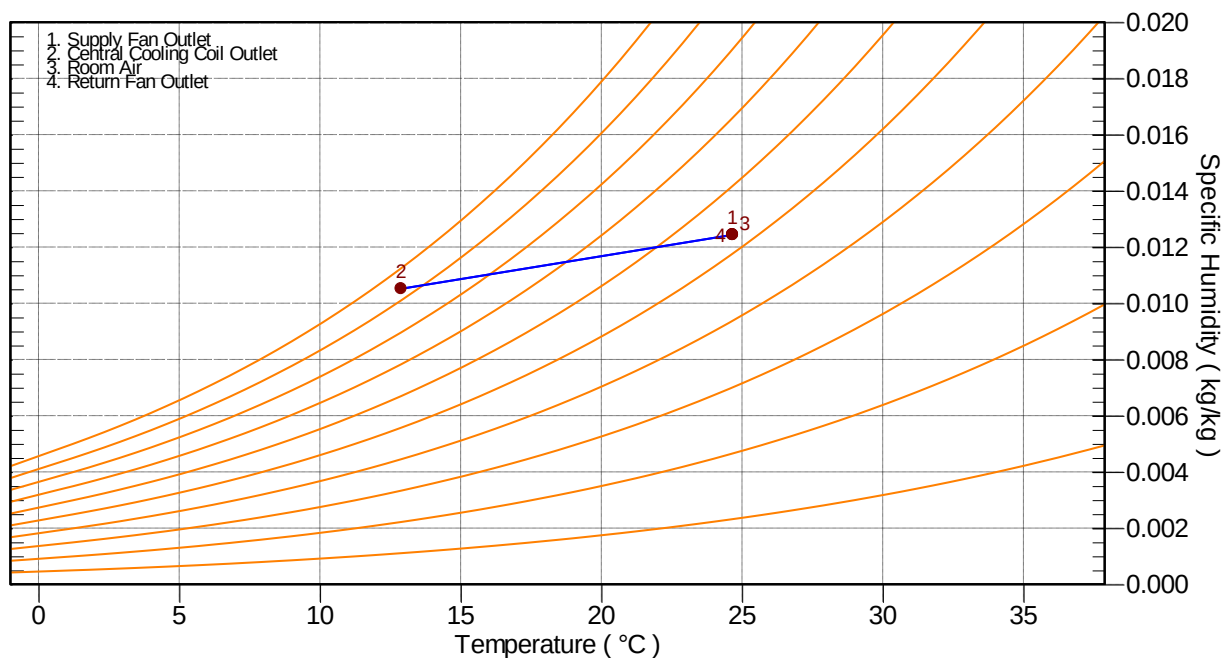


Figure 4. 19: Psychrometric analysis result for ventilation system of first floor.

Air heating value

If air used needed for heating the required air flow rate will calculated as:

$$q_h = H_h / (\rho \cdot c_p (t_s - t_r)) \quad (4.30)$$

Where:

- ✓ q_h = volume of air for heating (m^3/s)
- ✓ H_h = heat load (W)
- ✓ ρ = density of air
- ✓ c_p = specific heat of air
- ✓ t_s = supply temperature
- ✓ t_r = room temperature

Ground floor air heating value	First floor air heating value
$H_h = 28836 W$	$H_h = 18520 W$
$\rho = 1.276 \text{ Kg} / m^3$	$\rho = 1.276 \text{ Kg} / m^3$
$c_p = 1.006 \text{ KJ} / \text{KgK}$	$c_p = 1.006 \text{ KJ} / \text{KgK}$
$t_s = 24.3^\circ C$	$t_s = 24.6^\circ C$
$t_r = 23.5^\circ C$	$t_r = 17.1^\circ C$
$q_h = H_h / (\rho \cdot c_p (t_s - t_r))$	$q_h = H_h / (\rho \cdot c_p (t_s - t_r))$
$q_h = \frac{28836}{(1.276 * 1.006 (24.3 - 23.5))}$	$q_h = \frac{18520}{(1.276 * 1.006 (24.6 - 17.1))}$
$q_h = 27996.1 \text{ l} / s$	$q_h = 1923.2 \text{ l} / s$

Air cooling value

$$q_c = H_c / (\rho \cdot c_p(t_o - t_r))$$

Ground floor air cooling value	First floor air cooling value
$H_c = 45225W$	$H_c = 68163W$
$\rho = 1.276 \text{ Kg} / \text{m}^3$	$\rho = 1.276 \text{ Kg} / \text{m}^3$
$c_p = 1.006 \text{ KJ} / \text{KgK}$	$c_p = 1.006 \text{ KJ} / \text{KgK}$
$t_s = 24.3^\circ \text{C}$	$t_s = 24.6^\circ \text{C}$
$t_r = 23.5^\circ \text{C}$	$t_r = 17.1^\circ \text{C}$
$q_c = H_c / (\rho \cdot c_p(t_o - t_r))$	$q_h = H_c / (\rho \cdot c_p(t_s - t_r))$
$q_h = \frac{45225}{(1.276 * 1.006(24.3 - 23.5))}$	$q_h = \frac{68163}{(1.276 * 1.006(24.6 - 17.1))}$
$q_h = 43907.8 \text{ l} / \text{s}$	$q_h = 7078.2 \text{ l} / \text{s}$

Moisture

Humidifying

Outdoor air of Batu caustic soda was more humidified than indoor air. That outdoor relative humidity was 31% and indoor relative humidity was 21%, in this case it is necessary to humidify indoor air by transferring outdoor air outdoor air. The amounts of supply air can be calculated as:

$$q_{mh} = q_h / (\rho(X_1 - X_2))$$

Ground floor humidifying rate	First floor humidifying rate
$q_h = Q * \rho$	$q_h = Q * \rho$
$Q = 19694 \text{ l} / \text{s} = 1.9694 \text{ m}^3 / \text{s}$	$Q = 1977.8 \text{ l} / \text{s} = 1.9778 \text{ m}^3 / \text{s}$
$\rho = 1.276 \text{ kg} / \text{m}^3$	$\rho = 1.276 \text{ kg} / \text{m}^3$
$q_h = 1.9694 * 1.276$	$q_h = 1.9778 * 1.276$
$q_h = 2.513 \text{ kg} / \text{s}$	$q_h = 2.524 \text{ kg} / \text{s}$
$X_1 = 0.0125 \text{ kg} / \text{kg}$	$X_1 = 0.0121 \text{ kg} / \text{kg}$
$X_2 = 0.009 \text{ kg} / \text{kg}$	$X_2 = 0.011 \text{ kg} / \text{kg}$
$q_{mh} = \frac{2.513}{(1.276 * (0.0125 - 0.009))}$	$q_{mh} = \frac{2.524}{(1.276 * (0.0121 - 0.011))}$
$q_{mh} = 558.4 \text{ m}^3 / \text{s}$	$q_{mh} = 1801.2 \text{ m}^3 / \text{s}$

4.5 Duct Design

The design of duct systems is an important factor for effective, energy-efficient, and comfortable heating, ventilating, and air-conditioning (HVAC) systems. Duct system design begin with the calculation of duct area for air delivering by using continuity equation.

$$Q = AV \quad (4.31)$$

where:

$Q = \text{flow rate} (m^3 / s)$

$A = \text{area of the duct} (m^2)$

$V = \text{velocity of air} (m^2 / s)$

The total amount of air needed to be delivered to the ground floor and first floor of the building per each second was 1.9694 m³/s and 1.9694 m³/s.

The velocity from standard hand book of ASHRAE 1997 for office 1200 fpm (6.1 m/s), for meeting room 800 fpm (4.1 m/s) and for clinic also 800 fpm (4.1 m/s). In case of this research study round duct where best choose due to its high efficiency in transporting air, create less friction and allow the air to flow in the building easily. Area of round duct will be given by:

Velocity reduction method

With this method the same value friction loss per length of the duct is used to size each section of the duct in the system. The friction loss rate is chosen to result in an economic balance between duct cost and energy cost. A higher friction loss results in a smaller ducts friction loss results in a smaller duct but higher fan operating cost. This method is most universal used for supply duct, return and exhaust air duct for a low and medium pressure up to 4'' of Wg.

Main duct design

To design the main duct the distribution of air volume per each room of building play a vital role in controlling the aspect ratio value. Total amount of ventilation rate per given volume of the building was 3947.2 l/s, from this 1969.4 l/s was for ground floor and 1977.8 l/s for first floor.

Supply Main duct

After evaluation for the maximum ventilate air flow rate then the velocity of the duct must be assured as it is below the value. This will be carried out from the table 7 ASHRAE standard air duct design hand book 2003. From the table under velocity reduction method the maximum velocity for main duct was 2000 fpm (10.162 m/s), for return duct 1500 fpm (7.623 m/s), for branch supply duct 1600 fpm (8.13 m/s), for return branch duct 1200 fpm (6.1 m/s). The first room that the main duct is going to deliver first the air was clinic room on the ground floor.

The above area of main duct was the area of duct lied from air handling unit to clinic only, for record the room on the ground floor have 20 node And the first floor have 18 node. This means that for ground floor there the main duct diameter should be calculated was 19 and for first floor 17 diameter for main duct should be calculated.

The main duct from air filtration unit was has ventilation rate of 20214 m³/hr needed to delivered for the ground rooms as well as for those on first floor, from the Mcquay software calculation result was: velocity 8.052 m/s, diameter 942.2 mm (0.9422 m), flow area 0.6972 m², fluid velocity in the duct 8.054 m/s, Reynolds number 487399, friction factor 0.0157 and pressure 37.9845 Pa.

Table 4. 23: Main supply duct sizing for ground floor of the building.

Space Name	Maximum Supply Air (L/s)	Velocity (m/s)	Head Loss (Pa/m)	Flow Area (m ²)	Diameter Of duct (m)	Velocity Pressure (Pa)
AHU to Clinic	1969.4	10	2.117	0.1969	0.5007	60.11
Clinic to Toilet	1701	10	2.316	0.1701	0.4654	60.085
Toilet to Office-1	1635.8	10	2.372	0.1636	0.4564	60.07
Office-(1-2)	1570.4	10	2.44	0.1571	0.4472	58.52
Office-(2-3)	1505	10	2.504	0.1505	0.4377	58.56
Office-(3-4)	1439.6	10	2.574	0.1439	0.4281	58.61
Office-(4-5)	1374.2	10	2.648	0.1374	0.4183	58.58
Office-(5-6)	1308.8	10	2.728	0.1309	0.4082	58.55
Office-(6-7)	1242.6	10	2.817	0.1243	0.3978	58.53
Office-(7-8)	1133.1	10	2.981	0.1133	0.3798	58.57
Office-(8-9)	1034.9	10	3.152	0.1035	0.363	58.55
Office-(9-10)	944.8	10	3.334	0.0945	0.3468	58.54
Office-(10-11)	869.7	10	3.51	0.087	0.3328	58.52
Office-(11-12)	794.6	10	3.71	0.0795	0.3181	58.5
Office-(12-13)	719.5	10	3.994	0.072	0.3027	58.48
Office-(13-14)	644.4	10	4.222	0.0644	0.2864	58.64
Office-(14-15)	569.3	10	4.558	0.0569	0.2692	58.62
Office-(15-16)	494.2	10	4.976	0.0494	0.2508	58.61
Office-(16-17)	419.1	10	5.512	0.0419	0.231	58.59
Office-17 to Corridor	358.4	10	6.075	0.0358	0.2136	58.7

Table 4. 24: Main supply duct sizing for first floor of the building.

Space Name	Maximum Supply Air (L/s)	Velocity (m/s)	Head Loss (Pa/m)	Flow Area (m ²)	Diameter Of duct (m)	Velocity Pressure (Pa)
Office-(14-15)	911.9	10	3.407	0.0912	0.3407	58.55
Office-(13-14)	994.1	10	3.231	0.0994	0.3558	58.57
Office-(12-13)	1076.9	10	3.076	0.1077	0.3703	58.55
Office-(11-12)	1144.7	10	2.962	0.1145	0.3818	58.5
Office-(10-11)	1212.5	10	2.859	0.1212	0.3929	58.61
Finance Office to Office-10	1280.3	10	2.765	0.1280	0.4037	58.59
Office-(8-9)	1467	10	2.544	0.1467	0.4322	58.56
Office-(7-8)	1534.8	10	2.474	0.1535	0.4421	58.54
Office-(6-7)	1602.6	10	2.41	0.1602	0.4517	58.61
Office-(5-6)	1694.8	10	2.329	0.1695	0.4645	58.549
Office-(4-5)	1751.4	10	2.282	0.1751	0.4722	58.59
Office-(3-4)	1808	10	2.238	0.1808	0.4798	58.56
Office-(2-3)	1864.6	10	2.197	0.1864	0.4872	58.6
Office-(1-2)	1921.2	10	2.157	0.1921	0.4946	58.575
Source to Office-1	1977.8	10	2.119	0.1978	0.5018	58.55
Office-15 to Meeting Room	621.2	10	4.318	0.0621	0.2812	58.6
Office-9 to Finance Office	1348.1	10	2.679	0.1348	0.4143	58.57
Meeting Room to corridor	433.3	10	5.399	0.0433	0.2349	58.64

Supply branch duct sizing

Table 4. 25: Branch supply duct sizing result for ground floor of the building.

Space Name	Floor Area(m ²)	Maximum Occupants	Maximum Supply Air (L/s)	Velocity (m/s)	Head Loss (Pa/m)	Flow Area (m ²)	Diameter (m)
Clinic	72.0	3.0	268.4	4.1	0.765	0.0655	0.2887
toilet	24.0	1.0	65.2	6.1	5.038	0.01068	0.1166
Office-1	16.0	1.0	65.4	6.1	5.038	0.01068	0.1166
Office-2	16.0	1.0	65.4	6.1	5.038	0.01068	0.1166
Office-3	16.0	1.0	65.4	6.1	5.038	0.01068	0.1166
Office-4	16.0	1.0	65.4	6.1	5.038	0.01068	0.1166
Office-5	16.0	1.0	65.4	6.1	5.038	0.01068	0.1166
Office-6	16.0	1.0	65.4	6.1	5.038	0.01068	0.1166
Office-7	16.0	1.0	65.4	6.1	5.038	0.01068	0.1166
Office-8	20.0	3.0	109.5	6.1	3.628	0.01796	0.1512
Office-9	16.0	1.0	98.2	6.1	3.886	0.01611	0.1432
Office-10	20.0	5.0	90.1	6.1	4.102	0.01476	0.1371
Office-11	20.0	3.0	75.1	6.1	4.602	0.01231	0.1252
Office-12	20.0	3.0	75.1	6.1	4.602	0.01231	0.1252
Office-13	20.0	3.0	75.1	6.1	4.602	0.01231	0.1252
Office-14	20.0	3.0	75.1	6.1	4.602	0.01231	0.1252
Office-15	20.0	3.0	75.1	6.1	4.602	0.01231	0.1252
Office-16	20.0	3.0	75.1	6.1	4.602	0.01231	0.1252
Office-17	20.0	3.0	75.1	6.1	4.602	0.01231	0.1252
Corridor	9.0	2.0	60.7	6.1	5.266	0.00996	0.1126

Table 4. 26: Branch supply duct sizing result for first floor of the building.

Space Name	Floor Area (m ²)	Maximum Occupants	Maximum Supply Air (L/s)	Velocity (m/s)	Head Loss (Pa/m)	Flow Area (m ²)	Diameter (m)
Office-15	16.0	4.0	82.8	6.1	4.327	0.01358	0.1315
Office-14	16.0	4.0	82.8	6.1	4.327	0.01358	0.1315
Office-13	16.0	2.0	67.8	6.1	4.91	0.01112	0.119
Office-12	16.0	2.0	67.8	6.1	4.91	0.01112	0.119
Office-11	16.0	2.0	67.8	6.1	4.91	0.01112	0.119
Office-10	16.0	2.0	67.8	6.1	4.91	0.01112	0.119
Office-9	16.0	2.0	67.8	6.1	4.91	0.01112	0.119
Office-8	16.0	2.0	67.8	6.1	4.91	0.01112	0.119
Office-7	16.0	2.0	92.2	6.1	4.043	0.01511	0.1387
Office-6	16.0	1.0	56.6	6.1	5.505	0.0093	0.1087
Office-5	16.0	1.0	56.6	6.1	5.505	0.0093	0.1087
Office-4	16.0	1.0	56.6	6.1	5.505	0.0093	0.1087
Office-3	16.0	1.0	56.6	6.1	5.505	0.0093	0.1087
Office-2	16.0	1.0	56.6	6.1	5.505	0.0093	0.1087
Office-1	16.0	1.0	56.6	6.1	5.505	0.0093	0.1087
Meeting Room	48.0	20.0	290.7	4.1	1.971	0.0476	0.2463
Finance Office	24.0	3.0	118.9	6.1	3.445	0.01948	0.1575
corridor	90.0	20.0	187.9	6.1	2.587	0.03079	0.198

Return Main duct sizing

The minimum size of main return duct is 1 square inches per 3.5 CFM in other way CFM divided by 3.5 for area of return duct needed. The minimum size of filter in return duct is 1 square inches per 2 CFM, this means CFM divided by 2 for area of filter needed.

Table 4. 27: Main Return duct sizing for ground floor of the building.

Space Name	Maximum Return Air (L/s)	Velocity (m/s)	Head Loss (Pa/m)	Flow Area (m ²)	Diameter Of duct (m)	Velocity Pressure (Pa)
AHU to Clinic	16.3	7.6	21.47	0.00215	0.0523	33.72
Clinic to Toilet	15.14	7.6	22.53	0.002	0.0504	33.73
Toilet to Office-1	14.86	7.6	22.8	0.00196	0.0499	33.8
Office-(1-2)	14.55	7.6	22.11	0.0019	0.0494	33.74
Office-(2-3)	14.24	7.6	23.44	0.0019	0.0488	33.96
Office-(3-4)	13.93	7.6	23.78	0.00183	0.0483	33.86
Office-(4-5)	13.61	7.6	24.14	0.0018	0.0478	33.67
Office-(5-6)	13.3	7.6	24.51	0.00175	0.0472	33.83
Office-(6-7)	12.92	7.6	24.97	0.0017	0.0465	33.91
Office-(7-8)	12.36	7.6	25.7	0.0016	0.0455	33.84
Office-(8-9)	11.81	7.6	26.48	0.00156	0.0445	33.78
Office-(9-10)	11.27	7.6	27.3	0.0015	0.0435	33.68
Office-(10-11)	10.83	7.6	28.02	0.00142	0.0426	33.83
Office-(11-12)	10.35	7.6	28.86	0.0014	0.0416	33.97
Office-(12-13)	9.86	7.6	29.79	0.0013	0.0406	33.95
Office-(13-14)	9.34	7.6	30.86	0.00123	0.0396	33.66
Office-(14-15)	8.77	7.6	32.16	0.0012	0.0383	33.94
Office-(15-16)	8.16	7.6	33.72	0.0011	0.037	33.74
Office-(16-17)	7.5	7.6	35.64	0.00098	0.0354	34.02
Office-17 to Corridor	6.9	7.6	37.65	0.00091	0.034	33.82

Table 4. 28: Main Return duct sizing for first floor of the building.

Space Name	Maximum Return Air (L/s)	Velocity (m/s)	Head Loss (Pa/m)	Area of duct (m ²)	Diameter Of duct (m)	Velocity Pressure (Pa)
Office-(14-15)	11.1	7.6	27.569	0.00146	0.0431	33.9
Office-(13-14)	11.56	7.6	26.85	0.00152	0.044	33.83
Office-(12-13)	12	7.6	26.2	0.00158	0.0448	33.95
Office-(11-12)	12.4	7.6	26.65	0.00163	0.0456	33.77
Office-(10-11)	12.8	7.6	25.12	0.00168	0.0463	33.83
Finance Office to Office-10	13.14	7.6	24.39	0.00177	0.0474	33.76
Office-(8-9)	14.1	7.6	23.59	0.00186	0.0486	33.85
Office-(7-8)	14.38	7.6	23.29	0.00189	0.0491	33.79
Office-(6-7)	14.72	7.6	22.94	0.0019	0.0497	33.72
Office-(5-6)	15.1	7.6	22.56	0.00198	0.0503	33.82
Office-(4-5)	15.4	7.6	22.28	0.002	0.0508	33.8
Office-(3-4)	15.6	7.6	22.09	0.002	0.0511	33.88
Office-(2-3)	15.9	7.6	21.82	0.0021	0.0516	33.86
Office-(1-2)	16.1	7.6	21.65	0.00212	0.0519	33.9
Source to Office-1	16.3	7.6	21.47	0.00215	0.0523	33.72
Office-15 to Meeting Room	9.15	7.6	31.28	0.00121	0.0392	33.66
Office-9 to Finance Office	13.5	7.6	24.27	0.00178	0.0476	33.69
Meeting Room to corridor	7.6	7.6	35.33	0.001	0.0357	33.76

Return Branch duct sizing

Table 4. 29: Branch return duct sizing for ground floor of the building.

Space Name	Floor Area(m ²)	Maximum Occupants	Maximum Supply Air (L/s)	Velocity (m/s)	Head Loss (Pa/m)	Flow Area (m ²)	Diameter (m)
Clinic	72.0	3.0	6	4.1	8.63	0.0015	9.81
toilet	24.0	1.0	2.96	6.1	37.63	0.00049	21.63
Office-1	16.0	1.0	2.97	6.1	37.63	0.00049	21.63
Office-2	16.0	1.0	2.97	6.1	37.63	0.00049	21.63
Office-3	16.0	1.0	2.97	6.1	37.63	0.00049	21.63
Office-4	16.0	1.0	2.97	6.1	37.63	0.00049	21.63
Office-5	16.0	1.0	2.97	6.1	37.63	0.00049	21.63
Office-6	16.0	1.0	2.97	6.1	37.63	0.00049	21.63
Office-7	16.0	1.0	2.97	6.1	37.63	0.00049	21.63
Office-8	20.0	3.0	3.84	6.1	31.64	0.00063	21.83
Office-9	16.0	1.0	3.6	6.1	33.03	0.00059	21.8
Office-10	20.0	5.0	3.49	6.1	33.72	0.00057	21.73
Office-11	20.0	3.0	3.2	6.1	35.72	0.00052	21.92
Office-12	20.0	3.0	3.2	6.1	35.72	0.00052	21.92
Office-13	20.0	3.0	3.2	6.1	35.72	0.00052	21.92
Office-14	20.0	3.0	3.2	6.1	35.72	0.00052	21.92
Office-15	20.0	3.0	3.2	6.1	35.72	0.00052	21.92
Office-16	20.0	3.0	3.2	6.1	35.72	0.00052	21.92
Office-17	20.0	3.0	3.2	6.1	35.72	0.00052	21.92
Corridor	9.0	2.0	2.9	6.1	38.14	0.00048	21.83

Table 4. 30: Branch return duct sizing for first floor of the building.

Space Name	Floor Area (m ²)	Maximum Occupants	Maximum Supply Air (L/s)	Velocity (m/s)	Head Loss (Pa/m)	Flow Area (m ²)	Diameter (m)
Office-15	16.0	4.0	3.34	6.1	34.72	0.00055	21.83
Office-14	16.0	4.0	3.34	6.1	34.72	0.00055	21.83
Office-13	16.0	2.0	3	6.1	37.29	0.00049	21.86
Office-12	16.0	2.0	3	6.1	37.29	0.00049	21.86
Office-11	16.0	2.0	3	6.1	37.29	0.00049	21.86
Office-10	16.0	2.0	3	6.1	37.29	0.00049	21.86
Office-9	16.0	2.0	3	6.1	37.29	0.00049	21.86
Office-8	16.0	2.0	3	6.1	37.29	0.00049	21.86
Office-7	16.0	2.0	3	6.1	37.29	0.00049	21.86
Office-6	16.0	1.0	2.76	6.1	39.43	0.00045	21.84
Office-5	16.0	1.0	2.76	6.1	39.43	0.00045	21.84
Office-4	16.0	1.0	2.76	6.1	39.43	0.00045	21.84
Office-3	16.0	1.0	2.76	6.1	39.43	0.00045	21.84
Office-2	16.0	1.0	2.76	6.1	39.43	0.00045	21.84
Office-1	16.0	1.0	2.76	6.1	39.43	0.00045	21.84
Meeting Room	48.0	20.0	6.26	4.1	8.39	0.00153	9.84
Finance Office	24.0	3.0	4	6.1	30.8	0.00066	21.77
corridor	90.0	20.0	3.34	6.1	34.72	0.00055	21.83

Duct Diffuser design

A device for reducing the velocity and increasing the static pressure of a fluid passing through a system. The fluid's static pressure rise as it passes through a duct is commonly referred to as pressure recovery. Proper selection of air diffusion devices requires basic knowledge of the mechanics of room air distribution. During design or selection of the existing diffuser the following point should be considered.

- ✓ **Primary Air:** Primary air is defined as the conditioned air discharged by the supply outlet. This air provides the motive force for room air motion.
- ✓ **Total Air:** Total air is defined as the mixture of primary air and entrained room air which is under the influence of supply outlet conditions. This is commonly considered to be the air within an envelope of 50 fpm [0.25 m/s] (or greater) velocity. The temperature difference between the total air and the room air creates buoyant effects which cause cold supply air to drop and warm air to rise.
- ✓ **Throw:** Throw is the distance from the center of the outlet face to a point where the velocity of the air stream is reduced to a specified velocity. This velocity referred to as terminal velocity. Throw is primarily a function of mass flow and outlet velocity and therefore can be reduced by decreasing either of these values.
- ✓ **Drop:** The drop of cool total air is the result of vertical spread of the air stream due to entrainment of room air and the buoyancy effect due to the density differences between the total air package and the surrounding primary room air. The term density is very important as drop is primarily dependent upon the mass flow of the total air. Drop can be minimized by spreading air uniformly over the ceiling surface, thus reducing the mass flow per unit surface area.
- ✓ **Spread:** The spread of an outlet is defined as the divergence of the air stream in a horizontal or vertical plane and is a function of the outlet geometry.
- ✓ **Surface Effect:** Drop can also be effectively reduced by use of the surrounding ceiling surface. When supply air velocity is sufficiently high, a negative or low pressure area is created between the moving air mass and the ceiling at or near the supply air outlet. This low pressure area causes the moving air mass to cling to and flow close to the ceiling surface. This principle is known as the Coanda effect.

- ✓ **Stratification:** Mixing ventilation systems generally supply air in a manner such that the entire room volume is fully mixed. The cool supply air exits the outlet at a high velocity, inducing room air to provide mixing and temperature equalization. Since the entire room is fully mixed, temperature variations throughout the space are small. This variation in room air temperature from floor to ceiling is known as stratification.
- ✓ **Return:** The return air inlet has very little effect on room air diffusion, regardless of inlet type or location. However, return air inlets should be located a sufficient distance from the supply outlet so that short-circuiting of supply air does not occur. It may also be desirable to locate the returns in the stagnant zone to remove unwanted warm or cool air. For cooling, a high sidewall or ceiling return will remove warm air from the space. For heating a low sidewall return will remove warm stagnant air.
- ✓ **Convection Currents:** The total air package can easily be influenced by several factors within the space. One of these factors that occurs in exterior zones of buildings is the natural convection currents resulting from a hot outside wall during cooling or a cold outside wall during heating. The upward movement of air in the vicinity of the hot surface tends to oppose the total air movement in overhead cooling. This can act to reduce the outlet throw values or even cause the colder total air to leave the ceiling and create drop into the space. The downward movement of cold air in the vicinity of a cold surface can create cold drafts within the occupied space. In the case of overhead heating, the only effective way to minimize these drafts is to direct a high velocity jet of warm air over the wall surface to reduce the difference between the temperature of the surface and that of the room air.

The size of the diffuser for each supply and return duct will be calculated by:

$$\text{Diffuser size} = \sqrt{\frac{\text{CFM}}{400}} * 144 \quad (4.32)$$

The unit of diffuser size from this calculation was inch it should be converted to mm, cm or m.

Supply duct diffuser sizing

Table 4. 31: Supply duct diffuser sizing result of ground floor of building.

Space Name	Floor Area(m ²)	Maximum Occupants	Maximum Supply Air (L/s)	Velocity (m/s)	Head Loss (Pa/m)	Flow Area (m ²)	Diameter (m)
Clinic	72.0	3.0	96.6	4.1	1.456	0.024	0.1732
toilet	24.0	1.0	23.5	6.1	9.7	0.039	0.07
Office-1	16.0	1.0	23.5	6.1	9.7	0.039	0.07
Office-2	16.0	1.0	23.5	6.1	9.7	0.039	0.07
Office-3	16.0	1.0	23.5	6.1	9.7	0.039	0.07
Office-4	16.0	1.0	23.5	6.1	9.7	0.039	0.07
Office-5	16.0	1.0	23.5	6.1	9.7	0.039	0.07
Office-6	16.0	1.0	23.5	6.1	9.7	0.039	0.07
Office-7	16.0	1.0	23.5	6.1	9.7	0.039	0.07
Office-8	20.0	3.0	39.4	6.1	6.97	0.0065	0.091
Office-9	16.0	1.0	35.35	6.1	7.5	0.0058	0.0859
Office-10	20.0	5.0	32.44	6.1	7.89	0.0053	0.0823
Office-11	20.0	3.0	27.04	6.1	8.86	0.0044	0.0751
Office-12	20.0	3.0	27.04	6.1	8.86	0.0044	0.0751
Office-13	20.0	3.0	27.04	6.1	8.86	0.0044	0.0751
Office-14	20.0	3.0	27.04	6.1	8.86	0.0044	0.0751
Office-15	20.0	3.0	27.04	6.1	8.86	0.0044	0.0751
Office-16	20.0	3.0	27.04	6.1	8.86	0.0044	0.0751
Office-17	20.0	3.0	27.04	6.1	8.86	0.0044	0.0751
Corridor	9.0	2.0	21.85	6.1	10.163	0.0036	0.0675

Table 4. 32: Supply duct diffuser sizing result of first floor of building.

Space Name	Floor Area (m ²)	Maximum Occupants	Maximum Supply Air (L/s)	Velocity (m/s)	Head Loss (Pa/m)	Flow Area (m ²)	Diameter (m)
Office-15	16.0	4.0	29.81	6.1	8.33	0.0049	0.0789
Office-14	16.0	4.0	29.81	6.1	8.33	0.0049	0.0789
Office-13	16.0	2.0	24.41	6.1	9.5	0.004	0.0714
Office-12	16.0	2.0	24.41	6.1	9.5	0.004	0.0714
Office-11	16.0	2.0	24.41	6.1	9.5	0.004	0.0714
Office-10	16.0	2.0	24.41	6.1	9.5	0.004	0.0714
Office-9	16.0	2.0	24.41	6.1	9.5	0.004	0.0714
Office-8	16.0	2.0	24.41	6.1	9.5	0.004	0.0714
Office-7	16.0	2.0	33.2	6.1	7.77	0.005	0.0832
Office-6	16.0	1.0	20.4	6.1	10.62	0.003	0.0653
Office-5	16.0	1.0	20.4	6.1	10.62	0.003	0.0653
Office-4	16.0	1.0	20.4	6.1	10.62	0.003	0.0653
Office-3	16.0	1.0	20.4	6.1	10.62	0.003	0.0653
Office-2	16.0	1.0	20.4	6.1	10.62	0.003	0.0653
Office-1	16.0	1.0	20.4	6.1	10.62	0.003	0.0653
Meeting Room	48.0	20.0	104.7	4.1	1.38	0.026	0.1803
Finance Office	24.0	3.0	42.8	6.1	6.61	0.007	0.0945
corridor	90.0	20.0	67.6	6.1	4.94	0.011	0.1188

Temperature Loss in Ducts

The heat loss from a duct can be calculated as:

$$H = A k ((t_1 + t_2) / 2 - t_r) \quad (4.33)$$

Where:

H = heat loss (W)

A = area of duct walls (m²)

t₁ = initial temperature in duct (°C)

t₂ = final temperature in duct (°C)

k = heat loss coefficient of duct walls (W/m² K) (5.68 W/m² K for sheet metal ducts, 2.3 W/m² K for insulated ducts)

t_r = surrounding room temperature (°C)

The heat loss in the air flow can be expressed as:

$$H = 1000 q c_p (t_1 - t_2) \quad (4.34)$$

$$\rho = \frac{P}{R_{specific} T} \quad (4.35)$$

ρ = density of air

P = absolute pressure

$R_{specific}$ = specific gas constant

T = absolute temperature

$$T = \frac{P}{\rho \cdot R_{specific}} \quad (4.36)$$

Table 4. 33: Temperature and heat loose in the duct where:

Ground floor	First floor
Temperature loose	
$t_1 = 16^{\circ}C$ $t_2 = 13^{\circ}C$ $t_r = 24.3^{\circ}C$ Temperature loose in duct where the difference Between t_1 and t_2 . $T_l = t_1 - t_2$ $T_l = 16^{\circ}C - 13^{\circ}C$ $T_l = 3^{\circ}C$	$t_1 = 16^{\circ}C$ $t_2 = 12.3^{\circ}C$ $t_r = 24.3^{\circ}C$ Temperature loose in duct where the difference Between t_1 and t_2 . $T_l = t_1 - t_2$ $T_l = 16^{\circ}C - 12.5^{\circ}C$ $T_l = 3.5^{\circ}C$
Mass flow rate	
$\rho = 1.225 \text{ kg} / \text{m}^3$ $Q_1 = 1.9694 \text{ m}^3 / \text{s}$ $q = Q * \rho$ $q = 1.9694 \text{ m}^3 / \text{s} * 1.225 \text{ kg} / \text{m}^3$ $q = 2.41 \text{ kg} / \text{s}$	$\rho = 1.225 \text{ kg} / \text{m}^3$ $Q_2 = 1.9778 \text{ m}^3 / \text{s}$ $q = Q_2 * \rho$ $q = 1.9778 \text{ m}^3 / \text{s} * 1.225 \text{ kg} / \text{m}^3$ $q = 2.423 \text{ kg} / \text{s}$
Heat loose	
$C_p = 1.00 \text{ KJ} / \text{Kg}$ $H = 1000 q c_p (t_1 - t_2)$ $H = 1000 * 2.41 * 1 * 3$ $H = 7230 \text{ W} = 7.23 \text{ kW}$	$C_p = 1.00 \text{ KJ} / \text{Kg}$ $H = 1000 q c_p (t_1 - t_2)$ $H = 1000 * 2.423 * 1 * 3.5$ $H = 8512 \text{ W}$ $H = 8.512 \text{ kW}$

Return fan and motor selection

Total amounts of air needed to be sucked out per second was 203842.8 lpm (203.8428 m³/hr) which is 120 cfm, at this volume flow rate was selected from standard table:

- ✓ Motor power 1.7 hp/1000 cfm
- ✓ Fan size 315"

4.6 Design of liquid based air filtration System

Batu caustic soda and cement factory office building requires 3947.2 l/s or 8368.064 CFM amount of filtered air for each room on the building, occupancy in building was 80 people and the conditioned air needed to maintain indoor condition at 21 °C and relative humidity 60%. As the volume flow rate was pre-determined from standard ASHRAE hand book 1997 or 2014. The volume flow rate was between 8000 cfm and 9000 cfm, but it is most nearer to 12000 cfm.

Condition of air interring in to evaporative cooler.

- ✓ Dry Bulb temperature of interring air – 24.6°C
- ✓ Wet Bulb temperature of interring air – 17.5°C
- ✓ Dry Bulb temperature of air leaving supply fan – 16°C
- ✓ Specific humidity of interring air – 0.009 kg/kg
- ✓ Relative Humidity of interring air – 31%

$$e_c = \left[\frac{t_1 - t_2}{t_1 - t_3} \right] * 100 \quad (4.37)$$

Where:

e_c = evaporative cooling effectiveness or saturation effectiveness %

t_1 = dry bulb temperature of entering air

t_2 = dry bulb temperature of leaving air

t_3 = wet bulb temperature of entering air

$$e_c = \left[\frac{24.6 - 17.5}{24.6 - 16} \right] * 100$$

$$e_c = 82.56\%$$

The evaporative cooling is able to cool air to the temperature below and the dew point effectiveness spans 0.81 – 0.93, when the inlet air temperature was varied from 23 to 40°C and the humidity ratio was varied from 8 to 20 g kg⁻¹ (G Khater, 2014), mass flow rate of inlet air was:

$$m_a = \rho_a * Q_a = 1.225 \text{ kg} / \text{m}^3 * 3.9472 \text{ m}^3 / \text{s} = 4.84 \text{ kg} / \text{s}$$

Temperature drop achievable = (dry bulb – wet bulb) x (efficiency of the media)

Temperature drop achievable = (24.6 – 17.5) °C x (0.856)

Temperature drop achievable = 5.893 °C

Achievable temperature = dry bulb – temp drop achievable

Achievable temperature = 24.6 °C – 5.893 °C

Achievable temperature = 18.71 °C

From standard table at 24.6 °C dry bulb and 17.5 °C wet bulb outdoor air temperature it is possible to estimate the indoor temperature and humidity.

Condition of air living evaporative cooler.

- ✓ Dry Bulb temperature – 16 °C
- ✓ Wet Bulb temperature – 14.1 °C
- ✓ Specific humidity – 0.0118 kg/kg
- ✓ Relative Humidity – 53 %

Sizing the evaporative cooling equipment

Table 4. 34: Size of evaporative cooling equipment (ASHRAE hand book 2007)

Air flow			Dimension			
Flow Rate (cfm)	Minimum Flow rate (cfm)	maximum Flow rate (cfm)	Length (mm)	Maximum Length (mm)	Width (mm)	Height (mm)
9000	7600	10300	2900	3050	2225	1475

Table 4. 35: Size of evaporative cooling equipment (ASHRAE hand book 2007)

Flow Rate (cfm)	Base height (mm)	Outlet size (mm)		Capacity of Pump (kW)	Water requirement (lpm)	Mass flow rate (kg/s)
		Height	Width			
9000	100	714	714	0.25	89	875

Cubic Capacity of the structure or the portion of the structure to be evaporative cooled.

Cubic Capacity = Width X Length X Effective Cooling Height

Cubic Capacity = 2.225m 2.9m *1.475m*

Cubic Capacity =9.52m³

The leaving dry bulb temperature in case of direct spray of water over the air calculated as:

$$LDB = ODB - [e_c \times (ODB - OWB)] \quad (4.36)$$

Where

LDB is the leaving dry bulb temperature

ODB is the outdoor dry bulb temperature = 24.6°C

OWB is the outdoor wet bulb temperature =17.5°C

(ODB – OWB) is the wet bulb depression

wet bulb depression = ODB – OWB

wet bulb depression = 24.6°C –17.5°C

wet bulb depression = 7.1°C

LDB = 24.6°C – [0.8256 x (7.1°C)]

LDB = 18.74°C

The leaving dry bulb temperature in case of using cooling pad of 12” calculated as:

LDB = 24.6°C – [0.9 x (7.1°C)]

LDB = 18.2°C

Once the above two factors have been determined, by referring to table of air change to determine the proper number of air changes. LDB = 18.74 °C = 65.732 °F, leaving dry bulb temperature where less than 72 °F, at this temperature over ambient will be less than 72 °F and air change per hour was 10-15 1/hr, the average will 12.5 1/hr, in this case 13 1/hr.

The optimum time that needed to reach for the suitable temperature of air that exit from air cooler calculated from this relationship

Saturation Efficiency and Make up Water Requirements

$$e_c = \left[\frac{t_1 - t_2}{t_1 - t^1} \right] * 100$$

Where :

e_c = evaporative cooling effectiveness. %

t_1 = dry bulb temperature of entering air

t_2 = dry bulb temperature of leaving air

t^1 = thermodynamic wet bulb temperature of entering air

$$t_1 = 24.6^\circ\text{C}$$

$$t_2 = 17.5^\circ\text{C}$$

$$t^1 = 16.6^\circ\text{C}$$

$$e_c = \left[\frac{t_1 - t_2}{t_1 - t^1} \right] * 100$$

$$e_c = \left[\frac{24.6^\circ\text{C} - 17.5^\circ\text{C}}{24.6^\circ\text{C} - 16.6^\circ\text{C}} \right] * 100$$

$$e_c = 88.75\%$$

The evaporation rate is given by equation:

$$\text{evaporation rate (GPH)} = \frac{\text{Air flow rate in CFM} \times (T_i - T_{wb}) \times \text{Saturation Efficiency}}{8700} \quad (4.37)$$

$$\text{Air flow rate in CFM} = 8368.064 \text{ cfm}$$

$$T_i = 24.6^\circ\text{C}$$

$$T_{wb} = 17.5^\circ\text{C}$$

$$\text{Saturation Efficiency} = 0.8256$$

$$\text{GPH} = \frac{8368.064 \text{ cfm} \times (24.6^\circ\text{C} - 17.5^\circ\text{C}) \times 0.8256}{8700}$$

$$\text{GPH} = 5.64 \text{ lbs /pound} = 5.64 * 1.3558 = 7.65 \text{ Nm}$$

The moisture absorbed by air or the evaporation rate can also be calculated by:

$$\text{moisture absorbed by air} = w_o - w_i$$

$$\text{moisture absorbed by air} = 0.0118 \text{ kg / kg} - 0.009 \text{ kg / kg}$$

$$\text{moisture absorbed by air} = 0.0028 \text{ kg / kg}$$

From Standard table it is possible to determine the evaporative rate, from above calculation the wet bulb depression was 6.8 °C.

Evaporative rate value from above table at wet bulb depression was 6.8 °C approximately 7 °C and saturation efficiency 0.8256 is 2.04 Gallons per Hour evaporated per 1000 CFM. In case of this research study 8368.064 CFM.

$$\begin{aligned} \text{Evaporative rate} &= 2.04 * 8368.064 \text{ CFM} / 1000 \\ \text{Evaporative rate} &= 17.071 \text{ Gallons per Hour evaporated per 1000 CFM} \end{aligned} \quad (4.38)$$

The bleed off rate is given by:

$$\begin{aligned} \text{Bleed - Off Rate} &= \text{Evaporation Rate} \times 0.20 \text{ (...approximately 20\% of evaporation rate)} \\ \text{Bleed - Off Rate} &= 17.071 \text{ Gallons per Hour evaporated per 1000 CFM} * 0.2 \\ \text{Bleed - Off Rate} &= 3.414 \text{ Gallons per Hour evaporated per 1000 CFM} \end{aligned}$$

The water flow rate calculated as:

$$\begin{aligned} (\text{Recirculation}) \text{ Water flow Rate} &= 3 \text{ times the evaporation rate} \\ (\text{Recirculation}) \text{ Water flow Rate} &= 3 * 17.071 \text{ Gallons per Hour evaporated per 1000 CFM} \\ (\text{Recirculation}) \text{ Water flow Rate} &= 51.213 \text{ Gallons per Hour evaporated per 1000 CFM} \end{aligned}$$

The dry bulb temperature drop across the evaporative cooling pad or cooling media thickness will be calculated from flowing ASHRAE standard table, the thickness of cooling pad was 12 inch.

There is about 5.893 °C temperature drop across the cooling pad.

Sensible heat transfer during the process:

$$\begin{aligned} \text{Sensible heat} &= \text{CFM} * 1.08 * [IDB - LDB] \\ \text{Sensible heat} &= 8368.064 * 1.08 * [24.6^\circ\text{C} - 17.5^\circ\text{C}] \\ \text{Sensible heat} &= 64166.3 \text{ BTU} \\ \text{Sensible heat} &= 18.8 \text{ kW} \end{aligned}$$

The atmospheric pressure at wet bulb and dry bulb living air was 101289 Pa.

Face area where calculated as:

$$\begin{aligned} \text{Face Area} &= \text{Width} \times \text{Height of open face area through which air will flow (expressed in square feet.)} \\ \text{Face Area} &= 2.225 \text{ m} * 1.475 \text{ m} \\ \text{Face Area} &= 3.282 \text{ m}^2 = 35.33 \text{ ft}^2 \end{aligned}$$

Velocity of cooled air on the living face of the evaporative cooler was:

$$\text{Face Velocity} = \text{CFM} / \text{Face Area (Sq Ft)} \text{ (expressed in Feet per minute (FPM))}$$

$$\text{Face Velocity} = 8368.064 \text{ cfm} / 35.33 \text{ ft}^2$$

$$\text{Face Velocity} = 236.85 \text{ fpm} = 1.2 \text{ m/s}$$

The percent media efficiency and static pressure drop at given media depth will be calculated from ASHRAE standard table 2007, by using value of face velocity and cooling pad depth.

Percent Media Efficiency at Media Depth: 96%

Static Pressure Drop at Media Depth: 0.10 inch

Air density ratio can estimated from ASHRAE standard table by using the value of height in fit and outdoor dry bulb temperature Air density ratio: 0.82.

The recommended air change rates for direct evaporative cooling depend on living wet-bulb temperature conditions are given in table below.

Table 4. 36: Air change standard data.

Wet bulb temperature (°C)	Minimum air change rate (minutes)	Maximum air change rate (minutes)	Minimum air change per hour (AC/hr)	Maximum air change per hour (AC/hr)
15.6	5	2	12	30
17	4.5	2	13.3	30
18	4.3	2	11.1	30
19	4	2	15	30
20	3.8	2	16	30
21	3.5	2	17.1	30

Supply air pressure drop and thickness of efficiency cell may determine from the saturation effectiveness and face velocity graph.

From direct evaporative cooling performance chart at face velocity 300 fpm and saturation effectiveness 83%.

- ✓ Efficiency cell = 0.32 inch
- ✓ Supply air pressure drop = 0.33 in.W.C

Fan and motor selection

Fan size, capacity and motor selection for direct evaporative cooling system, from standard table.

- ✓ Blower: 7.5 HP, 2850 RPM.
- ✓ Blower size: 27"
- ✓ Blower efficiency grade: 85%
- ✓ Quantity of blower: 1
- ✓ Pump capacity: 0.25 kW
- ✓ Pipe sizing: 0.25"-1"
- ✓ Motor power was 0.5 HP, 1200 RPM
- ✓ Quantity of motor: 1

The volume of water storage was calculated depending on the amount of water volume flow rate of water from tank storage to pad for cooling, the flow rate of water required to cool 5.615 m³/s of air for ventilation was 89 lpm (0.089 m³/m), so depending on this flow rate value from standard table the length, width and height of storage tank was 4m, 3m, 2m. Its volume was 24 m³.

Pipe dimension

Connections from water tank to pump and from pump to nozzles are made with help of plastic pipes and joint. Depending on ASHRAE standard hand book 1997, the diameter of the pipe could be selected from table depending on the flow rate of the water required for air filtering.

The total length of pipe required was 10m, with flow rate of 660 m³/s from standard table the nominal diameter of the pipe where 12 cm, from all above value it is possible to calculate the pressure.

$$Q = (1946.4 * D^{1.857}) * \left(\frac{P}{l}\right)^{0.54} \quad (4.39)$$

$$Q = (1946.4 * D^{1.857}) * \left(\frac{P}{l}\right)^{0.54}$$

$$660 = (1946.4 * 0.12^{1.857}) * \left(\frac{P}{10}\right)^{0.54}$$

$$660 / 38 = \frac{P^{0.54}}{10^{0.54}}$$

$$P^{0.54} = 17.37 * 3.47 = 60.23$$

$$P = (60.23)^{1/0.54} = (60.23)^{1.852} = 1977.83 Pa = 1.97783 KPa$$

CHAPTER FIVE

EXPERIMENTAL SETUP AND PROCEDURE OF LIQUID BASED AIR FILTRATION SYSTEM (DIRECT EVAPORATIVE COOLING)

5.1 Introduction

This chapter presents the conceptual model, working principle, manufacturing process, experimental setup and procedure followed while testing the prototype of an Central liquid based air filtration system. The conceptual model of the system was developed using Solid Work 2015. The working principle of the system is explained in detail. Manufacturing process of each component for preparing the prototype is presented with descriptive photos. Finally, the measuring and testing devices used are presented.

5.1.1 Conceptual Model

The schematic diagram of Central liquid based air filtration (direct evaporative cooling) drawn using Solid Work 2015. The system consists of an Air filtering chamber, Water storage tank, cooling pad, water distribution header, water filter tank, pump, motor and Air exhaust, closed-loop piping network as a connection for the whole system, blower, frame, and support. Figure 5.1 illustrates the schematic diagram of Liquid Based Air Filtration System.

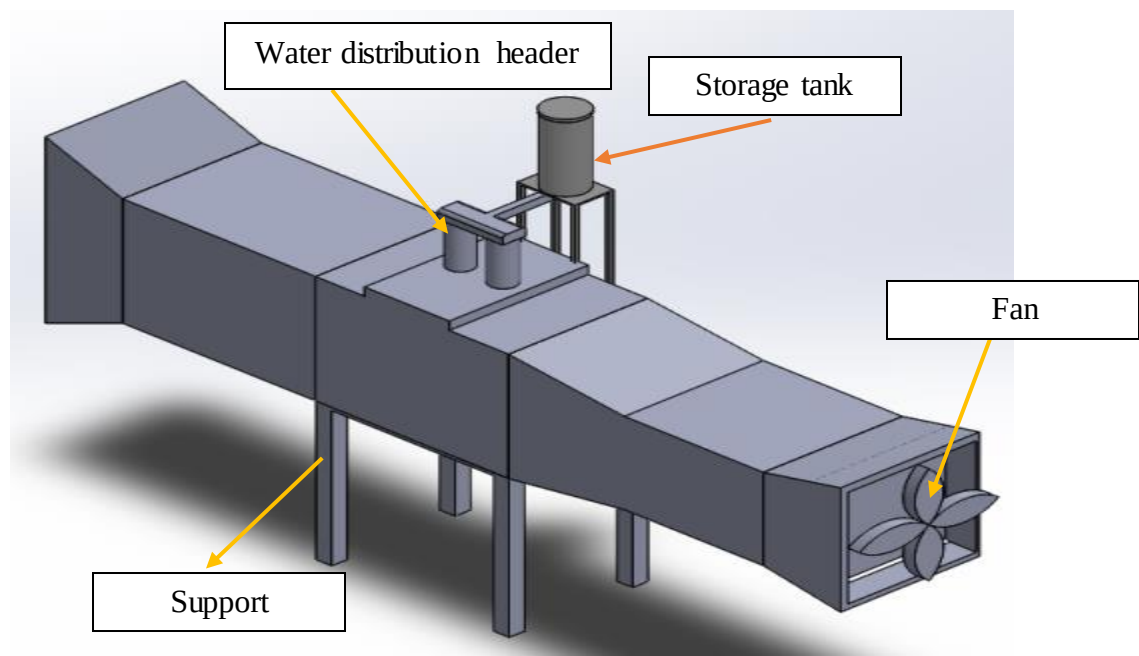


Figure 5. 1: Conceptual schematic of a Liquid Based Air Filtration System

5.3 Working Principle

The developed Liquid Based Air Filtration System consists of an air filtering chamber with cooling pad, water distribution header, a blower, air inlet and outlet as illustrated in Figure 5.2. In this system, a blower will suck air in to the system and bring it to the cooling pad, where air is make contact with liquid water. As air pass through cooling pad water distribution header release water over cooling pad, the water receives heat and pollution from atmospheric air and send it to exit water filtering tank where unnecessary particle inside water from cooling chamber where filtered. The air will pass through four 2 mm thick cooling pad to make sure that the air cooled and humidified to the required extent. During the process dry bulb temperature where decreased, humidity increased and pollution content in the air decreased.

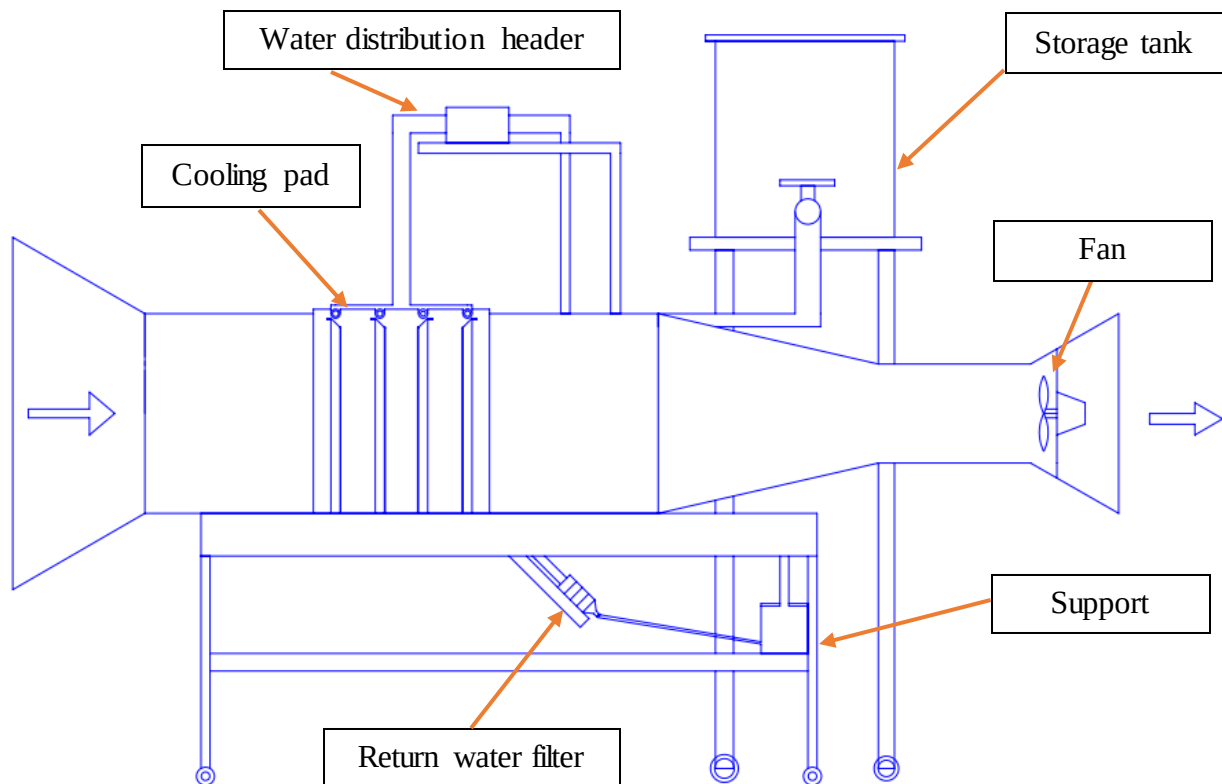


Figure 5. 2: Schematic layout of the Liquid Based Air Filtration System.

5.4 Manufacturing Process

The components of the Liquid Based Air Filtration System, cooling chamber, cooling pad, water storage tank, water distribution header, water filter tank, pump, motor and Air exhaust, closed-loop piping network as a connection for the whole system, blower, frame, and support. Each step used to construct components of the Air Filtration and its assembly is shown, described, and discussed in this section as follow.

5.4.1 Construction of the cooling chamber

It is a box made of Galvanized steel sheet in which air filtration (direct evaporative cooling) process takes place. The dimensions of the chamber was, length = 3.05 m, width = 2.225 m, h = 1.475 m as per design numerical analysis. But in case of experimental design the dimension of chamber was, length = 0.76 m, width = 0.5 m, h = 0.5 m. It should be such that it can accommodate the fan for drawing air. During air filtering or cleaning process it should be sure that all air passes through the chamber without leakage, but in case if there is return air on the chamber bottom right hand there is return air hole. The provision also provide for the accommodation of the nozzles, for water exit provision also provide at bottom of chamber so that water can be collected and its temperature can be measured, the main requirement of chamber is to provide leak proof passages to achieve the desirable results.



Figure 5. 3: Air filtration chamber

5.4.2 Water Tank

Water tank is located below the air filter chamber. It is connected to the water pipe from suitable source for continuous supply of the water. The central air filter needs 0.66 L quantity of the water per each second and 57024 L per day. By considering the water volume required for one hour, which is 2376 L, the dimension of water tank will be (1.5 x 1.5 x 1.5) m, length, width and height. With this dimension the water tank can store 3375 L of water, more than water needed for one hour. Due to the water continuous circulation water lack was not mater. But in case of this research study the dimension of water tank was (0.5 x 0.5 x 0.25) m³ can store 62.5 L of water. The water in used in this experiment was ordinary water or chilled water.

The Water tank supporter should withstand loads like wind load, stress loads, etc., so it was a rigid body in this experimental design, for this work, flat galvanized steel is used. For strength of tank the internal reinforcement was welded from steel rectangular metal, the bottom surface of the tank where conical shape for easily remove of water during cleaning through drainage. One side of the storage tank where has a hole for water pipe connected to water distributor in air filter chamber.

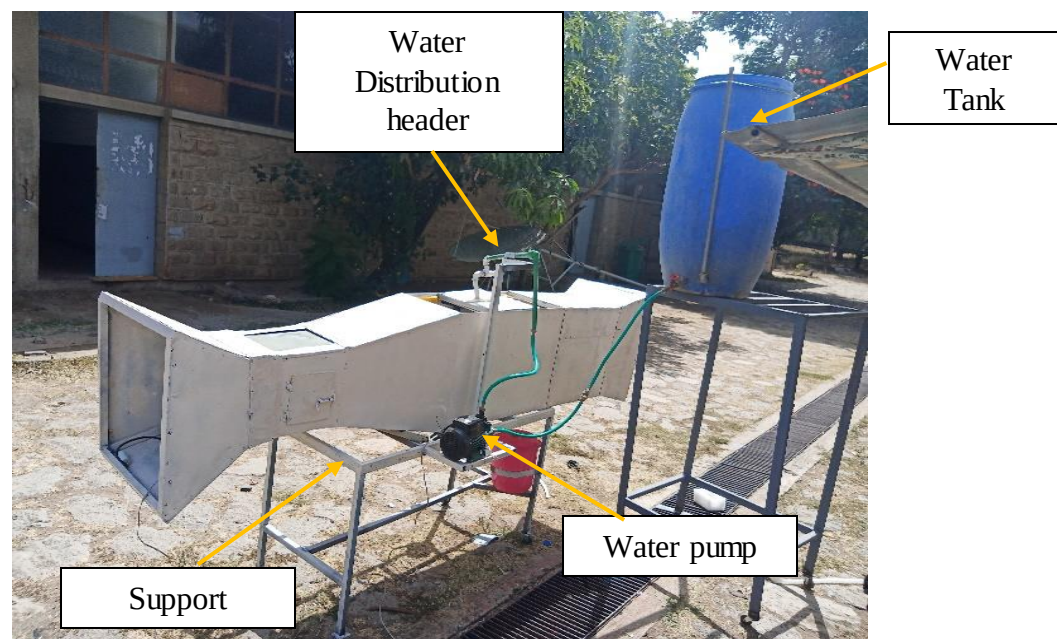


Figure 5. 4: Water tank.

5.4.3 Cooling pad

Cooling pad is used for the purification purpose of the air washer. In this study cellulose pads, nylon filter where used, but in this model nylon filter is used which is fixed in a panel frame of air filter chamber. The size of cooling pad where 0.5 x 0.5 m with thickness of 2 mm.



Figure 5. 5: Cooling pad

5.4.4 Return water cleaner

The filter water from air cleaning chamber where recycled for re use purpose, so before recycling back to the water tank it will filtered and return back to water tank by gravity.



Figure 5. 6: Return water cleaner

5.4.3 Construction of Supporter Structure

The supporting structure were prepared using welding process to hold the casing of the air filtering system, It provides the rigidity and with help of small four while possible to move from place to place. The air filter is mounted on a 2 m × 1m frame constructed from square metal pipe. The whole supporting structure is painted varnish to improve its life as it always remains in the open environment.

5.4.4 Pipes and fittings

Connections from water tank to pump and from pump to spray drain are made with help of plastic pipes and joint, For desirable experimental result during process there must be no leakage in connection so that no loss in flow rate and in pressure. For different types of arrangement of setup in this research study two types of joint used, they are T joints, right angle joint (elbow). Diameter of the pipe for experiment was 2.54cm nominal diameter and 3m length. The pressure change in pipe was 0.00432 Pa and it was most negligible, from this it is possible to conclude that the loss in the pipe were negligible. It is also important factor that the flow rate of water is depended on diameter of pipes, for high flow rates diameter of the pipes should be large. In this research study purifier pipes and joints Where used the for the experiment setup. Connections are made from water tank to centrifugal pump and then from pump to spray drain provided in chamber via plastic pipes and joints.



Figure 5. 7: Fitting and pipes

5.4.5 Water distribution header

Water from the water tank is pumped to the water distribution header via plastic or mild steel piping. The spray nozzle sprays the water inside the air filter chamber. The nozzle should produce finely atomized particles so that the entire chamber is filled with the mist. In order to break the water into fine droplets, the nozzle should have low coefficient of discharge. The spraying nozzles are made of brass, full cone type. They are responsible for supplying water to the equipment wash system. The pressure is calculated so that the nozzles provide a perfect cone to cover 100% of the area of the chamber.



Figure 5. 8: Water distribution header

5.4.6 Fan

Fan is used to draw the air out the chamber so that supply of conditioned air is possible. For carrying out the performance analysis and find out the effect of air flow rate on process, speed of the fan changed through the regulator.



Figure 5. 9: Fan

5.4.7 Pump

Pump is required for supplying water from tank to the spray drain with pressure. Capacity of pump depends on the height of the system, such that it can pump the water from the tank to the desire height without pressure loss. Location of the pump is also important as per the performance aspects. The height where the pump is located should be such that it would provide positive head. In case of this research study centrifugal pump is used, in experimental procedure small centrifugal pump where used as the flow rate and capacity of setup is small.



Figure 5. 10: Pump

5.5 Experimental setup

A direct central liquid based air filtration system was designed, developed, and installed at Adama Science and Technology University, Adama, Ethiopia. Photograph of the complete experimental setup of the direct central liquid based air filtration system used for the present investigation is shown in Figure 5.8.



Figure 5. 11: Experimental setup of the central liquid based air filtration system.

The experimental setup consists of four foremost parts, the first was diverged air inlet portion where inlet air velocity, pressure, temperature, humidity and air quality measured, the second where air filtering (washing) part, where inlet polluted air contacted with water and polluting particle and compound where washed by water, the other was clean air exit part, in this section clean air where characterized and the last section was water cleaner and circulating section, in this section the water returned from cooling chamber where filtered and flow back to storage tank. The air where sucked by blower which were placed in clean air exit portion.

5.6 Experimental Procedure

The experimental study was carried out for two different conditions: i. cleaning present air and characterizing it ii. By creating false pollution clean and characterizing air quality. The first experiment is done on cleaning and characterizing the present air in atmosphere. For this digital thermometer and pitot tube were placed at the air inlet portion to measure the change of atmospheric temperature, Pitot tube to measure inlet air velocity and pressure and air quality detector for clean air characterization, these sensor where placed in the clean air exit section to detect air quality. This experiment where cared out to measure and reduce the concentration of pollutant in present air. The second experiment was conducted to investigate the performance of central liquid based air filtration system. In this experiment falls pollution where created by burning woods and other material to create smoke and concentration of pollution where measured by air quality detector and it late to be sucked by blower in to air cooling chamber for cleaning, then after the same characterizing process where followed as first condition. In each experiment, the inlet and outlet air property was recorded every 1 minute interval.



Figure 5. 12: False pollution creation.

5.7 Measuring Instruments

The temperature during the experiments were measured using digital thermometer, air quality where evaluated by air quality sensor and pitot tube for air velocity and air pressure evaluation based on the procedure described in APPENDIX, at inlet and outlet section of the central liquid based air filtration. The digital thermometer can read values between $-50\text{ }^{\circ}\text{C}$ to $300\text{ }^{\circ}\text{C}$ with an error of ± 1 , air quality detector where read temperature value between $-10\text{ }^{\circ}\text{C}$ to $50\text{ }^{\circ}\text{C}$ with an error of $\pm 1\text{ }^{\circ}\text{C}$, CO_2 value 400 ppm to 6000 ppm with an error of ± 1 ppm, TVOC detection range 0 to 2 mg/m^3 , HCHO detection range 0 to 0.6 mg/m^3 and humidity measurement range 0% to 99% with accuracy of $\pm 2\%$, pitot tube measure air velocity from the range of 0 to 36 m/s and pressure 0 to 80 mmWS



Figure 5.13: Measuring instruments (a) Air quality detector, (b) Pitot tube, (c) Digital thermometer.

CHAPTER SIX

RESULTS AND DISCUSSIONS

6.1 Introduction

In this chapter, the experimental results obtained from the designed liquid (water) based air filtration system is presented. Performance of each component of the air filtration and essential points that arise from the results are discussed in detail. Finally, the economic analysis of the air filtration is presented.

6.2 Ambient air Temperature and Air quality

The air filtering experiment was performed for five consecutive days (22-26 may 2022), starting from morning 6:00 to after noon 18:00. The variation in ambient temperature during testing period and the estimated hourly average of the ambient air temperature was in the range of 23°C and 36 °C, in case of air quality, relative humidity was in the range of 11% to 26%, CO₂ was in the range of 860 ppm to 3519 ppm, TVOC where in the range of 0.076 mg/m³ to 1.958 mg/m³ and HCHO where ranged between 0.034 mg/m³ to 0.573 mg/m³.

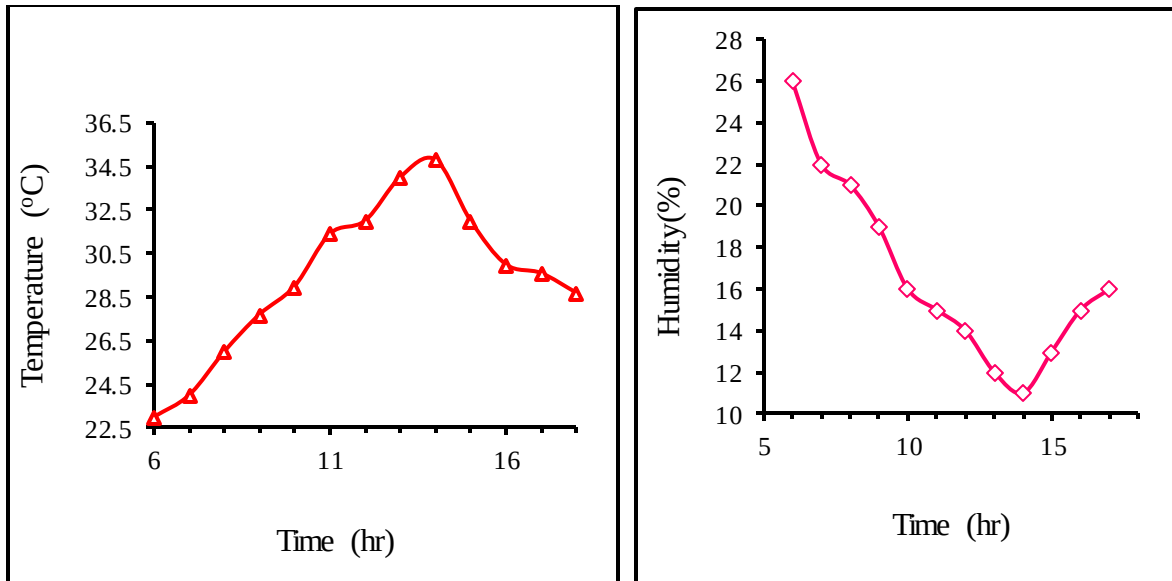


Figure 6. 1: Ambient temperature and humidity versus time.

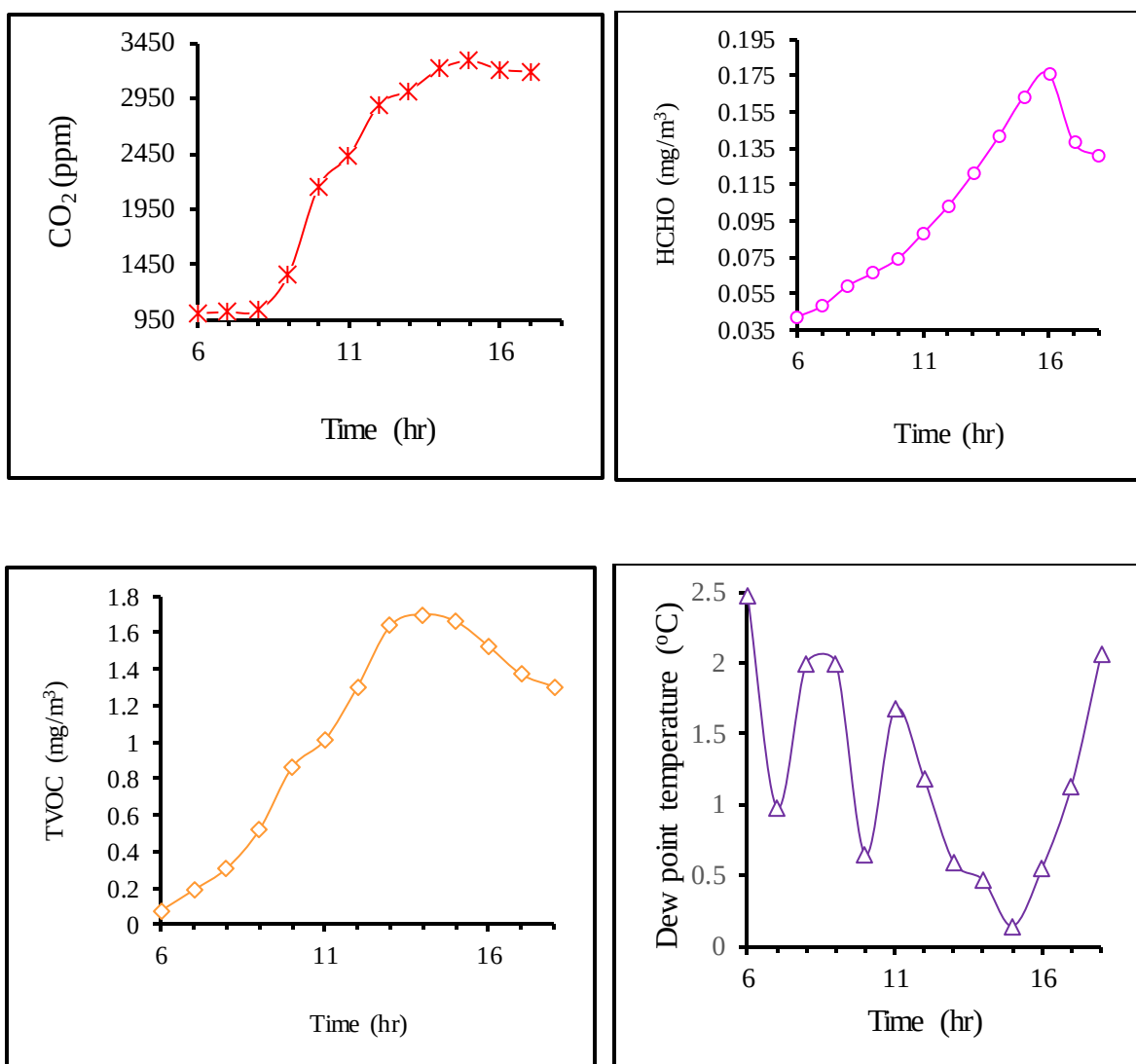


Figure 6. 2: Atmospheric air quality versus time.

Table 6. 1: Ethiopian and international indoor air quality standard

Parameter's	Standard limit
Temperature	19 °C – 27 °C
Humidity	40 % - 70 %
CO ₂	1000 ppm
HCHO	100 mg/m ³
TVOC	0.5 mg/m ³
Dew point temperature	-1 °C – 5 °C (cold area)
	10 °C – 15.6 °C (Hot area)

6.3 Experimental Results of the air filtration system.

To investigate the performance of the central liquid based air filtration system, inlet (ambient) and air quality and outlet temperature and air quality were recorded in a 1 hr interval. The various results obtained are presented and discussed in the following sub sections. The effectiveness of the system where tested in this study under two condition. The first where by evaluating pollutant concentration in present atmospheric air and cleaning it and evaluate the effect under three different cooling pad thickens. The second where by creating false pollution interred in to air filtering system and evaluate the result under three different cooling pad thickens.

6.3.1 Outlet Temperature Variation

Graph 6.2 exhibit the inlet and outlet temperature variations of the air filter system with time for 12 hr. As the Graph illustrates, the outlet temperature of the air filter system varied from 21 °C to 24.8 °C with two cooling pad (4mm) thickens, 19 °C to 21.9 °C with three cooling pad (6mm) thickens, 17 °C to 19.7 °C with four cooling pad (8mm) thickens. The outlet temperature of the air filter system increases with decreasing in number of pad. The maximum outlet air temperature of the air filter system was 24.8 °C, recorded in (n=2) experiment at 14:00 hr and the minimum outlet air temperature of the air filter system was 17 °C, recorded in (n=4) experiment at morning 6:00 hr and afternoon 18:00 hr.

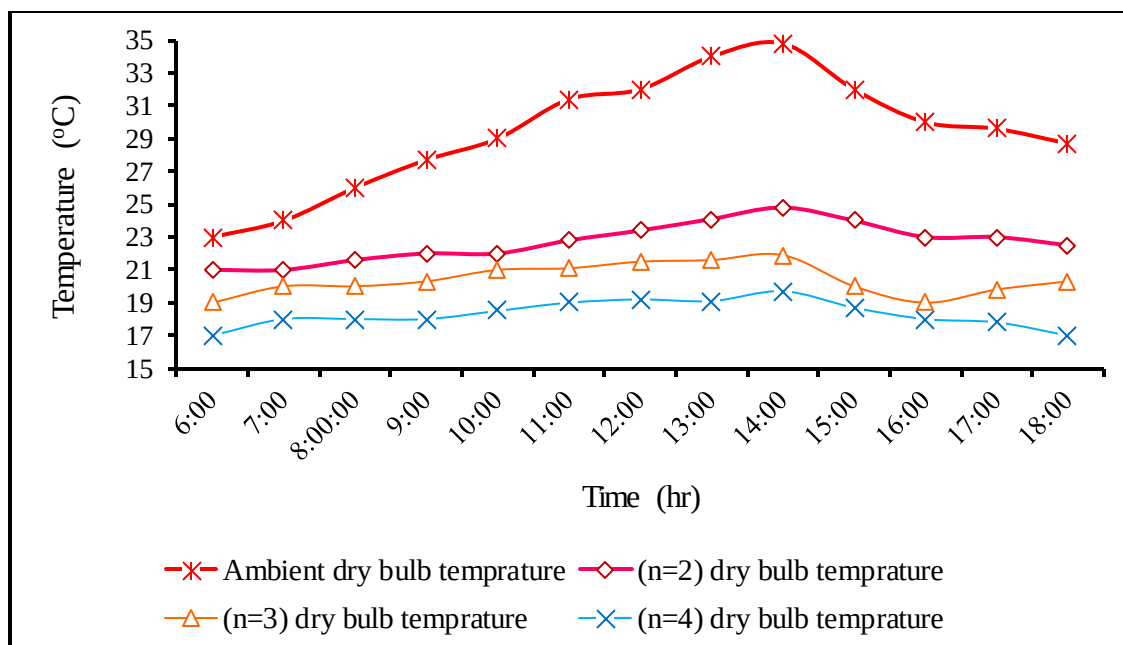


Figure 6. 3: Ambient air inlet temperature and outlet temperature versus time.

From Experimental result it's possible to conclude that ambient air temperature where beyond human comfort range except morning 6:00 to 8:00. With cooling pad thickness of 4 mm (n=2) it's possible to reduce ambient air temperature maximum 10 °C, with cooling pad thickness of 6 mm (n=3) it's possible to reduce ambient air temperature maximum 12.9 °C and with cooling pad thickness of 8 mm (n=4) it's possible to reduce ambient air temperature maximum 15.1 °C, in general the following point where concluded from experimental test result.

- ✓ As number of cooling pad increase out let air temperature decrease.
- ✓ During morning outlet air temperature decrease slightly from inlet ambient air temperature, this is due to high humidity presence in ambient air compered to midday and after noon.

6.3.2 Relative Humidity Variation

Relative humidity of the ambient air of Adama area where less than human comfort range, that relative humidity of this city where between 11 % to 26%, maximum humidity will measure during morning and minimum where measured between 13:00 hr to 14:00 hr in ambient air. As stated by HEVAC humidity group on April 2016, in their study named as humidity and its impact on human comfort and wellbeing in occupied buildings, the relative humidity of any office building should be from 40% to 70%, in general from experimental test the highest maximum humidity was achieved With cooling pad n = 4 and the minimum where with cooling pad n = 2.

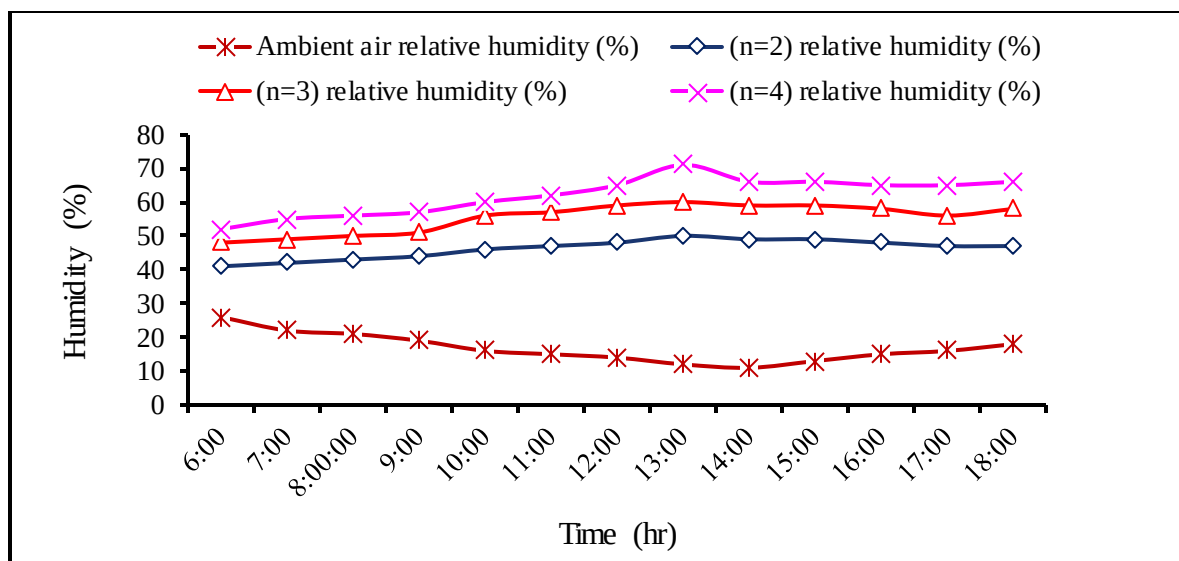


Figure 6. 4: Ambient air relative humidity and outlet temperature versus time.

6.3.3 Saturation effectiveness compression

From experimental result maximum saturation effectiveness where evaluated at 13:00 with cooling pad thickens of 8 mm (n=4) and minimum saturation effectiveness where evaluated at 6:00 with cooling pad thickens of 4 mm (n=2). From 12 hr continuous evaluation it's possible concluded that.

- ✓ As number of cooling pad or thickness of cooling pad increase the saturation effectiveness of the system where increase.
- ✓ Saturation effectiveness where very low during the morning and it start to decline during after noon. This happen due to high moister ambient air during the morning and during afternoon with declining of ambient air temperature
- ✓ Saturation effectiveness where affected by number of the pad (cooling pad thickens), water temperature, change in ambient air dry bulb temperature and change in ambient air wet bulb temperature.
- ✓ Saturation effectiveness where increase as the difference between inlets ambient wet bulb temperatures and outlet dry bulb temperature decrease.

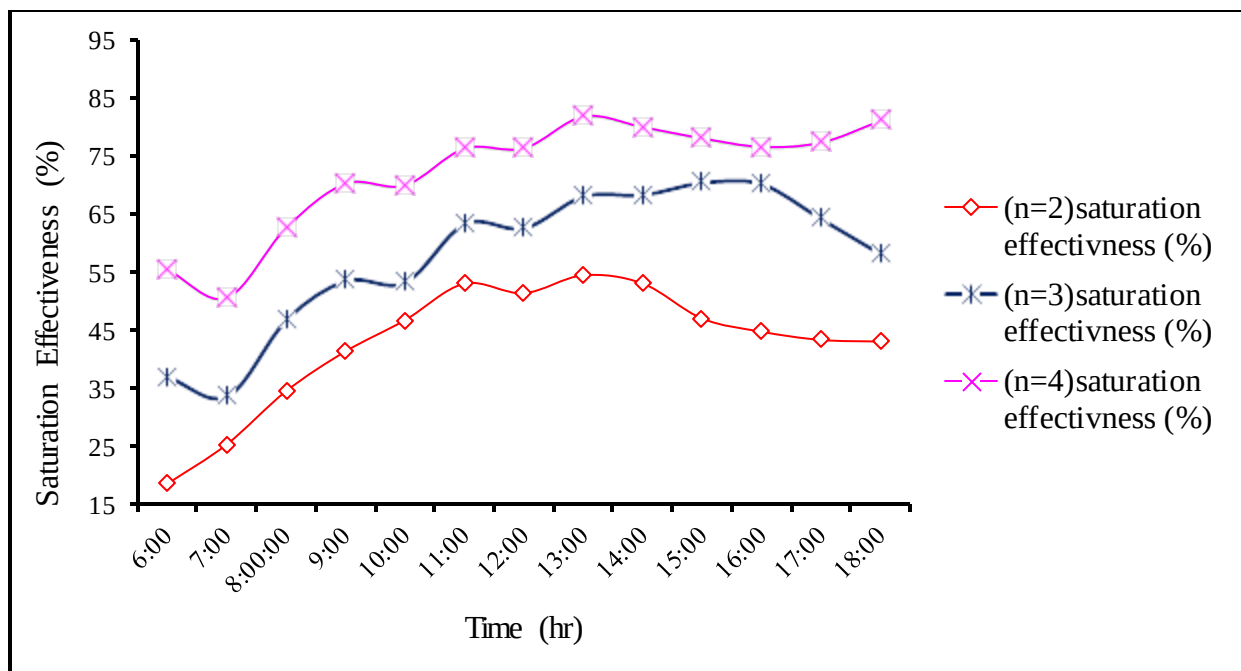


Figure 6. 5: Saturation effectiveness versus time.

6.3.4 Achievable temperature drop

Achievable temperature drop was the difference between evaluated outlet dry bulb temperature and outlet wet bulb temperature from evaporative cooling system multiplied by saturation effectiveness. The following point were concluded from experimental test result.

- ✓ The maximum achievable temperature drop where occur from 10:00 to 14:00 with cooling pad thickness 4 mm (n-2). This occurs due to high temperature difference between outlet dry bulb temperature and outlet wet bulb temperature at maximum saturation effectiveness point.
- ✓ The maximum achievable temperature drop occurred where there is maximum phase velocity take place at phase of cooling pad in this experiment at air inlet velocity of 3.8091 m/s and pressure 9.8 pa.
- ✓ achievable temperature drop highly affected by fan air sucking speed, outlet temperature, area of air cooling system and pressure of the air.

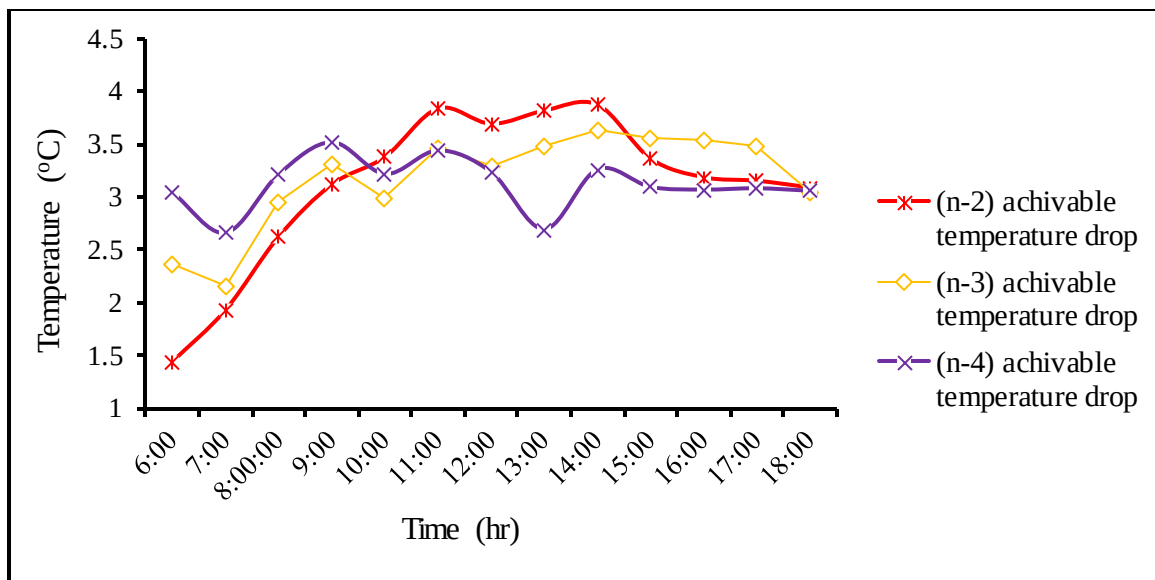


Figure 6. 6: Achievable temperature drop versus time.

6.3.5 Achievable temperature

Achievable temperature indicate the possibility of water vapour creation in the air, when the temperature drop was high enough the water vapour in the air will condensed in to droplet. During experimental test the maximum achievable temperature where occur at 14:00 pm, with n-2 cooling pad 4mm thick and from experimental test the minimum achievable temperature where occur at 18:00 pm, with n-4 cooling pad 8mm thick.

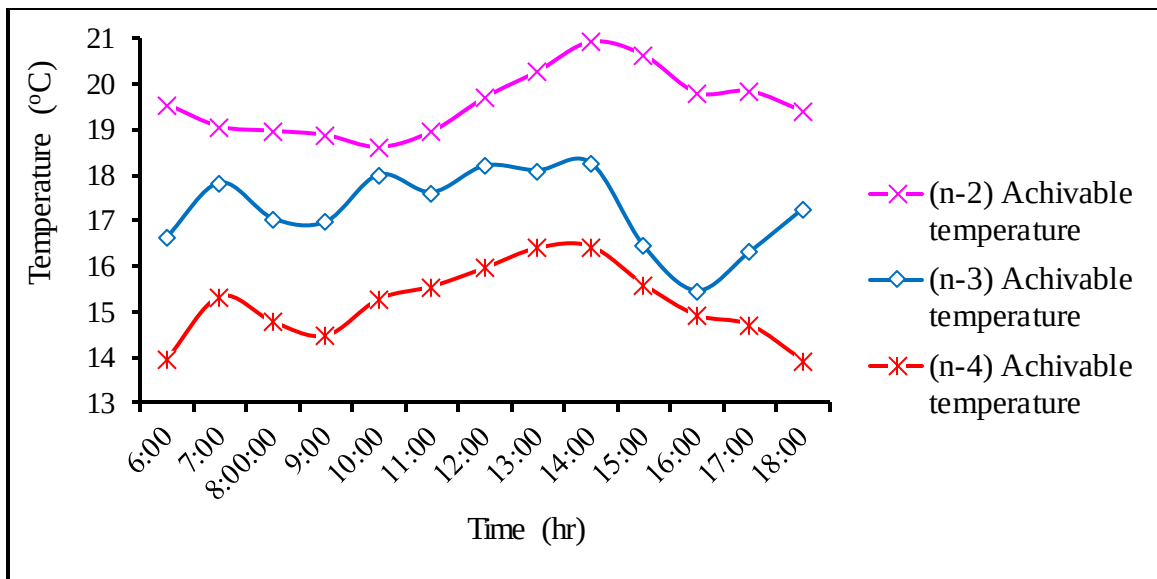


Figure 6. 7: Achievable temperature versus time.

6.3.6 Dew point temperature

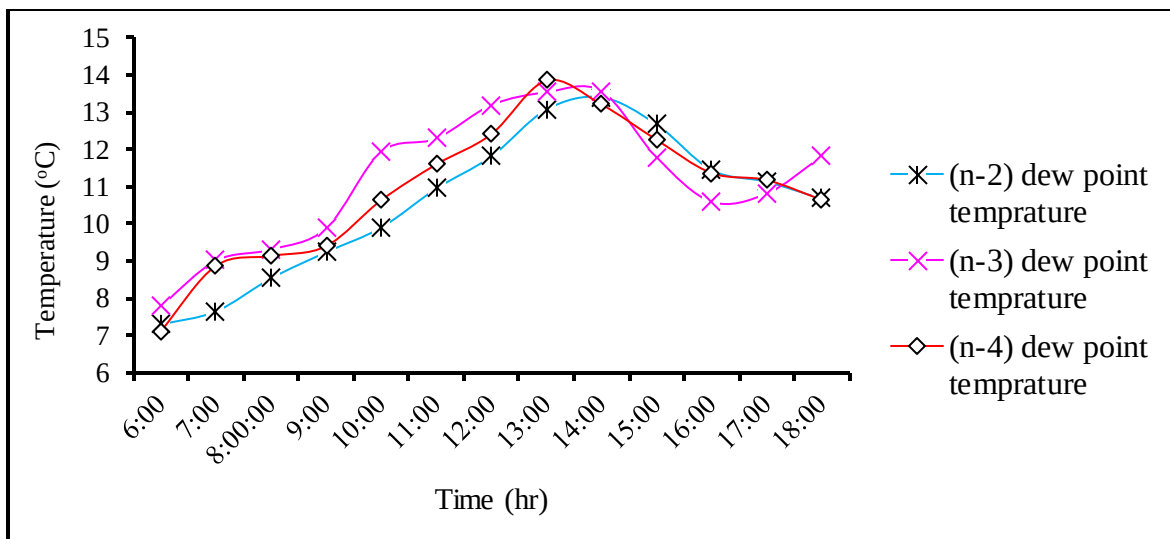


Figure 6. 8 Dew point temperature's versus time.

Dew point temperature is the temperature that air needs to be cool cooled at to achieve high relative humidity at constant pressure. During test maximum due point temperature 13.8524 °C where occur at 13:00 with 8 mm thick n-4 cooling pad. There is proper relationship between dew point temperature and relative humidity, as dew point temperature increase saturation effectiveness also increase.

6.3.7 Evaporation rate

Graph 6.8 exhibit that the evaporation rate where directly proportional with dry bulb temperature, because evaporative rate are higher at high temperature, as temperature increase the amount of energy necessary for evaporation decreases. High evaporative rate occurred at 13:00 and 14:00 with 8 mm thick n-4 cooling pad. This indicates that at high evaporation rate relative humidity will be high, dry bulb temperature where high, saturation effectiveness was also high, dew point temperature is high, achievable temperature where low and moisture absorbed by the air will be high.

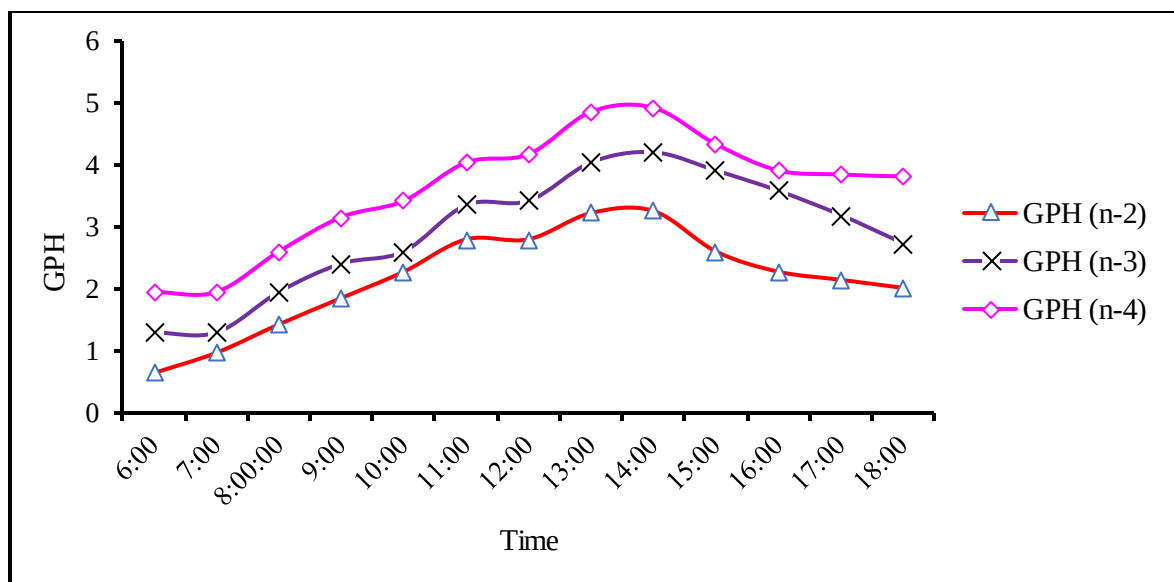


Figure 6. 9: Evaporative rate versus time.

6.3.8 Compression of Moisture absorption rate by air

Moisture absorption rate is rate at which moisture of water absorbed by air, graph 6.9 revile that maximum absorption rate take place at 13:00 and that was 0.005911 kg/kg. There is direct relationship between moisture absorption rates, evaporation rate, relative humidity, dew point temperature and inverse relation between achievable temperature drop and achievable temperature.

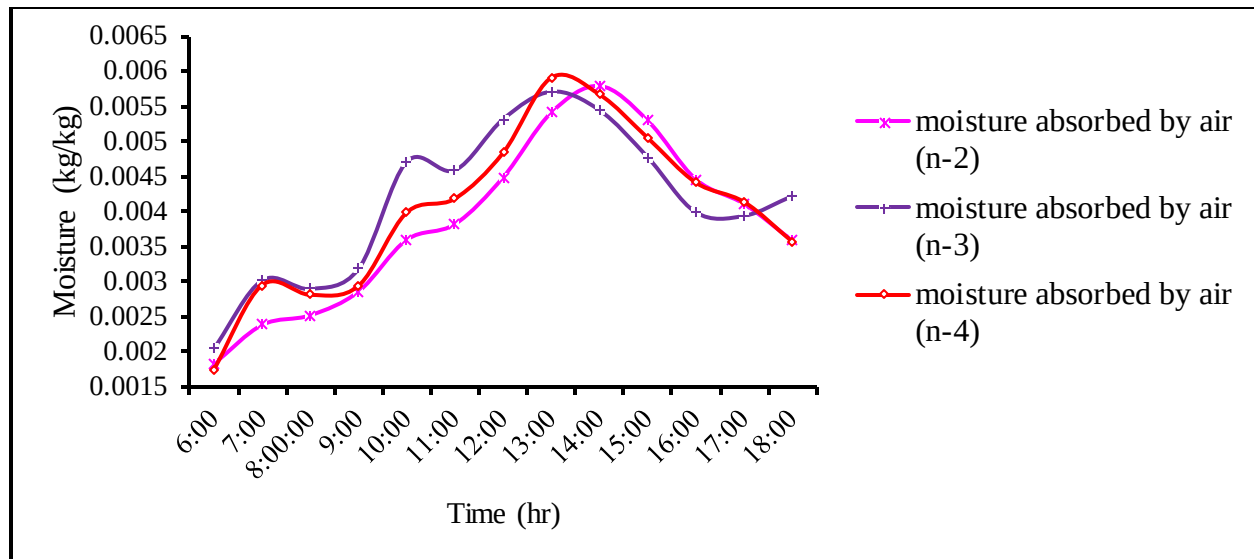


Figure 6.10: Moisture absorption rate versus time.

6.3.9 Compression of sensible heat transfer

The variations of sensible heat transfer of air filtering system with respecting to time is illustrated in graph 6.10. The sensible heat transfer between the air and water is evaluated from measured inlet air dry bulb temperature of ambient air minus outlet dry bulb temperature of air times by air flow rate. It is observed that the sensible heat transfer increases with the increase in the difference of dry bulb temperature of inlet ambient air and outlet air, from 7:00 to 14:00 and decreases when the air temperature start to decrease.

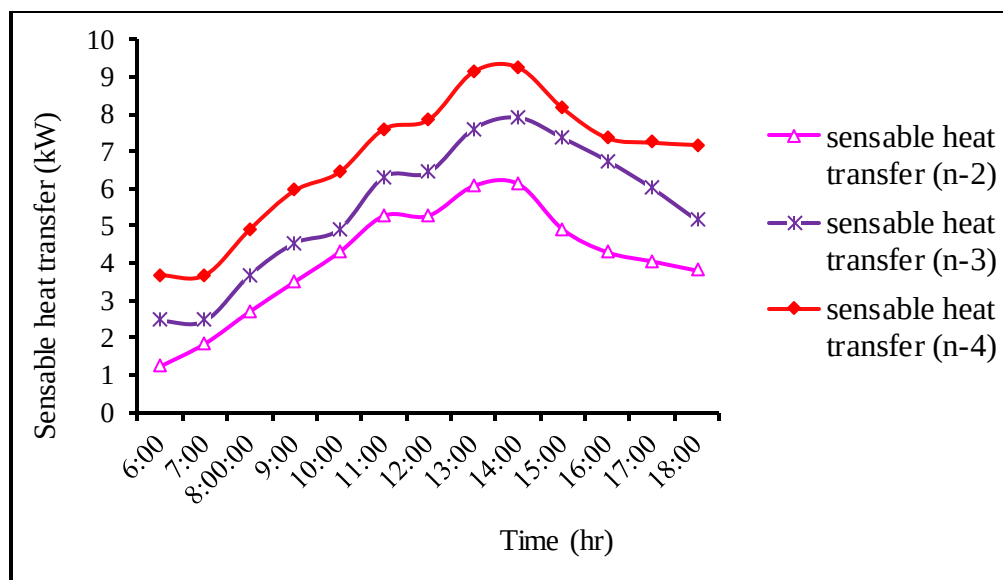


Figure 6.11: Sensible heat transfer versus time.

This is because sensible heat transfer is strongly influenced by the dry bulb temperature of air and air flow rate. During the air filtering process, the average sensible heat transfer for the consecutive 12 hour of a days from n-2 to n-4 ranged from 1.22592 kW to 9.2557 kW.

6.3.10 Compression of mass flow rate and air flow rate.

Graph 6.11 below shows the relationship between air flow rate and mass flow rate in liquid (water) based air filtration system. The maximum air flow rate and mass flow rate was 0.9872928 m³/s and 1.20943368 kg/s at 14:00, mass flow rate and air flow rate have direct effect on heat transfer.

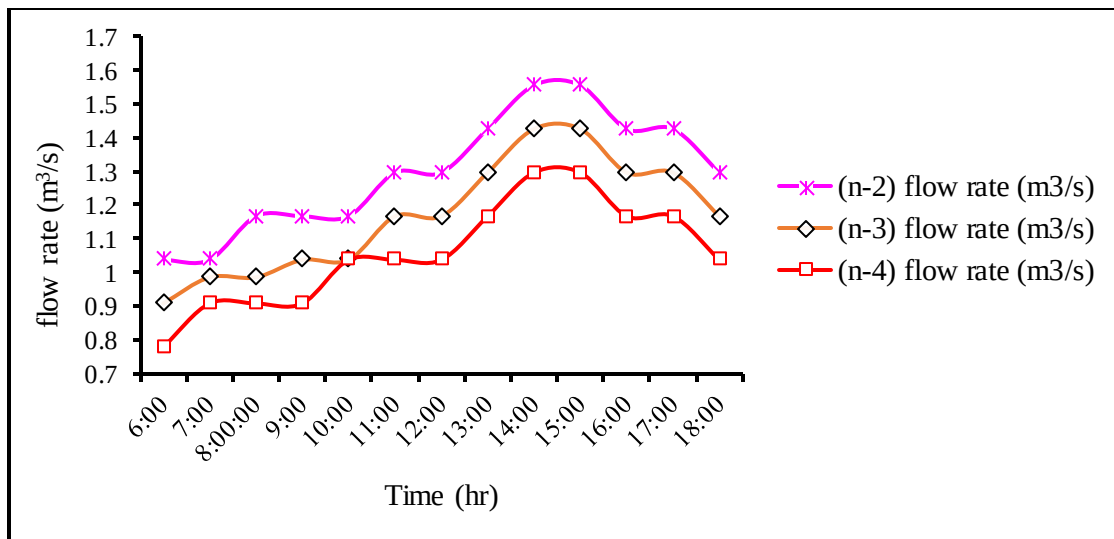


Figure 6. 12: Air flow rate versus time.

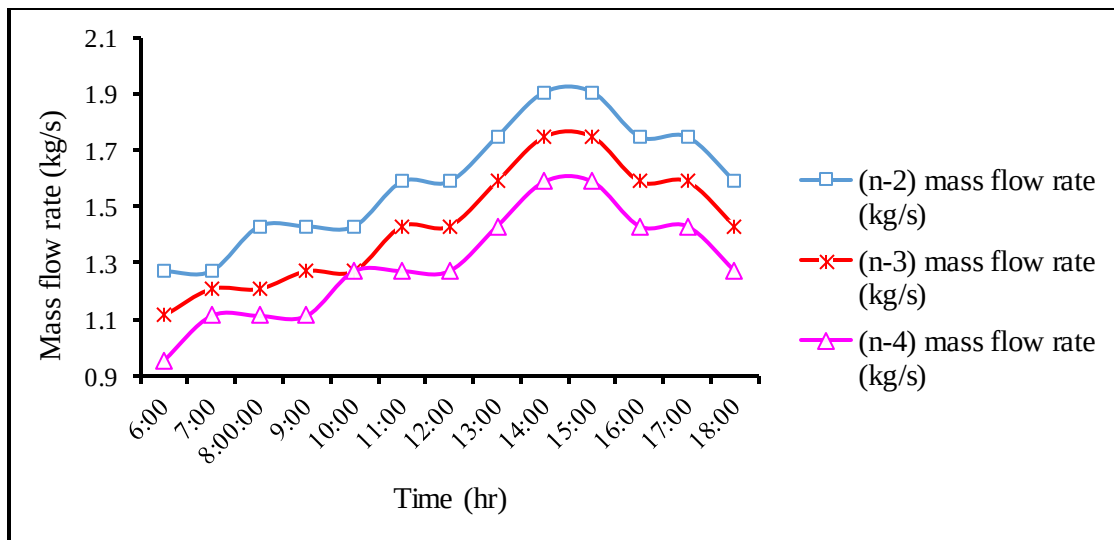


Figure 6. 13: Air mass flow rate versus time.

As mass flow rate doubled evaporative rate and heat transfer where doubled, during this dry bulb temperature of inlet polluted air where decreased, relative humidity where increase. The effect of mass flow rate and sensible heat transfer where direct proportional.

6.3.11 Experimental Results of the ambient air quality.

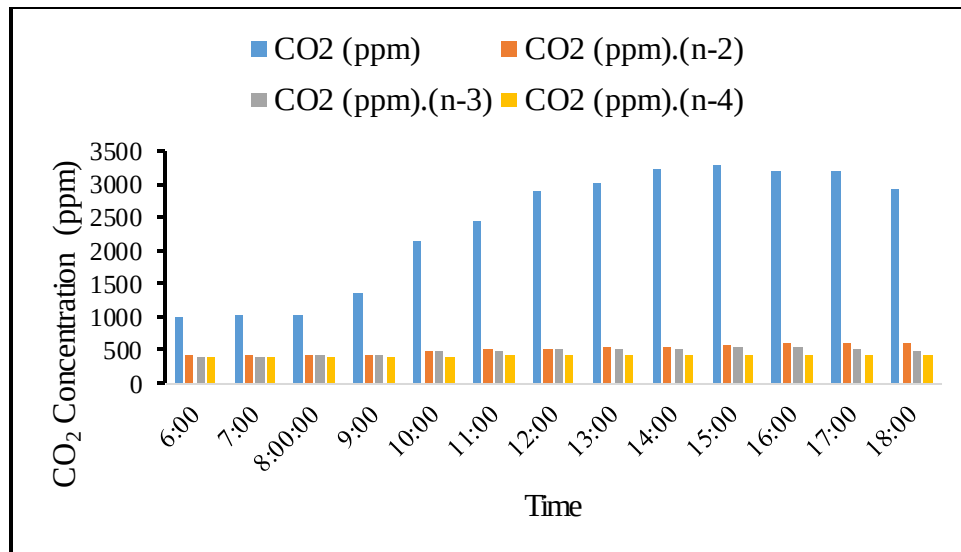


Figure 6. 14: CO₂ versus time.

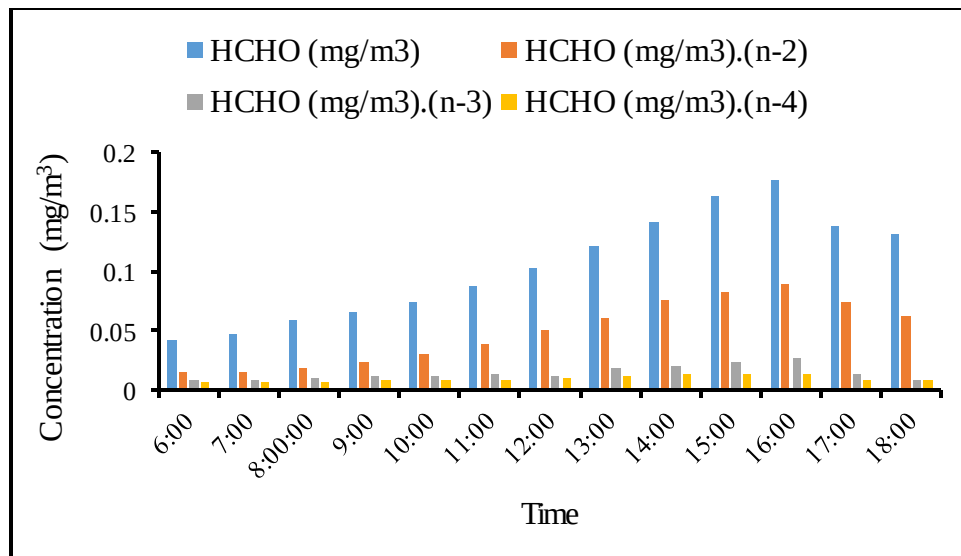


Figure 6. 15: HCHO versus time.

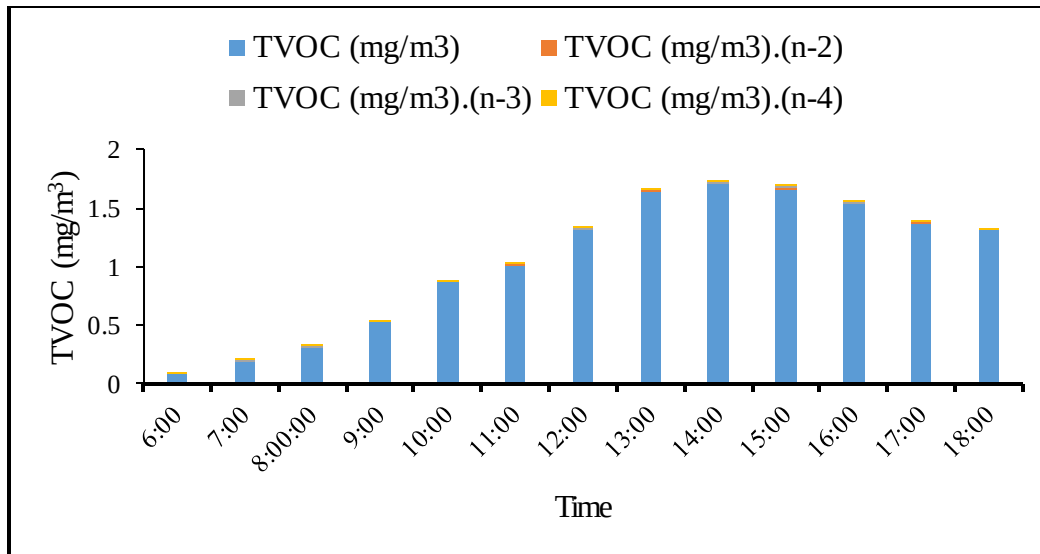


Figure 6. 16: TVOC versus time.

The change in a number of pad highly affect total volatile organic compound, this is due to volatile organic compound plays a significant role in formation of particulate matter or the maximum number of total volatile organic compound indicate that high value of PM and ozone. Particulate matter where washed by water easily, this is why there is a lot of decrement where noticed on TVOC during test. Increasing number of pad has direct effect on relative humidity, as number of cooling pad increase relative humidity where increase and simultaneously concentration of pollutant where decreased, during this test cooling pad n-4 where noticed as effective.

6.4 Experimental Results of the false pollution

Idea of false pollution creation where to create more concentrated pollution to evaluate the performance of liquid based air filtration system at maximum polluted condition as well as to estimate the performance of the system in the polluted area. During experiment it where possible to crate false pollution effectively from wood and different plastic by burning, this made possible to elevate inlet dry bulb temperature to 38 °C, maximum relative humidity of inlet air where 27%, dew point temperature 15.78 °C. During false pollution test it is noticed that from evaluated data maximum saturation effectiveness evaluated with n-2 where 45%, with n-3 where 63%, and with n-4 where 74%, this indicate that as thickness increase evaporative rate increase.

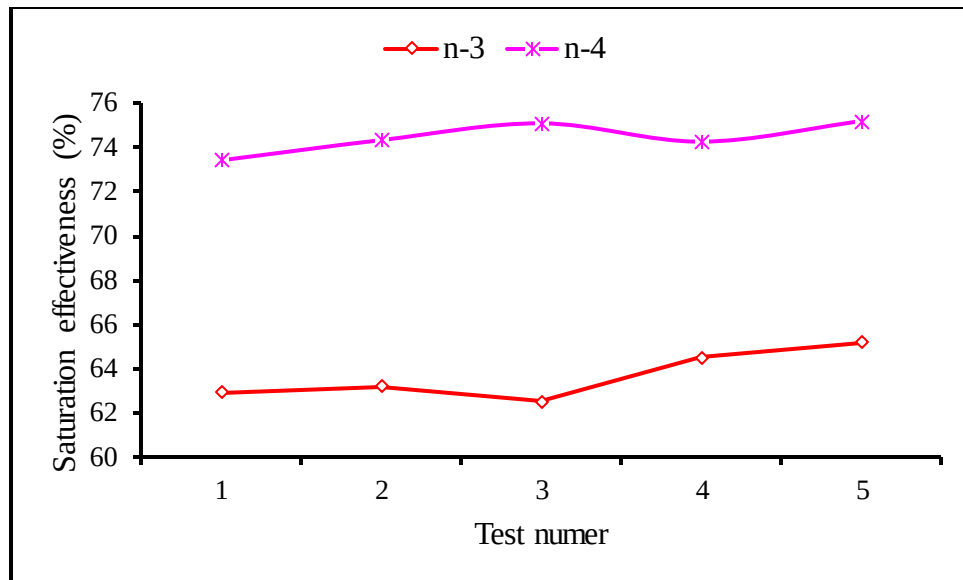


Figure 6. 17: Saturation effectiveness compression.

Dry bulb temperature where decreased from 35 °C to 24 °C with n-4 cooling pad this indicate high relative where achieved, as the value of inlet wet bulb temperature and out let dry bulb temperature become nearer to each other saturation effectiveness of the system where increased enough.

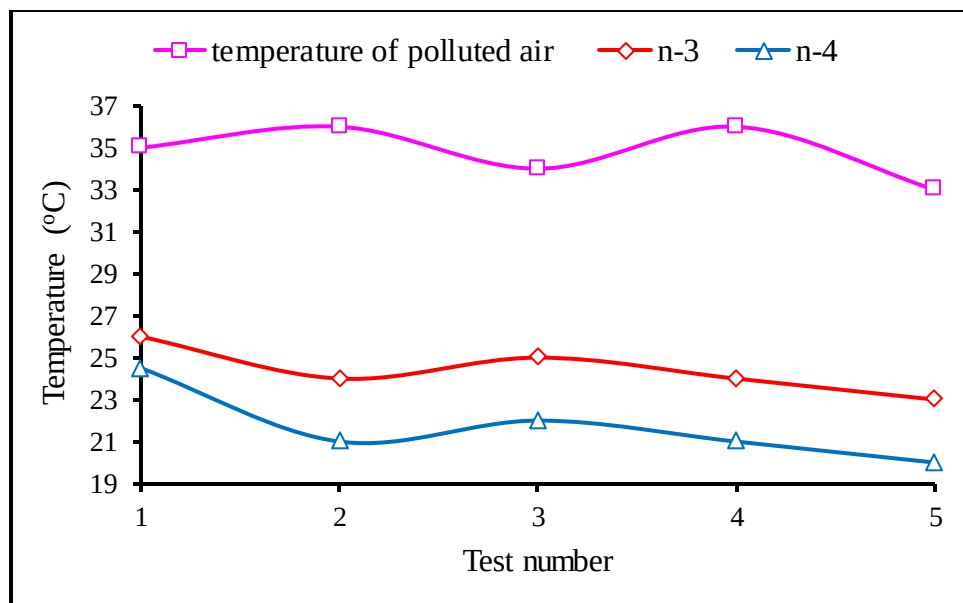


Figure 6. 18 Temperature compression.

Maximum achievable temperature drop where 6.4 °C with n-4 cooling pad during experiment this indicate as inlet air temperature drop the evaporative rate become maximum at this point the air can hold water and humidified. On other hand as achievable temperature drop increase the saturation effectiveness were also increase. From this it's clear that the difference between inlet dry bulb and outlet dry bulb where high, the difference between inlet dry bulb air and inlet wet bulb will be less.

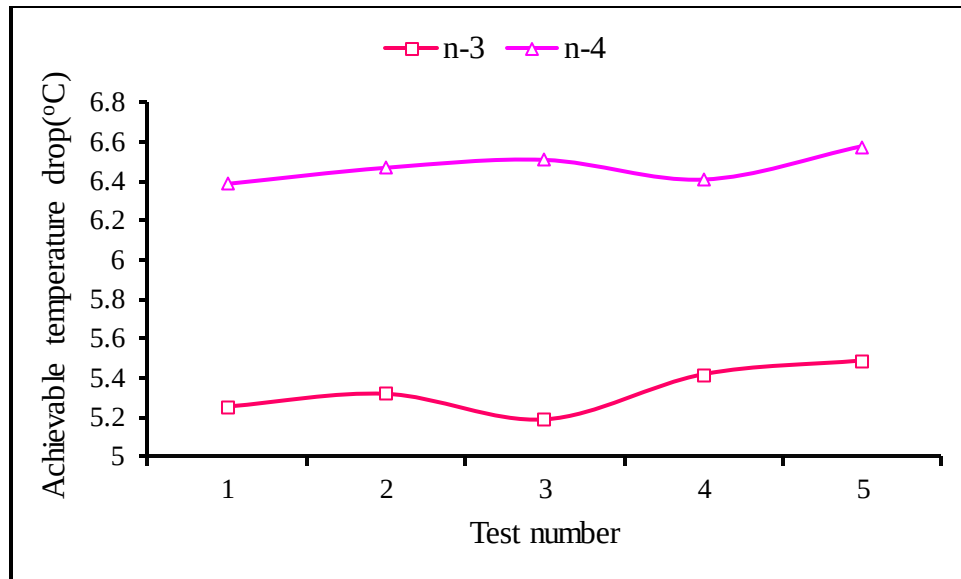


Figure 6. 19: Achievable temperature drop compression.

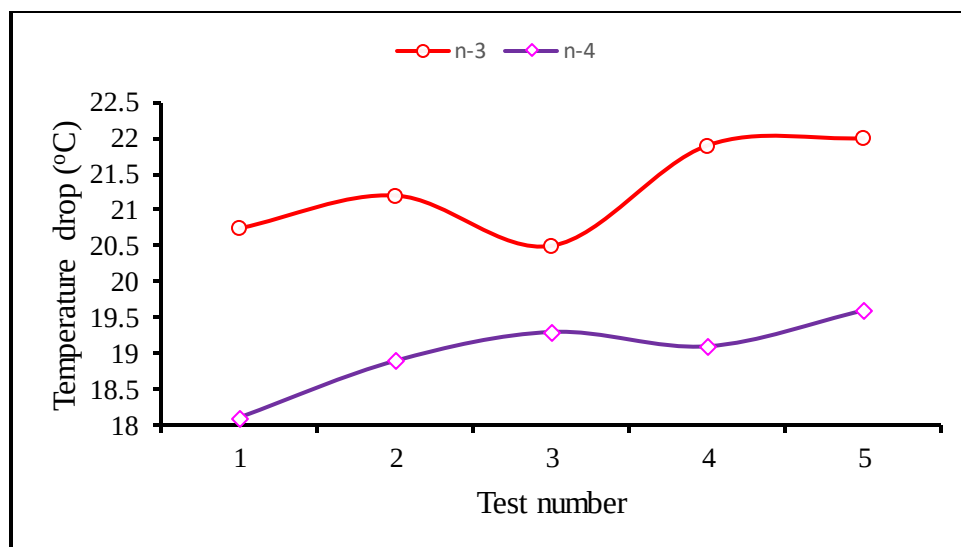


Figure 6. 20: Temperature drop compression.

Temperature drop indicates the difference between inlet dry bulb and outlet dry bulb, as it gets high the relative humidity of outlet temperature were also high, evaporative rate also increase and there is an inverse relation between temperature drop and water temperature. As air temperature highly drop the water temperature highly increase.

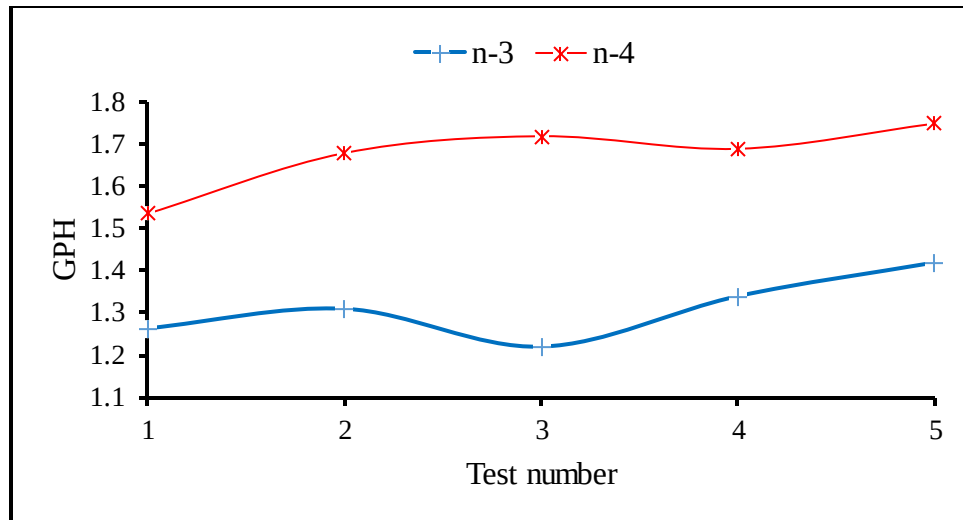


Figure 6. 21: Evaporative rate compression.

During test the maximum evaporative rate evaluated where 1.7, as evaporative rate increase the time for air to be humidified were also increase, the result showed that the maximum evaporative rate where achieved with n-4 cooling pad, the large thickens of the cooling pad where increase the time of contact between water and air, and the area were also increased this let the air to be more humidified.

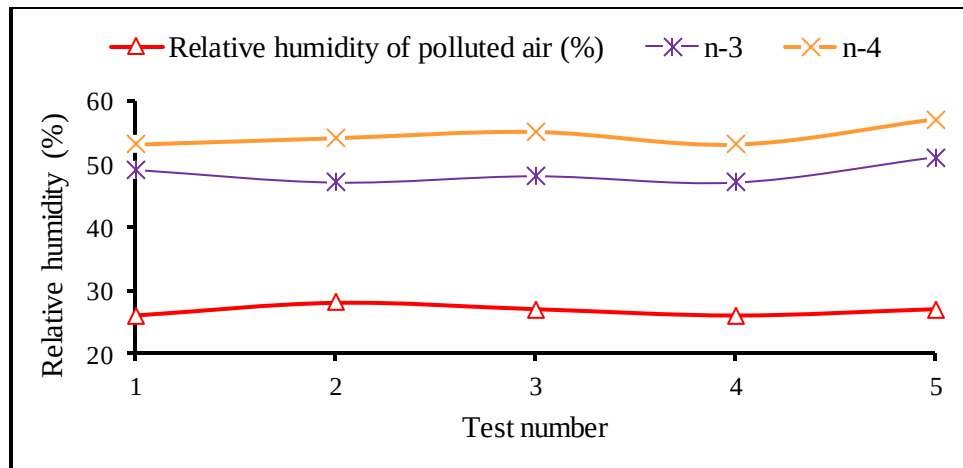


Figure 6. 22: Relative humidity compression.

6.4.1 Experimental Results of the false pollution air quality

The change in a number of pad highly affect total volatile organic compound, this is due to volatile organic compound plays a significant role in formation of particulate matter or the maximum number of total volatile organic compound indicate that high value of PM and ozone. Particulate matter where washed by water easily, this is why there is a lot of decrement where noticed on TVOC during test. Increasing number of pad has direct effect on relative humidity, as number of cooling pad increase relative humidity where increase and simultaneously concentration of pollutant where decreased, during this test cooling pad n-4 where noticed as effective.

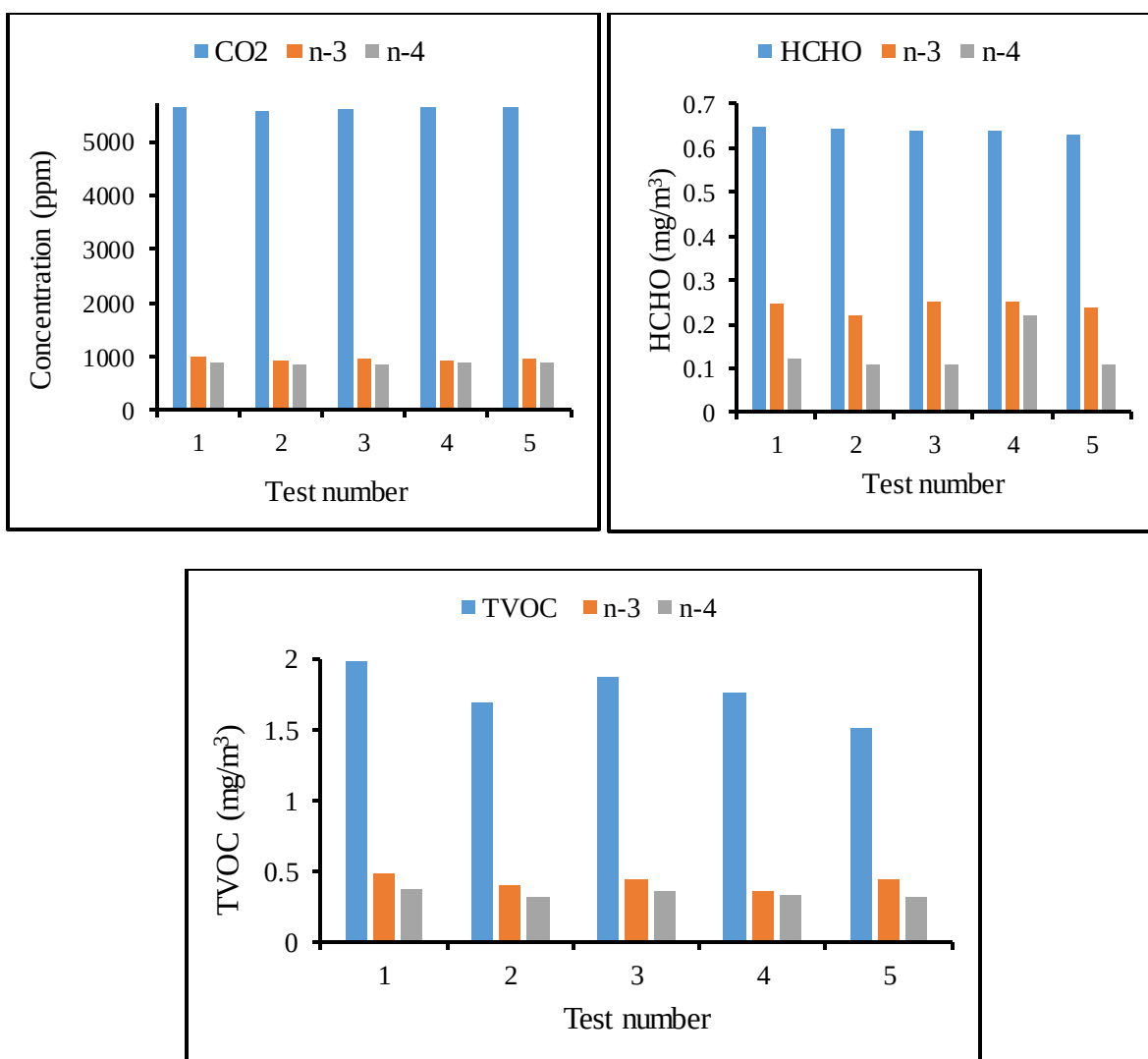


Figure 6. 23: False pollution air quality result.

CHAPTER SEVEN

CONCLUSIONS AND RECOMMENDATIONS

7.1 Conclusions

Electric power based central liquid based air filtration system is the best alternative to shift the country's dependence on HVAC and minimize the environmental pollution. In this study experimental investigation of central liquid based air filtration system has been carried out. The system uses a cooling chamber, cooling pad, water storage tank, water distribution header, water filter tank, pump, motor and Air exhaust, closed-loop piping network as a connection for the whole system, blower, frame, and support. Cooling chamber contain in inside four 36 cm length and 56 cm height cooling pad and 30 cm by 36 cm water distribution header. To control heat transfer from external to internal portion of air filtering system, the use of white colored paint over the surface of prototype, use insulator for keeping internal temperature as it is during test inside, and the use of a reflector inside the system. Every cooling pad has 2 mm thickens and pump has capability of delivering water up to 30 l/min to head 1m to 30m. The main challenges faced while conducting the experimental test of the system were the difficulty of managing measurement during false pollution test as there is only one air quality sensor and water linkage. The major aim of this thesis work was to size, manufacture, and conduct an experimental investigation on the performance of central liquid based air filtration system. The following conclusions are drawn from this study:

- Based on the current status of the air pollution a method of Electrical power application has been tested and new technical and technological opportunities of an Electrical power application in liquid based air filtration system for office building have been established.
- To reduces the environmental air pollution and the dependence on HVAC system.
- The range of ambient dry bulb temperature of Batu caustic soda where ranged per year from 298.41K to 301.8K, The ground level or surface wind speed ranged from 2.94 m/s to 5.824 m/s, The wind speed at 10m height where ranged from 5.054 m/s to 7.522 m/s, wet bulb temperature ranged 287.6 K to 288.42 K, SO₂ concentration where ranged from 273.1µg/m³ to 289.2µg/m³ NO₂ concentration where ranged from 21.24µg/m³ to 22.81µg/m³, CO concentration where ranged from 35184µg/m³ to 36975µg/m³, O₃ concentration where ranged from 43.02µg/m³ to 45.22µg/m³, PM₁₀ concentration where

ranged from $149.8\mu\text{g}/\text{m}^3$ to $395.3\mu\text{g}/\text{m}^3$ and $\text{PM}_{2.5}$ concentration where ranged from $4472\mu\text{g}/\text{m}^3$ to $5327\mu\text{g}/\text{m}^3$ yearly from 2010 to 2021 respectively.

- $\text{PM}_{2.5}$, PM_{10} , and O_3 where the most concentrated pollutant over the area specially $\text{PM}_{2.5}$.
- Maximum air supply for ground floor was 1675.5 l/s, maximum sensible cooling load of ground floor where 12597 W and latent cooling load of ground floor where 1720 W.
- Maximum air supply for ground floor was 1977.8 l/s, maximum sensible cooling load of ground floor where 18433 W and latent cooling load of ground floor where 7341 W.
- The maximum average inlet dry bulb temperature, The minimum average outlet dry bulb temperature, maximum relative humidity, maximum saturation effectiveness and maximum dew point temperature of result during experimental test on present atmospheric air where 34.8°C , 17°C , 71%, 80.22%, and 15.7°C .
- The maximum average inlet dry bulb temperature, The minimum average outlet dry bulb temperature, maximum relative humidity, saturation effectiveness and maximum dew point temperature of result during experimental test during false pollution where 38°C , 21°C , 62%, 74%, and 14.2°C .

7.2 Recommendations

Air conditioner contain CFC's and HCFC's, which are the cooling agents that can punch holes in our ozone layer. Evaporative cooling is the most environmentally friendly cooling techniques. It uses water as a refrigerant and it has much lower emission than AC unit. Generally, this study shows that liquid (water) based air filtration was possible by scaling up the system and creating awareness to the society about the benefits of the technology. Therefore, the main points to be recommended for future work based on the result of this study;

- In this study, the air filter was designed for delivering 6000 l/hr of clean air. Thus, the air filter can be investigated further for large scale application.
- Since electrical power is Expensive, using other cheap renewable energy source like solar energy, biomass instead for the air filtering system can be studied.
- The maximum performance (saturation effectiveness) of air filtration system in this study where 80.22% with cooling pad thickens 8mm, by increasing cooling pad thickens, by using water or air heater in the process it is possible to increase saturation effectiveness of the system.

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APPENDIX

APPENDIX A

Calcium oxide (CaO₂) plant Emission Numerical Analysis result.

Year	Pollutant Concentration ($\mu\text{g}/\text{m}^3$)					
	SO ₂	NO ₂	CO	O ₃	PM ₁₀	PM ₁₀
2010	21.85	170.8331	2761.663	324.5602	11.28839589	336.994035
2011	19.26	152.6402	2472.624	288.8186	18.63000158	301.7337541
2012	19.03	144.7885	2464.805	288.6438	22.11158215	299.6429357
2013	21.39	159.1186	2711.308	323.5561	23.06619914	383.6375634
2014	19.96	161.7164	257961.6	308.5952	22.37065253	371.8692974
2015	18.9	139.7831	2314.413	283.513	20.5300256	343.3730652
2016	21.24	163.6625	2542.003	311.8034	22.89253533	375.5502011
2017	20.08	154.2212	2514.441	301.613	21.95893158	363.7056932
2018	20.51	158.6476	2571.323	303.2289	24.75238242	367.6117185
2019	19.51	147.4372	2465.546	288.832	26.45276614	355.1653119
2020	20.97	162.6518	2617.998	310.0214	28.13941081	380.118813
2021	18.58	143.4521	2318.502	285.1387	24.92598887	335.8986661

Caustic Soda (NaOH) Plant Emission Numerical Analysis result.

Year	Pollutant Concentration ($\mu\text{g}/\text{m}^3$)					
	SO ₂	NO ₂	CO	O ₃	PM ₁₀	PM _{2.5}
2010	21.8158	175.253	2833.115785	33.2958	11.58046127	345.7131028
2011	19.7958	160.2353	2595.658084	30.319	19.55700164	316.7475589
2012	19.5699	152.078	2588.897394	30.3176	23.22480772	314.7287028
2013	21.3823	163.4256	2784.698068	33.2314	23.69055413	394.0218503
2014	20.2081	167.7473	2675.816845	32.0104	23.20491573	385.7373268
2015	19.4935	147.1534	2436.444697	29.8462	21.6125054	361.477982
2016	21.433	169.3574	2630.456524	32.2653	23.68912453	388.6181829
2017	20.4387	160.678	2619.712692	31.4241	22.87828689	378.9329714
2018	20.8582	165.1607	2676.885021	31.5678	25.76856086	382.703563
2019	20.0286	154.6511	2586.180201	30.2964	27.74705224	372.5429095
2020	21.191	168.5025	2712.169348	32.1173	29.15160235	393.7919155
2021	19.3367	152.1126	2458.474074	30.2353	26.43081948	356.1775243
Average	20.46264	161.3629	2633.209	31.41053	23.21131	365.9328

Boiler Plant Emission Numerical Analysis result.

Year	Compounds(mg/m ³) = 1 x 10 ⁴ μ g/m ³					
	SO ₂	NO ₂	CO	O ₃	PM ₁₀	PM _{2.5}
2010	23.5005	1.88787	3051.899596	35.867	12.47474786	372.4103633
2011	20.9433	1.69523	2746.1085	32.0763	20.69057121	335.1069885
2012	20.6949	1.6082	2737.721689	32.0604	24.5598995	332.821068
2013	23.0122	1.75883	2996.963515	35.7645	25.49638225	424.0564257
2014	21.5802	1.79137	2857.502641	34.1838	24.78051071	411.9285789
2015	20.5781	1.55341	2572.00271	31.5067	22.81497402	381.5897609
2016	22.9317	1.812	2814.390397	34.5215	25.34557936	415.7921912
2017	21.7512	1.70996	2787.939534	33.442	24.34743347	403.2664402
2018	22.2111	1.75874	2850.519464	33.6154	27.44002217	407.527386
2019	21.2023	1.63714	2737.734967	32.0718	29.37307891	394.3745874
2020	22.6519	1.80119	2899.142993	34.3314	31.16127825	420.9394496
2021	20.2982	1.59677	2580.727259	31.7388	27.74515177	373.8892575
Annual	21.77963	1.717558	2802.721	33.43164	24.6858	389.4752

Ground level Emission Numerical Analysis result.

Year	Compounds(mg/m ³) = 1 x 10 ⁴ μ g/m ³					
	SO ₂	NO ₂	CO	O ₃	PM ₁₀	PM _{2.5}
2010	19.02317	1.52819	2470.450193	29.0336	10.0980528	301.4585589
2011	44.20773	3.57835	5796.577437	67.7078	43.674348	707.3550112
2012	34.00633	2.64264	4498.690717	52.6824	40.35742288	546.8996558
2013	22.96032	1.75487	2990.212108	35.6839	25.43894529	423.101133
2014	18.22618	1.51295	2413.383229	28.8709	20.92906865	347.9057235
2015	45.42402	3.42899	5677.437716	69.548	50.36176419	842.32108
2016	48.92152	3.86564	6004.097433	73.6466	54.07115095	887.0328831
2017	26.10643	2.05235	3346.167804	40.1381	29.22251254	484.0123547
2018	22.4323	1.77625	2878.903226	33.9501	27.71325344	411.5852992
2019	51.11706	3.94701	6600.462904	77.3226	70.81617472	950.8060007
2020	26.62723	2.11729	3407.930038	40.3565	36.62994768	494.8125008
2021	21.10475	1.66021	2683.265747	32.9999	28.84753324	388.7447751
Annual	31.67975	2.488727	4063.965	48.49503	36.51335	565.5029

APPENDIX B

Experimental setup during the air filtration process

