A DISSERTATION ON

DESIGN AND ECONOMIC ANALYSIS OF HEAT RECOVERY WHEEL USED IN HVAC SYSTEM OF A BUILDING

Submitted in partial fulfillment

for the award of degree of

Masters of Technology

(Thermal Engineering)

Submitted By

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DECLARATION

I hereby declare that the work, which is being presented in this dissertation, entitled "***DESIGN AND ECONOMIC ANALYSIS OF HEAT RECOVERY WHEEL USED IN HVAC SYSTEM OF A BUILDING***" towards the partial fulfillment of the requirements for the award of degree of Master of Engineering with specialization in Thermal Engineering, from Delhi Technological University, Delhi is an authentic record of my own work carried out under the supervision of **DR. R. S. MISHRA (**Professor, Department of Mechanical Engineering, Delhi Technological University, Delhi) and **DR. K. MANJUNATH** (Assistant Professor, Department of Mechanical Engineering Delhi Technological University, Delhi). The matter embodied in this dissertation report has not been submitted by me for the award of any other degree.

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This is to certify that the above statement made by the candidate is correct to the best of my knowledge.

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CERTIFICATE

It is certified that SANILTUMBER Roll no. 2K13/THE/26, student of M.Tech. Thermal Engineering, Delhi Technological University, has submitted the dissertation titled **"*DESIGN AND ECONOMIC ANALYSIS OF HEAT RECOVERY WHEEL USED IN HVAC SYSTEM OF A BUILDING*"** under my guidance towards the partial fulfillment of the requirements for the award of degree of Master of Technology.

His work is found to be satisfactory and his discipline impeccable during the course of the project. His enthusiasm and attitude towards the project is appreciated.

I wish him success in all his endeavors.

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ABSTRACT

Heat Recovery Wheel is a type of heat exchanger positioned between the supply and exhaust air streams of an air handling system of a building. It is also known as a rotary heat exchanger or a rotary air to air enthalpy wheel. This heat exchanger pre cools the fresh air entering the building by removing the heat from the return air stream and providing the same to the fresh air stream. This reduces the load on the central chillers. The aim of the project was to evaluate the payback period of the heat recovery wheel in an office building in Kolkata by doing an economic analysis with respect to the conventional system.

In order for the Payback period to be less and economically feasible, the operational costs of the Heat Recovery Wheel should be significantly less in comparison to the conventional system so as to offset its high capital cost. Below are the various costs of both the systems and the Payback period associated with the Heat Recovery Wheel:

Capital Cost with Heat Recovery Wheel	66 Lakhs
Capital Cost without Heat Recovery Wheel	23 Lakhs
Operational Cost with Heat Recovery Wheel	72 Lakhs
Operational Cost without Heat Recovery Wheel	93 Lakhs

Therefore from the above data the Payback Period comes out to be 2.1 years which is economically feasible and after which the Heat Recovery Wheel would continue to give substantial energy and electrical savings for the rest of its life.

The result shows significant potential that Heat Recovery Wheels have in modern day mechanical ventilation of buildings and why they should be promoted an building regulations.

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LIST OF ABBREVIATIONS/ SYMBOLS

a) HVAC- Heating Ventilation and Air Conditioning
b) ZEB- Zero Energy Building
c) EIA- Energy Information Administration, USA
d) DOE- Department of Energy, USA
e) ε- Sensible, or total effectiveness of Heat Recovery Wheel
f) x_1- OA temp (°Fdb) or enthalpy (btu/lb.)
g) x_2- SA temp (°Fdb) or enthalpy (btu/lb.)
h) x_3- RA temp. (°Fdb) or enthalpy (btu/lb.)
i) V_s- Supply (or outside) air volume (cfm)
j) V_{min}- The lower of the exhaust or supply air volume (cfm)
k) E- The total energy demand due to the ventilation air (kJ)
l) m_{ven}- Mass flow of ventilation dry air (kgdry-air/h)
m) h_{out}- Enthalpy of the outside air (kJ/kgdry-air)
n) h_{in}- Enthalpy of the internal air (kJ/kgdry-air)
o) t- System working time (h)
p) C_{pair}- Dry air specific heat capacity at constant pressure(1.006 kJ/kg K)
q) C_f- Water heat vaporization at 0◦C (2501 kJ/kg)
r) C_{pv}- Water vapour heat capacity (1.86 kJ/kg K)
s) T_{out}- Outside air temperature (◦C)
t) T_{in}- Inside air temperature (◦C)
u) w_{out}- Outside air specific humidity ($kg/kg_{dry-air}$)
v) w_{in}- Inside air specific humidity ($kg/kg_{dry-air}$)
w) Cc- Capital Cost of the Fan
x) Q- Quantity of the Fans
y) R- Rate of each Fan
z) Co- Operational Cost of the Chillers
aa) A_{TR}- Annual Tonnage consumption by the Chillers
bb) R_{unit}- Rate of Electricity per kilowatt

Chapter - 1

INTRODUCTION

The cost of energy is an increasingly important issue, and it is especially true for businesses. According to the U.S. Department of Energy, 40% to 60% of the energy used in industrial and commercial facilities is consumed through HVAC systems. Fossil fuels are the main source of primary energy consumption that make upward of 82% as per EIA, 2011. These resources of energy are scarce and contribute majorly to carbon dioxide (CO_2) emissions. The emissions are rising globally at more than 2% per year as per DOE, 2011. Global carbon dioxide emissions had exceeded 31 billion metric tons in 2010 with the contribution of US more than 5 billion metric tons as per EIA, 2011. This has attributed to increasing energy consumption and inefficient energy use.

The continued increase of the CO_2 emissions due to burning of fossil fuels such as coal, gas or oil into the atmosphere is predicted to lead to crucial changes in climate. Buildings have a long life span that last for minimum of 50 years or more and thus minimizing the energy consumption levels of buildings has a remarkable potential that contribute in mitigation of greenhouse gas concentrations for longer periods of time.

Recovering heat from the exhaust air can be considered whenever there is a controlled ventilation system is available. In fact, the fresh air is the ideal receiver available for the recovered heat. The flow rate is alike to the exhaust and its temperature levels (winter low and summer high temperatures) would always make it possible to employ the heat recovery.

1.1 HEAT RECOVERY WHEEL

Heat Recovery Wheels are a type of energy recovery heat exchangers that substantially decrease the energy needed to cool and heat ventilation make-up air. This technology is cost effective and has payback periods in the range of 1 to 3 years in most of the applications. The technology can be used effectively in any building that is tightly constructed to a reasonable amount and where the ducts of the return/ exhaust air are located close to the fresh make-up air intakes. They are made of thin metal, plastic, paper or ceramic surfaces, such as honeycomb or a random woven screen mesh, that create very large surface areas. They incorporate desiccant material which is typically silica gel or a molecular sieve (adhered to the matrix material), that enables total enthalpy transfer that is both mass (moisture) and heat transfer.

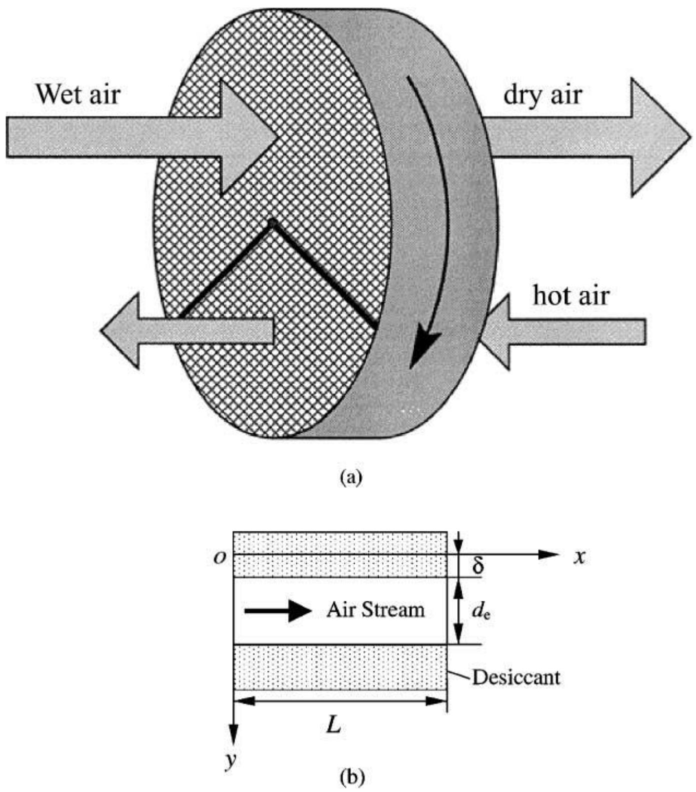

Fig. 1. Schematic of the desiccant wheel showing (a) the entire wheel, (b) a side view of one of the ducts.

Figure 1 shows the fundamental arrangement of the Heat Recovery Wheel wherein the exhaust air from the interior of the building passes through one side of the exchanger which is counter flow to the incoming make-up air that passes through the other side. During the cooling season, the cooler indoor air passes through the heat wheel and cools that portion of the wheel. When the cooled portion of the wheel rotates into the hotter outdoor air stream, it pre-cools the incoming outdoor air. The transfer of heat reverses during the heating season, i.e. the heat wheel transfers heat from the warmer indoor air to pre-heat the incoming outdoor air. The heat exchanger may transfer sensible heat only or it may transfer both sensible and latent heat.

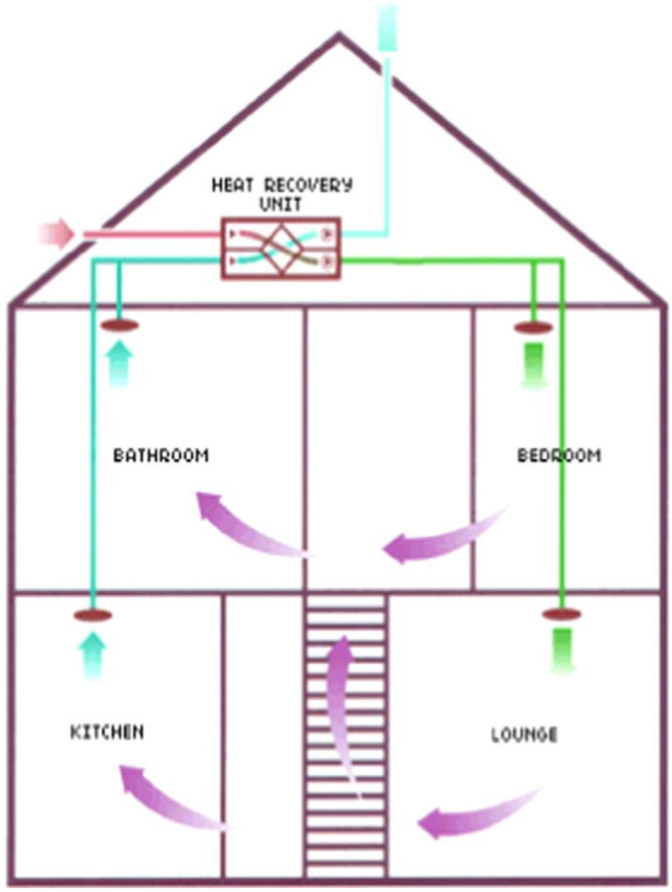

Fig. 2. Typical heat recovery system

Figure 2 shows a typical heat recovery schematic and how it works. It is necessary for the exhaust airflow to meet two key requirements:

a) The flow rate must be a significant fraction of the make-up air flow rate (more than, say, 75%)
b) The temperature and humidity of the exhaust air must be close to that of the conditioned space (i.e., heat loss or gain in the return or exhaust ductwork must be small).

1.2 ENERGY SAVING DUE TO HEAT RECOVERY

The amount of energy saved by installing a heat recovery device is equal to the energy recovered less the extra energy used in operating pumps, fans, etc. The final decision on installing heat recovery systems depends on its economic viability. As the cost of electricity is greater than the cost of fossil fuels, the heat recovery device will need to recover enough energy to economically justify its inclusion, while delivering a reasonable payback period. The advantage of heat recovery from economic point of view on the exhaust air depends on the volume and duration of the ventilation. The costs of equipment and installation are subjected to a scale factor so that the air unit treatment cost decreases for increase in size. Moreover as the annual utilization hours increase the payback period decreases. It is therefore more advantageous for long and severe winters or hot summers.

CHAPTER - 2
THEORY

2.1 THERMAL LOADS ON BUILDINGS

Leaving aside internal gains, there are two main categories of thermal loads of the buildings: Losses or gains through the envelope and the ventilation loads. As the air purity standards and fresh air supply rates are increasing, the ventilation loads are becoming more critical. As the insulation standards are also getting improved, this trend is also accelerating. Insulation is the most adapted technique to reduce thermal losses or gains. The technique available to reduce the ventilation loads is heat recovery which is achievable only when the building is equipped with a controlled ventilation system. Ventilation systems results in better control of room conditions.

A Heat Recovery Wheel is also known as a rotary air-to-air heat exchanger. It consists of a revolving wheel which is filled with an air-permeable medium that has a significant internal surface area. The wheel is mounted on a structure that supports it and also the motor that drives up to 20 revolutions per minute. The supply and exhaust air streams are adjacent to each other and flow through half of the wheel in a counter flow pattern.
The heat transfer medium and the type of heat transfer between the surface varies from manufacturer to manufacturer and can be configured to pick up ‘Sensible’ heat only or ‘Total’ heat (sensible and latent heat). The sensible heat is picked from the exhaust section of the warm room (exhaust air) and released into the outside air section (supply air) which is cold during winters. Since Heat Recovery Wheels are compact, counter-flow devices with small flow passages can achieve high heat transfer effectiveness.

2.2 EFFECTIVENESS OF HEAT RECOVERY WHEEL

Effectiveness is the ratio of the amount of energy transferred by the energy recovery device to the difference in energy levels of the two incoming airstreams. The total amount of energy transferred by the wheel is a function of the effectiveness of the wheel, the airflow volumes of the two airstreams and the difference in energy levels between the two airstreams. The below equation shows the calculation of effectiveness as defined by ASHRAE Standard 84-1991:

$\varepsilon = [V_s \times (x_1 - x_2)] / [V_{min} \times (x_1 - x_3)]$

Where:

ε = Sensible, or total effectiveness

x_1= OA temp (°Fdb) or enthalpy (btu/lb.)

x_2 = SA temp (°Fdb) or enthalpy (btu/lb.)

x_3 = RA temp. (°Fdb) or enthalpy (btu/lb.)

V_s = Supply (or outside) air volume (cfm)

V_{min} = The lower of the exhaust or supply air volume (cfm)

2.3 MODES OF CONTAMINATION

Air contamination, temperature of the exhaust air and properties of supply air influence the selection of materials for the casing, rotor structure and heat transfer medium for the Heat Recovery Wheel. The media of the exchanger can be fabricated from mineral, metal or synthetic materials that provide either random or directionally oriented flow across their structures. The contamination between the supply and exhaust streams of air can occur in a Heat Recovery Wheel by two ways:

a) Carry over
b) Seal leakage

a) **Carry over**- This occurs when the exhaust stream of air is carried into the supply air stream. This can happen each time the portion of the matrix passes through seals that divide the supply and exhaust air sections. This can be prevented by installation of a purge section on the heat exchanger. The introduction of a purge section reduces the level of carry over to reasonable limits, e.g. 0.1%, however it cannot completely eliminate it.

b) **Cross-leakage**- This occurs because of the pressure differential between air streams. Air leakage is driven from the region of high pressure to the region of low pressure. This can be reduced by avoiding huge pressure differentials, providing an effective seal, and the placement of fans such that it promotes leakage into the exhaust part of the air stream.

In most of the HVAC applications carry-over or cross-contamination is not a significant issue, however in applications that are critical such as clean rooms, laboratories or operation theatres stringent control of carry-over is required.

2.4 MODES OF CONTROL OF HEAT RECOVERY WHEEL

Two methods are generally employed to control the operation of a Heat Recovery Wheel:

a) **Supply air by-pass control**- In supply air by-pass control method a by-pass damper which is controlled by supply air temperature sensor regulates the volume of outside air allowed to pass through the rotating wheel and thus controls the temperature of the supply air.

b) **Rotary wheel speed control-** This method regulates the rate of heat recovery by changing the speed of the rotary wheel through a variable speed drive. Heat recovery increases with wheel speed but it also results in increase of carry over. Wheel speed is therefore limited by carry-over.

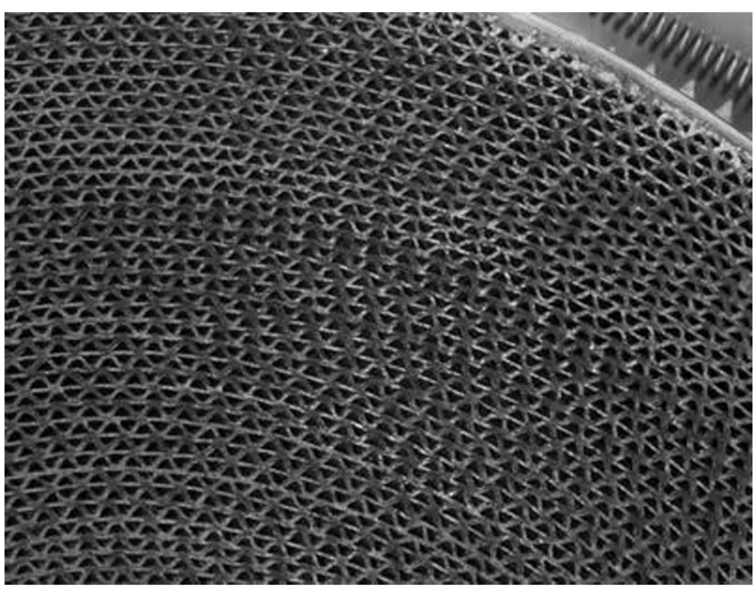

Fig. 3. Honeycomb structure of Heat Recovery Wheel

2.5 TECHNICAL PARAMETERS FOR DESIGNING

The Technical parameters that are considered during designing of the HVAC system with Heat Recovery Wheel are as follows;

a. The space requirement for the fitment of the Heat Recovery Wheel and its auxiliaries.
b. The distance between the supply and extract air streams.
c. The type of energy to be recovered ('Sensible' only or 'Total' Energy (Sensible and Latent)
d. The supply and extract air quantities (mass flow rates) in the system.
e. The 'Effectiveness' of the Heat Recovery Wheel.
f. The quality and condition requirements of the supply air stream (whether cross-contamination is acceptable? – consider the risk of cross contamination between exhaust air streams and supply air streams due to carry-over or leakage).
g. The quality and condition of the exhaust air stream (corrosive, dust laden, high temperature, high static pressure), leading to additional costs for anti-corrosion coatings, additional filtration, robust construction, etc.
h. Construction materials – consider corrosion, location, differential pressures, contaminants, etc.
i. Additional operating energy and service requirements for the Heat Recovery system, e.g. electrical supplies and condensate drains.
j. Modification requirements to existing plant and service routes to accommodate the new equipment.
k. Move-in space provision, i.e. consider the move-in path for a large item of equipment.
l. Construction costs.
m. Disruption to existing occupied areas, e.g. downtime.
n. Additional maintenance costs – filtration, cleaning, motor drives, etc.
o. Impact on the performance of existing air handling equipment, e.g. additional pressure drop resistance on existing fans.
p. Control method (additional controls and upgrading of existing BMS systems).
q. Condensation formation and frosting on the exhaust air side of the heat exchanger.

r. Maximum allowable pressure drop through the unit (particularly when the unit is being retrofitted to an existing system).
s. Air flow arrangements (plate heat exchangers – cross-flow or counter-flow)
t. Face velocities at the heat exchanger.

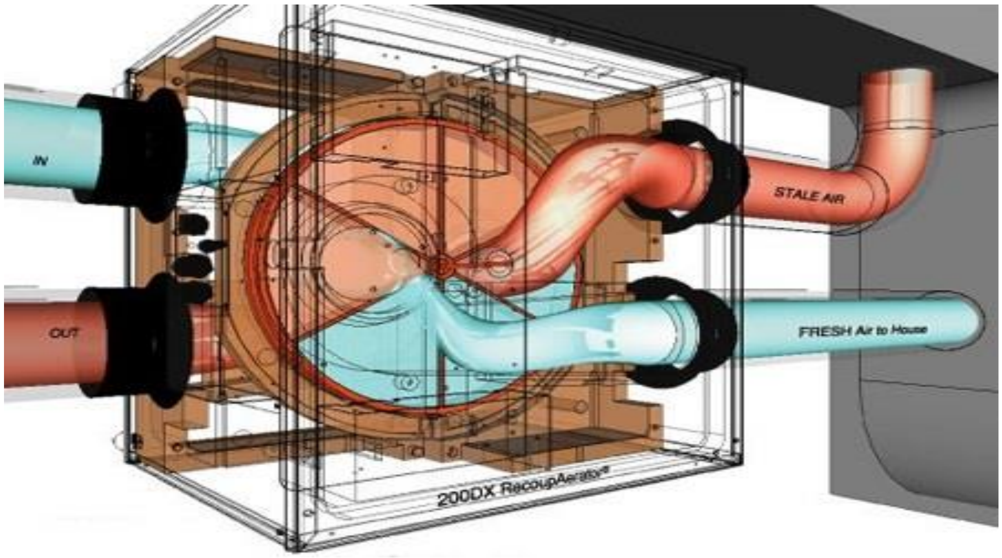

Fig. 4. Shows a three dimensional view of the Heat Recovery Wheel and the placing of Fresh and Return air duct.

2.6 ECONOMIC JUSTIFICATION

Assuming the technical requirements can be met, the final decision on the installation of a heat recovery device will be made on the grounds of its economic justification. This justification requires an assessment of the potential energy savings in comparison to the system capital cost and the increased operating costs. In order to carry out an assessment the following information is required:

a. The operating period for the equipment - annual hours run.
b. The outside air conditions (temperature and hours of occurrence) for the heat recovery season – Local Weather Data.
c. The extract air temperature.
d. The supply and extract air volume flow rates.

e. The ‘Effectiveness’ (efficiency) of the heat recovery device being considered. Effectiveness quoted in manufacturer’s data is usually based on a balanced air flow, i.e. equal supply and extract air volumes.
f. The cost of Electricity and Fossil Fuels.
g. The boiler efficiency
h. Supply and Extract Fan efficiencies (also include circulating pump efficiencies where a run around coil assembly is being considered)
i. The installed cost of the selected heat recovery equipment (including design costs, controls costs, pipework costs (run-around coils), electrical costs and the cost of any necessary modification to existing services).

The economic investment is considered justified if the assessment shows the capital cost investment can be recovered by the energy cost savings related to the installation of the heat recovery system, within a reasonable Payback Period. A reasonable Payback Period would be considered to be of the order of 1- 6 years for most businesses.

2.7 PAYBACK PERIOD

The payback period of the Heat Recovery Wheel increases due to the following:

a. The number of air changes taking place per hour and the length of the season increases.
b. The increase in the difference in the temperature of supply and extract air streams.
c. The supply and extract air streams are in close vicinity to each other.
d. The length of operation of the system. e.g. a system which operates 24 hours a day for seven days a week will yield higher savings than a system which operates ten hours a day for five days a week.

CHAPTER - 3

LITERATURE REVIEW

3.1 LITERATURE

The research papers published by other authors in this area are the best sources to gauge how much work has been done on this topic and what all can be done further to . These sources are authentic and reliable, and provide a global perspective over the selected topic. In this chapter, the important outcomes of the various research journals has been mentioned which were referred to during the working on this project. The various literature that have been taken as reference for this study are as follows:

Technological and economical analysis heat recovery in building ventilation systems

Renato M. Lazzarin, Andrea Gasparella [1]

Ventilation loads are becoming more important as air purity standards and thus fresh air supply rates are increasing. In Italy an Act (Act 9 gennaio 1991 n.10, Rules to actuate the New National Energy Plan as regards rational use of energy, energy saving and the development of renewable energy sources) obliges building operators to recover heat in ventilation systems whenever ventilation rates and annual working hours are higher than values determined by the climate zones.

Also, a minimum efficiency of 50% is prescribed for the heat recovery unit. Heat recovery in ventilation systems can be categorized in Sensible and Total heat recoveries. In sensible heat recovery only sensible heat is recovered whereas in total heat recovery both sensible and latent heat is recovered at the Heat Recovery Wheel. Latent heat recovery is not effective whenever the outside air humidity arrives at values higher than needed by the inlet air.

In economic analysis of the Heat Recovery Wheel, following costs are to be examined and analyzed to conclude:

a) Investment specific costs.
b) Operational costs

Performance comparisons of desiccant wheels for air dehumidification and enthalpy recovery

L.Z. Zhang, J.L. Niu [2]

In their research, the performances of the dessicant wheels for dehumidification and enthalpy recovery were compared with each other through a two dimensional, dual diffusion transient heat and mass transfer model. Their results showed that the ideal rotary speed is much less in case of air dehumidification than for enthalpy recovery. In both the cases, honeycomb-type structure is recommended as a thick wall of channel is harmful for heat mass transfers in solids. An NTU of 2.5 is required by desiccant wheels to show good performance. The enthalpy recovery wheel has more homogeneous temperature distribution and humidity fields in comparison to an enthalpy wheel for dehumidification. In dehumidification, desiccant structure near the warm air inlet has a larger temperature wave than that near the cool air inlet, but the amplitudes of humidity waves are identical at these two positions. The cycles in the psychometric chart show that for air dehumidification, four phases, i.e., isosteric heating, desorption, isosteric cooling and adsorption are present. However, for enthalpy recovery, the heating and the cooling processes can be represented by a single straight line, but in opposite directions.

Desiccant wheel regenerated by thermal energy from a microcogenerator: Experimental assessment of the performances

Giovanni Angrisani, Alfonso Capozzoli, Francesco Minichiello, Carlo Roselli a, Maurizio Sasso [3]

Experiments were carried out to analyze the performances of a silica gel desiccant wheel inserted in a test facility characterized by a desiccant air handling unit coupled to an electric chiller, a boiler and a microgeneraotor at temperatures less than seventy degrees centigrade. Further, an analysis was presented to evaluate the desiccant wheel performance in handling the ventilation and internal latent load for various cities around the world. The results showed that, for outdoor humidity ratio higher than about 15.5 g/kg, the desiccant wheel can balance at least the ventilation latent load only for low outdoor temperatures. In this way, the designer can easily evaluate the specific internal latent load for the analyzed indoor ambient and check if the silica gel desiccant wheel with low regeneration temperature is able to balance it, otherwise, higher regeneration temperatures are crucial.

Adequacy of air to air heat recovery ventilation system applied in low energy buildings.

Younness El Fouih, Pascal Stabat, Philippe Riviere, Phuong Hoang, Valerie Archambault [4]

In this study, different types of low energy buildings and different French climate zones were considered to analyse and test the performance of Heat Recovery Ventilation in function of design options. To evaluate the HRW adequacy, two low energy building categories were considered: Residential and Commercial. From the results obtained from simulation, it was concluded that the adequacy of heat recovery ventilation in low energy buildings varies with the building type, the heating load and the characteristics of the ventilation scheme.

It was observed that for office buildings the heat recovery ventilation was less efficient in comparison to mechanical ventilation primarily because the ventilation periods are reduced to day time only and also because the free heat gains inside the office building reduce the heating load and thus the potential for heat recovery.

In the case of residential buildings as the ventilation is performed all day long, the heat recovery ventilation is still competitive as compared to mechanical ventilation.

Modeling analysis of an enthalpy recovery wheel with purge air

Wei Ruan, Ming Qu, W. Travis Horton [5]

A purge section is typically used to reduce the carryover of contaminants in enthalpy recovery wheels. A one-dimensional transient heat and mass transfer model was developed to analyze the performance of enthalpy recovery wheels both with and without purge air. After making some assumptions to simplify the governing equations, trial and error method was used to solve the variables. The performance of the enthalpy recovery wheel with purge air was studied under summer and winter design conditions. The profiles of the air temperature and the humidity ratio in the enthalpy recovery wheel showed that the wheel depth and rotation speed have a positive influence on the performance of the wheel. However, increasing the wheel depth and the rotation speed require more purge air to prevent the carryover of contaminants to the supply air stream, which will degrade the wheel performance.

A review of different strategies for HVAC energy saving

Vahid Vakiloroaya, Bijan Samali, Ahmad Fakhar, Kambiz Pishghadam [6]

The main objective of this study was to do an in depth study of the various technologies that can reduce the HVAC energy consumption of the buildings. The various technologies that were studied in this paper are as follows:

a) Evaporative cooling
b) Ground coupled HVAC system
c) Thermal storage systems
d) Heat recovery sytems

Rotary Wheel heat exchangers can be used for comfort to comfort heat recovery systems. An experimental analysis was carried out by Fernandez-Seara et al on an air-to-air heat recovery unit equipped with a sensible polymer plate heat exchanger for ventilation systems in residential buildings. As a part of their results, the heat transfer rate increased almost linearly as the air flow rate increased. The heat transfer rate increased around 65% by increasing the air flow rate from 50 m3/h to 175 m3/h. The impact of energy recovery ventilator on annual cooling and heating energy consumption was investigated for a 10 story-office building by Rasouli et al. Their results indicated that up to 20% and 40% annual cooling and heating energy consumption can respectively be saved using the system.

Investigation on a ventilation heat recovery exchanger: Modeling and experimental validation in dry and partially wet conditions.

Samuel Gendebien, Stephane Bertagnolio, Vincent Lemort [7]

The aim of their study was to investigate the air to air heat recovery device through experimental approaches and modeling. A comparison between prognosis by the model and computations was conducted. Hydraulic performance was predicted by the model developed with a reasonable accuracy (the range of the mean deviation between the measurement and the prediction was 4.6%).

The thermal performance results were also of the same order of magnitude (the absolute mean deviation was 5.2% in terms of its effectiveness). The variable boundary model that was developed seemed to be able to predict with a good accuracy of the total heat transfer rate (within 5% of error) as well as latent (within 10% of error) and sensible (within 3% of error) ones.

An optimal design analysis method for heat recovery devices in building applications

X.P. Liu, J.L. Niu [8]

They developed a new optimization method for analysing the heat recovery potential of the various heat recovering devices in building applications. Keeping in mind the essential characteristics when heat recovery is employed in building applications, that objective of the new optimization method was that at any given mass flow rate, temperature difference and desired heat recovery effectiveness, minimizing the material cost at a specified fan energy use, or alternatively, minimizing the fan energy use at a given material cost. Several standard shaped duct geometries were analysed. The duct geometries used were: equilateral triangle (Tri), circular (Cyl), square (Squ), rectangle with aspect ratio 1/2 (Rec(1/2)), 1/4 (Rec(1/4)), and 1/8 (Rec(1/8)). A novel channel structure named cross-corrugated triangular (CCT) duct developed by Zhang and Chen [34] was also considered for comparisons. During the analysis, the hydraulic diameters for all the compared shapes were varied from 5 mm to 20 mm. The fan power requirements for Rec(1/8) were found to be the lowest when compared with the other shapes in the laminar flow region under the same hydraulic diameter. The values for the CCT duct were the highest which indicates larger energy consumption when novel structure is used. Conversely, with a specified fan power consumption, the required total surface areas of Rec(1/8) is the smallest, which shows that it is the best geometry from material cost saving point of view.

Review of heat/energy recovery exchangers for use in ZEBs in cold climate countries.

Maria Justo Alonso, Peng Liu, Hans M. Mathisen, Gaoming Ge, Carey Simonson [9]

A comprehensive study of the following heat recovery systems used in building ventilation were studied:

a) **Flat plate heat exchanger**- Plate heat exchangers are a well-known technology. Their effectiveness is high and when enthalpy recovery is also used, they rarely have frosting problems. They are also not likely to transfer odours since both air streams are separated by the heat/ energy exchange layer.

b) **Heat recovery wheel-** In cold climate energy wheels could improve the humidity level of the indoor air. They are distinguished from other heat exchangers for their high effectiveness ranging from 50 to 90% and large internal surface areas. The biggest concern to be taken into account with this technology is a certain risk of freezing for heat wheels and the leakages.

Their summary for the other types of heat recovering devices is tabulated below:

Table 1. Effectiveness of various types of heat recovering units.

Type of exchanger	Freezing problems	Odour transference	Moving parts	Latent effectiveness	Sensible effectiveness	Use in toxicenvironments
Flat plate heat exchanger cross flow	Yes	No	No	No	60–80%	Yes
Flat plate heat exchanger counter- flow	Yes	No	No	No	70–90%	Yes if good sealing
Flat plate membrane quasi counter flow	No	No	No	46–76% [46]	80–85% [22]	Yes
Heat wheels	Moderate problems	Yes	Yes	No	50–80%	No
Energy wheels	No	Yes	Yes	50–85% [2]	50–85% [2]	No
Dessicant drying wheels	No	Yes	yes but low rotational speed	70–90% [5]	95% [5]	No
Run around heat exchangers	Yes	No	No	No	65–70%	Yes
Run around membrane energy exchangers (only preliminary experimental data [32])	No	No	No	50–65% [32]	60–80% [32]	Yes

A Study of Passive Ventilation Integrated with Heat Recovery

Dominic O' Connor, John Kaiser Calautit, Ben Richard Hughes [10]

In this experiment a 1:10 scale prototype of the system was created and tested experimentally in a closed-loop subsonic wind tunnel to validate the Computational Fluid Dynamics (CFD) investigation. A Heat Recovery Wheel was proposed in the wind tower and the experiments were carried out. The wind tower with rotary thermal wheel was capable of meeting the guideline ventilation rates above an inlet air velocity of 3m/s for a standard occupancy density of 1.8 square metre per person. It was seen that the air supply rates through the wind tower with rotary thermal wheel were approximately one third of the values from the wind tower which correlates to the porosity of the thermal wheel of 70%, added to the reduced area of the casing and this shows that the rotary thermal wheel does not have a significant effect on the air flow through the wind tower and into the room model.

Design Optimization of Heat Wheels for Energy Recovery in HVAC Systems

Stefano De Antonellis, Manuel Intini, Cesare Maria Joppolo and Calogero Leone [11]

In this paper a thorough optimization of the various design parameters of heat wheels was performed in order to maximize the sensible effectiveness and also to the minimize pressure drop across the heat recovery wheel. A one dimensional lumped heat wheel model was used to analyse, which solved heat and mass transfer equations. Appropriate correlations were used to estimate the pressure drop. The following parameters were taken into consideration:

a) Wheel length
b) Channel base
c) Height and thickness
d) Air face velocity
e) Revolution speed

The following considerations were highlighted by them as conclusion to their study:

a) Small channel thickness is best suited configuration. (for example s = 0.05 mm)
b) The revolution speed of the wheel does not affect the wheel performance much so it should be kept between 10 to 15 revolutions per minute.
c) Large channel hydraulic diameters should be adopted.
d) If the Heat wheels are optimized through length, then they do not require more matrix material. However, if they are optimized through reduction of the channel hydraulic diameter then additional material is required.

Performance comparisons of honeycomb-type adsorbent beds (wheels) for air dehumidification with various desiccant wall materials.

Li-Zhi Zhang, Huang-Xi Fu, Qi-Rong Yang, Jian-Chang Xu [12]

Their study aimed at comparing the performance of honeycomb type adsorbent beds (or desiccant wheels) for air dehumidification with various other solid desiccant wall materials, in view of system operation. A mathematical model was proposed and validated to predict the cyclic behaviours of the cycling beds or wheels. The influences of regeneration air temperature, process air temperature, and humidity on the coefficient of performance (*COP*), specific dehumidification power (*SDP*) and dehumidification efficiency (ε_d) were predicted with various desiccant wall materials. In total ten most commonly used desiccant materials were considered, with different adsorption and thermo physical properties. It was found that of the 10 materials, the silica gel 3A and silica gel RD performed better than other desiccants for air dehumidification under typical working conditions and driven by low grade waste heat. The results provided some insights and guidelines for the design and optimization of honeycomb type adsorption beds or desiccant wheels.

Experimental Performance Analysis of a Heat Recovery System for Mechanical Ventilation in Buildings

Francesco Asdrubali, Giorgio Baldinelli, Francesco Bianchi, Matteo Cornicchia [13]

An experimental test bench was built at the Thermal Engineering Laboratory of the University of Perugia, aimed at evaluating the performance of air heat recovery devices. The first measurements were carried out on a commercial plate-type heat exchanger, made of polystyrene. This plastic material is characterized by a low value of thermal conductivity, but its easiness of workability allows to increase the heat exchange surface, overcoming also issues linked to the weight and the cost of the product. The flow-rates, the pressure drops, and all temperatures of interest for the heat exchanger were acquired. The energy efficiency index of the heat recovery system was assessed with several tests conducted with different boundary conditions of the indoor and outdoor ambient, as well as different air flow rates. A graph between the efficiency of the system and mass flow rate was plotted as below:

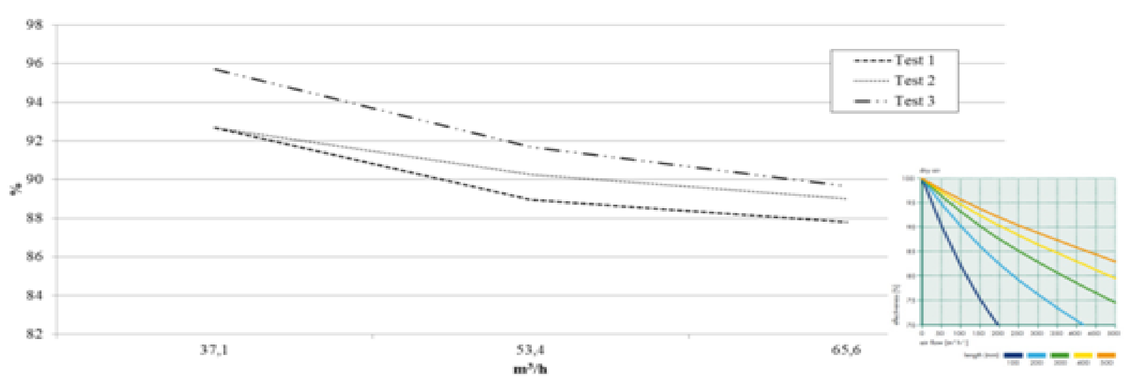

Fig. 5 Efficiency of the system with different mass flow rates and the efficiency trend according to manufacturer.

Analysis of mechanical ventilation system with heat recovery in renovated apartment buildings

Edite Kamendere, Gatis Zogla, Agris Kamenders, Janis Ikaunieks, Claudio Rochas [14]

Two buildings were analysed in their study. The main objective of their work was to assess the efficiency of mechanical ventilation systems with heat recovery in renovated multi-family residential buildings. The buildings were located adjacent to each other in the city of Cesis in Latvia. The average heating season in Cesis is 208 days long with an average outside temperature of -1.1 degree centigrade. Both the buildings were built in 1974 and renovated together with the only difference being one was equipped with heat recovery unit in mechanical ventilation whereas the other was with natural ventilation. The monitoring system in the two similar residential apartment buildings was set and monitoring was performed in April, 2014. Field measurements showed that the heat recovery energy efficiency of the air-handling unit for supply air is equal to 77 %. It is a good result if it is compared to the given efficiency in the air handling unit's Passive House certificate which is 75 %. The regression analysis showed that the heat exchanger's thermal efficiency is dependent on ambient conditions.

A comprehensive review of heat recovery systems for building applications

Pinar MertCuce, SaffaRiffat [15]

In this paper, a comprehensive review on building applications of heat recovery systems is presented along with evaluation of their environmental impacts. The current cost of the heat recovery systems completely depends on the technology that they have. The payback period of them varies from a couple of years to 15 years with respect to the heat recovery technology and their lifetime is mostly greater than 30 years. A major problem for the air-to-air heat recovery systems is the disturbing noise due to the fans of the ventilation unit. Recent works indicate that heat recovery panels can be modified to work in natural or hybrid ventilation systems in order to eliminate the noise. There are many different type of heat recovery systems in the market today. However, as previously reported by Mardiana-Idayu and Riffat, heat pipe, fixed plate, run-around and rotary wheel heat exchangers are the most common types and rotary wheel heat recovery systems still have the highest efficiency as tabulated below:

Table 2 Advantages of various heat recovering units.

A comparison of heat recovery types by their efficiency ranges and potential advantages [9].

System type	System efficiency	Advantages
Fixed-plate	50–80%	Compact, highly efficient due to high heat transfer coefficient, no cross contamination, can be coupled with counter-current flow which enabling to produce close en-temperature differences.
Heat pipe	45–55%	No moving parts, no external power requirements, high reliability, no cross contamination, compact, suitable for naturally ventilated building, fully reversible, easy cleaning.
Rotary-wheel	Above 80%	High efficiency, capability of recovering sensible and latent heat.
Run-around	45–65%	Does not require the supply and exhaust air ducts to be located side by side, supply and exhaust duct can be physically separated, no cross contamination.

The results from the literature analysis indicate that heat recovery systems have a remarkable potential to mitigate the energy demand of buildings, and thus the greenhouse gas emissions in the atmosphere.

Theoretical energy saving analysis of air conditioning system using heat pipe heat exchanger for Indian climatic zones.

T.S. Jadhav, M.M. Lele b [16]

Heat pipe heat exchanger (HPHX) is an excellent device used for heat recovery in air conditioning systems. The Energy Conservation Building Code (ECBC)- Bureau of Energy Efficiency (BEE) India classifies Indian climatic zones into five categories viz., hot and dry (e.g. Ahmedabad, Jodhpur etc), warm and humid (e.g. Mumbai, Chennai etc), composite (e.g. Nagpur, Jaipur etc), cold (e.g. Guwahati etc) and temperate (e.g. Bengaluru etc). The analysis was carried out for total 25 Indian cities representing different climatic zones.

The maximum energy saving potential was revealed for hot and dry, warm and humid and composite Indian climatic zones. The energy savings varied with the variation in the input parameters such as outdoor air quantity, outdoor air temperature, return air temperature, compressor power consumption and plant operating hours. However, the energy savings were significant where the air conditioning plant was operating for more than 12 h. This showed the tremendous potential of heat recovery systems in building ventilation and air conditioning.

A novel design of a desiccant rotary wheel for passive ventilation applications

Dominic O’Connor, John Kaiser Calautit, Ben Richard Hughes [17]

They conceptualised, designed and tested a new structure for Heat Recovery Wheels. The traditional honeycomb/sinusoidal wave matrix structure of the desiccant wheel was replaced with blades which extend out from the centre of the wheel. It was visualised that a high level of moisture adsorption could be achieved and also lower the regeneration temperature and pressure drop across the wheel due to the large openings between the blades. Adsorption of moisture in the inlet airstream up to 65% was noted whilst increasing the air temperature of the inlet air by 9.6 degrees centigrade.

Furthermore, constant regeneration of the desiccant material was achieved at a regeneration temperature of 48.5 degrees centigrade, significantly lower than regeneration temperatures commonly used in desiccant systems. The pressure drop across the desiccant rotary wheel was measured as 2.06 pascals, lower than the pressure drop across the matrix of traditional desiccant rotary wheel designs.

Evaluation of the potential energy recovery for ventilation air in dwellings in the South of Europe

Silvia Guillén-Lambeaa, Beatriz Rodríguez-Soriaa, José M. Marín [18]

This study was made to analyze the climate data and see which areas require recovery of heat only and which areas require recovery of latent energy. Their study was restricted to many cities in southern Europe and the results were compared to the northern cities of Europe.

It was seen that in winter season the cities located in the Canary Islands have remarkable potential to recover latent energy. For the rest of the cities the recovery potential of latent energy was very less as compared to the sensible heat. The areas near the south of the Mediterranean coast should preferably use latent recovery system in place of a sensible one.

Research on heat recovery technology for reducing the energy consumption of dedicated ventilation systems: An application to the operating model of a laboratory

Lian Zhang, and Yu Feng Zhang [19]

In their research in China, the provision of heat pipes in the air handling unit was considered so as to decouple dehumidification from cooling so as to decrease the consumption of energy. This was investigated by experimental studies and simulations. The results showed that the heat pipes can be used to save over 80% of the energy during its hours of operation along with the air conditioning. The reduction in the overall rate was from 3.2% to 4.5%. It was also established that the energy saving capability in a laboratory is higher than the other kinds of buildings. It was concluded that the dedicated ventilation system combined with heat recovery technology can be applied to buildings especially in laboratories in subtropical areas. They also concluded that using heat recovery combined with the ventilation system is possible and efficient in saving energy thereby reducing the energy consumption of air conditioning systems.

3.2 GAPS IN LITERATURE

While studying the Literature Review it has been observed that no real time analysis has been done with respect to the Heat Recovery Wheels. Various studies have been done on the following:

a) Structure of Heat Recovery Wheel
b) Efficiency of Heat Recovery Wheel
c) Use in ZEB's in cold climates
d) Modeling and experimental validation in dry and partially wet conditions.
e) Modeling with purge air

The various studies performed till now do give us insights about the ideal/ theoretical performance of the Heat Recovery Wheel but none has done a real time analysis with respect to a particular building where all other parameters have been designed and calculated. A real time study of a given building will give us the actual performance of the Heat Recovery Wheel and give us the practical savings that can be incurred by the use of it and also apprise us about the payback period of the device and its auxiliaries.

3.2 OBJECTIVE OF THE PROJECT

The objective of this project is to calculate the real time Payback Period of an Office Building in Kolkata and thus to determine the potential that Heat Recovery Wheels have in the application of mechanical ventilation. A study shall be done to calculate the various costs that would have to be incurred if the Heat Recovery Wheel is used in the mechanical ventilation of the building under consideration. The next step would be to determine the operational costs in both the cases and see whether the Heat recovery Wheel is feasible. The feasibility of the Wheel shall depend on its Payback period which shall be calculated.

CHAPTER – 4

WORKING

4.1 WEATHER IN KOLKATA

The Office building in consideration for which the Heat Recovery Wheel analysis is being done is situated in Kolkata. Kolkata is characterized by tropical wet and dry climate. It is situated near the subtropical moist forest biome. The following are the characteristics of the weather of Kolkata:

a) The annual mean temperature is 26.9 degrees Celsius.
b) Average monthly temperatures vary by 10.6 degree Celsius.
c) Total annual precipitation averages 1800 mm.
d) On average there are 2609 hours of sunshine per year.

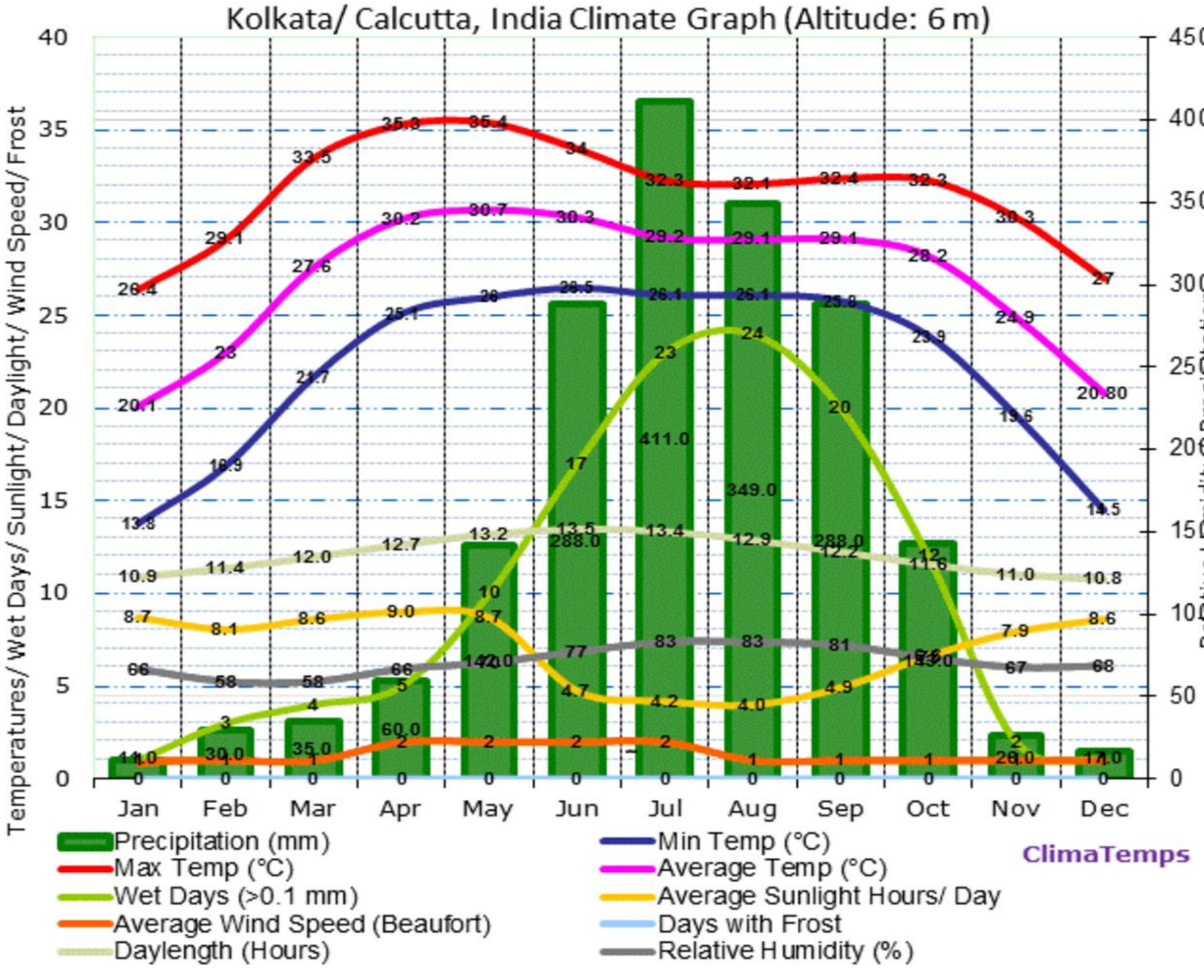

Fig.6 Characteristics of Kolkata weather

4.2 OFFICE BUILDING CONFIGURATION

The structure of the Office building in Kolkata is as follows:

a) 2 Basements used for car parking and services area.

b) Ground floor used for car parking, gymnasium and services area.

c) First floor that comprises of the food court.

d) Second to fourteenth floor is typical and has office spaces.

e) Terrace which is used for services.

The built up area of the building excluding the basements is approximately 5,00,000 square feet. The technical drawings below show the building layout of a typical floor and the section/ orientation of the building.

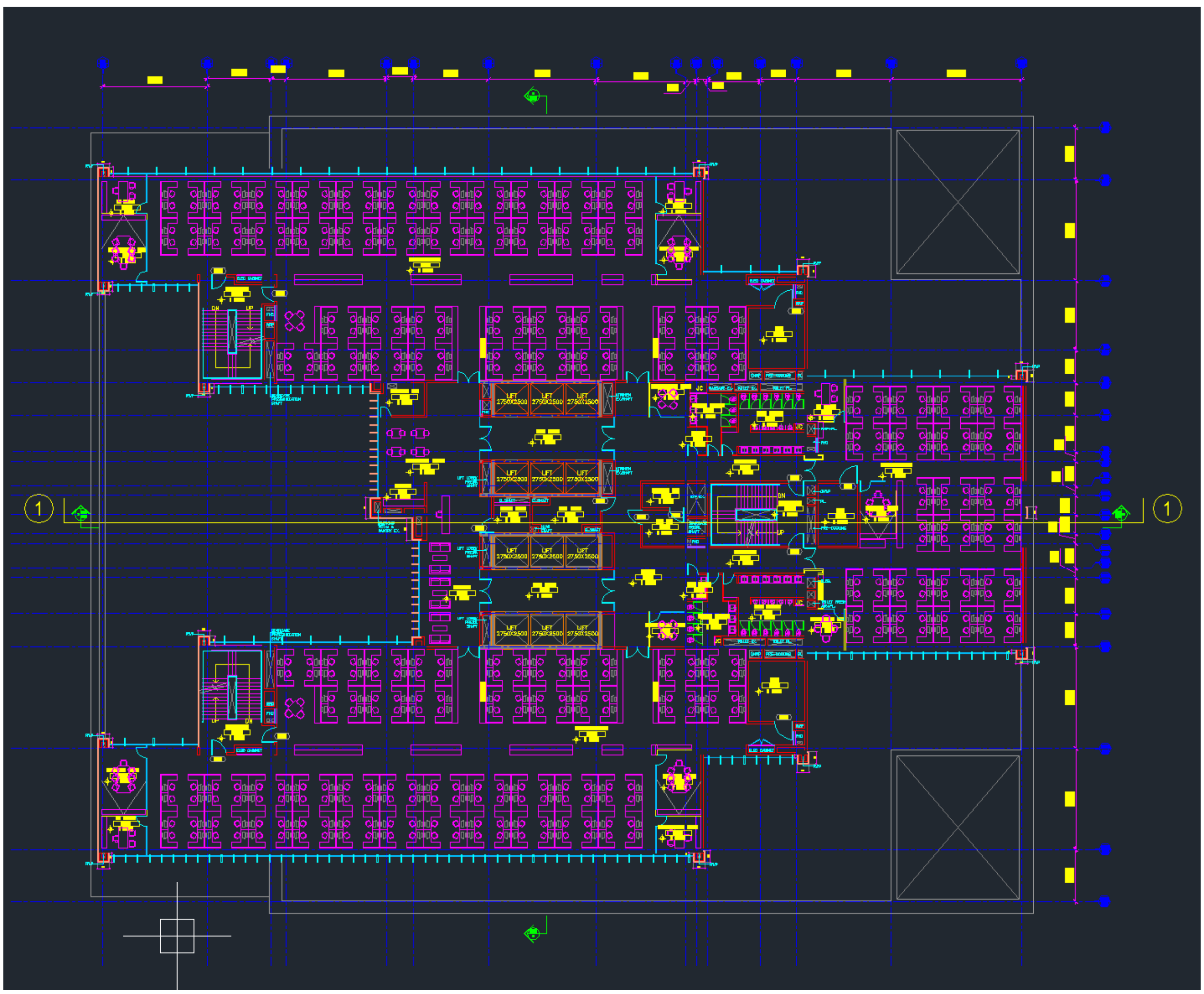

Fig. 7 Typical floor layout of Office building

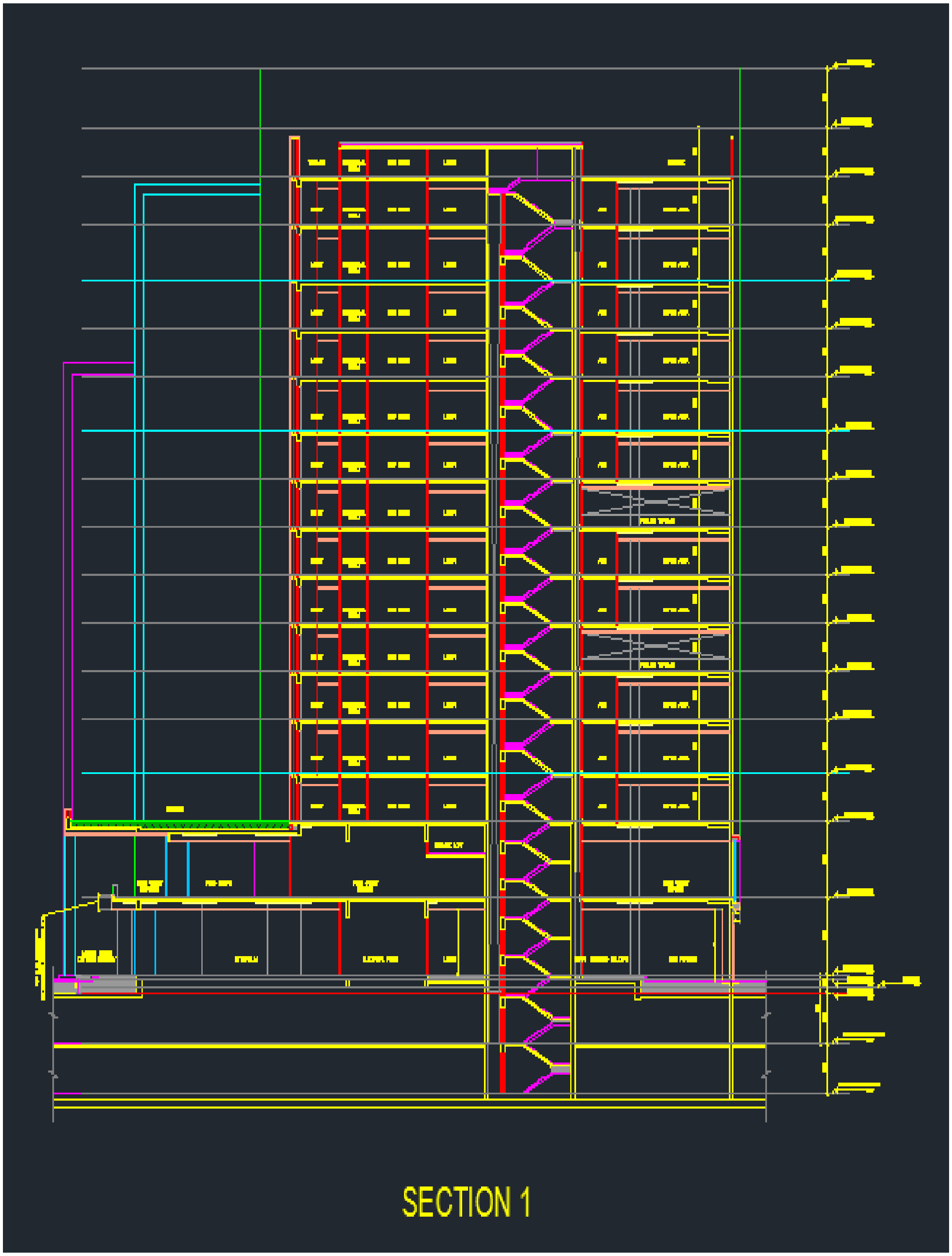

Fig. 8 Section of the building

4.3 CALCULATIONS ON LOAD

The weather bin data was taken from ISHRAE 2014 for Kolkata and the calculations were done for each hour in a year. The weather bin gives us the dry bulb temperature, relative humidity and moisture for each hour of a year. The following parameters were considered for the calculations taken from the psychometric chart or

a) Return air temperature – 80 (F)
b) Return air moisture – 76.6 (grain/ pounds)
c) Off coil enthalpy (cooling of coil at 55 deg F/ 98% RH) – 23.1 (btu/ pound)
d) Return air relative humidity – 55%
e) Wheel effectiveness (sensible) – 75%
f) Wheel effectiveness (latent) – 70%

The equations of mass and heat transfer that are summarized below were used to do the calculations of the condition of the air after it leaves from the Heat Recovery Wheel.

$$\mathbf{E = m_{vent} \times (h_{out} - h_{in})\, t} \qquad (1)$$

Where,

E is the total energy demand due to the ventilation air (kJ)

m_{ven} is the mass flow of ventilation dry air (kgdry-air/h)

h_{out} is the enthalpy of the outside air (kJ/kgdry-air)

h_{in} is the enthalpy of the internal air (kJ/kgdry-air)

t is the system working time (h)

$$\mathbf{h = Cpair \times T + w\,(Cf + Cpv T)} \qquad (2)$$

Where,

C_{pair} is the dry air specific heat capacity at constant pressure(1.006 kJ/kg K)

C_f is the water heat vaporization at 0◦C (2501 kJ/kg)

C_{pv} is the water vapour heat capacity (1.86 kJ/kg K)

$$\mathbf{E = m_{vent} \times ((C_{pair} + C_{pv} \times w_{in}) \times (T_{out} - T_{in}) + (C_f + C_{pv} \times T_{out}) \times (wout - w_{in}) \times t} \qquad (3)$$

Where,

T_{out} is the outside air temperature (◦C)

T_{in} is the inside air temperature (◦C)

w_{out} is the outside air specific humidity ($kg/kg_{dry\text{-}air}$)

w_{in} is the inside air specific humidity ($kg/kg_{dry\text{-}air}$)

There is a sensible thermal load due to the temperature change and a latent thermal load due to the change of humidity. These values correspond to the two terms in Eq. (3), resulting in Eq. (4) and Eq. (5), respectively.

$$\mathbf{Q_s = m_{vent} \times (C_{pair} + C_{pv} \times w_{in})\,(T_{out} - T_{in}) = m_{vent} \times C_{pair}\,(T_{out} - T_{in}) \times t} \qquad \mathbf{(4)}$$

$$\mathbf{Q_1 = m_{sup} \times (C_f + C_{pv} \times w_{out})\,(w_{out} - w_{in}) \times t = m_{vent} \times C_f\,(w_{out} - w_{in}) \times t} \qquad \mathbf{(5)}$$

Table 3. Characteristics of exhaust air after Heat recovery Wheel.

Month	Day	Hour	DBT (°F)	RH (%)	Moisture (gr/lb)	Moisture (lb/lb)	Enthalpy (btu/lbs)	DBT (°F)	Moisture (gr/lb)	lbs/lbs	Enthalpy (btu/lbs)	Btu/h	Btu/h	Btu/h
			Before wheel Enthalpy conditions					After wheel Enthalpy conditions				Fresh air load w/o Wheel	Fresh air load with Wheel	Saving
04	13	01	81.70	87.00	143.08	0.0204	40.50	80	97	0.0138	33.49	77	46	31
04	13	02	81.00	88.00	141.82	0.0203	40.14	80	96	0.0137	33.40	76	46	30
04	13	03	79.90	92.00	142.45	0.0204	39.96	80	96	0.0138	33.35	75	46	29
04	13	04	79.50	94.00	144.20	0.0206	40.11	80	97	0.0138	33.41	76	46	30
04	13	05	79.90	91.00	141.54	0.0202	39.83	80	96	0.0137	33.32	74	45	29
04	13	06	82.00	85.00	142.66	0.0204	40.52	81	96	0.0138	33.50	78	46	31
04	13	07	84.60	80.00	145.73	0.0208	41.62	81	97	0.0139	33.79	82	48	35
04	13	08	87.80	74.00	147.69	0.0211	42.71	82	98	0.0140	34.07	87	49	38
04	13	09	90.10	69.00	150.21	0.0215	43.66	83	99	0.0141	34.33	91	50	42
04	13	10	92.50	65.00	150.21	0.0215	44.27	83	99	0.0141	34.48	94	51	44
04	13	11	94.30	61.00	150.00	0.0214	44.69	84	99	0.0141	34.58	96	51	45
04	13	12	96.40	56.00	147.90	0.0211	44.92	84	98	0.0140	34.62	97	51	46
04	13	13	97.70	53.00	146.08	0.0209	44.99	84	97	0.0139	34.62	97	51	46
04	13	14	98.60	51.00	143.36	0.0205	44.82	85	97	0.0138	34.56	97	51	46
04	13	15	97.50	52.00	141.33	0.0202	44.25	84	96	0.0137	34.41	94	50	44
04	13	16	95.50	54.00	139.44	0.0199	43.47	84	95	0.0136	34.20	91	49	41
04	13	17	92.50	59.00	138.11	0.0197	42.52	83	95	0.0136	33.96	86	48	38
04	13	18	90.00	64.00	137.20	0.0196	41.76	83	95	0.0135	33.76	83	47	36
04	13	19	87.30	69.00	136.71	0.0195	41.00	82	95	0.0135	33.57	80	47	33
04	13	20	86.00	73.00	138.74	0.0198	40.97	82	95	0.0136	33.58	80	47	33
04	13	21	84.70	78.00	141.96	0.0203	41.10	81	96	0.0137	33.63	80	47	33
04	13	22	84.40	81.00	146.22	0.0209	41.64	81	97	0.0139	33.80	82	48	35
04	13	23	83.80	84.00	149.51	0.0214	41.96	81	98	0.0141	33.90	84	48	36
04	13	24	83.70	85.00	149.72	0.0214	41.96	81	99	0.0141	33.91	84	48	36

`

The above table computes the enthalpy conditions of the air after it exits from the Heat Recovery Wheel. Similarly, a computation was done for each hour in a year and the Fresh air load with and without the Heat Recovery Wheel was calculated. The total Fresh air load without Heat Recovery Wheel comes out to be 5,28,424 btu/ hr whereas the Fresh air load with the heat Recovery Wheel comes out to be 3,55,719 btu/hr. Therefore, a total saving of 1,92,339 btu/hr was envisaged from the working.

4.4 CALCULATIONS ON PAYBACK PERIOD

An analysis was done on Total Fresh Air with and without the Heat Recovery Wheel keeping in mind the Capital and Operational costs. The Base case is with the Heat Recovery Wheel whereas Case-1 summarises without Heat Recovery Wheel. Table 4 below summarises the Capital costs of both the cases.

To calculate the Capital Cost of the fans the following equation has been used:

$$\mathbf{C_c = Q \times R} \qquad \textbf{(6)}$$

Where,

C_c is the Capital Cost of the Fan

Q is the Quantity of the Fans

R is the Rate of each Fan

The below table summarizes the Capital Cost of the fans that have been used in the Building under consideration. It also includes the cost of the Starters that are used for the Variable Frequency Drive in the fans.

`

Table 4. Capital costs of Fresh Air System with and without Heat Recovery Wheel.

Fresh Air Analysis - Capital Cost				
	Description	**Qty**	**Rate**	**Amount (in Lakhs)**
Base Case	**TFA Unit with HRW (all Zones)**			
Zone 1	6500	2	1135370	23
Zone 2	7700	1	1195264	12
Zone 3	7200	2	1195264	24
	VFD Starters (Total 60 kW	10	@ Rs 75000 / starter	8
			Total	**66**
Case 1	**TFA Unit without HRW (All Zones)**			
Zone 1	6500	2	@ Rs 40 / cfm	5
Zone 2	7700	1	@ Rs 40 / cfm	3
Zone 3	7200	2	@ Rs 40 / cfm	6
	VFD Starters (Total 28 kW	5	@ Rs 75000 / starter	4
Fan Sections				
For Exhaust	12000	1	@ Rs 15 / cfm	2
	14500	1	@ Rs 15 / cfm	2
	VFD Starters (Total 10 kW	2	@ Rs 75000 / starter	2
			Total	**23**

Next we come to the operational costs. As we see from above the above table that the Capital costs of Heat Recovery Wheel are high so to the operational costs of the same should be drastically less so that the overall system is justified and economically feasible. The less the payback period of the system, more economically viable it would be. In case of the operational costs of the chillers, the below equation has been used.

$$\mathbf{C_o = A_{TR} \times 0.68 \times R_{unit}} \qquad \mathbf{(7)}$$

Where,

C_o = Operational Cost of the Chillers

A_{TR} = Annual Tonnage consumption by the Chillers

R_{unit} = Rate of Electricity per kilowatt

The below table summarizes the Operational Costs of the Chillers with and without the use of the Heat Recovery Wheel.

`

Table 5. Operation costs of Fresh Air System with and without Heat Recovery Wheel.

Fresh Air Analysis - Operational Cost (On Chillers)								
	Description	Annual BTU/hr/cfm consumption*	Annual TR/cfm consumption	Annual TR consumption	Annual Consumption (kW)	Rate (on State Electricity)	Rate (on DG)	Annual Operating Cost (in Lakhs)
Base Case	**TFA Unit with HRW**	355719	30	1040478	707525	7	15	**55**
Case 1	**TFA Unit without HRW**	528424	44	1545640	1051035	7	15	**82**
***This value is as per annual fresh air energy load analysis based on Kolkata Weather Bin.**								

Due to the use of the Heat Recovery Wheel, there is additional electrical load as the capacities of the fan motors change. This should also be kept in mind while calculating the Total Operational Costs. The fan motors have been equipped with the Variable Frequency Drives so as to limit the operational costs and unnecessary working of the fans even when the load is low. The following assumptions have been taken while calculating their impact:

a) The fan shall work at 100% of the capacity for 4 hours in a day.
b) The fan shall work at 75% of the capacity for 6 hours in a day.
c) The fan shall work at 50% of the capacity for 6 hours in a day.
d) The fan shall work at 25% of the capacity for 8 hours in a day.
e) The monthly working days are 30.
f) Diversity as 0.75
g) The charges for running on State Electricity are Rs 7 per Kilowatt per hour.
h) The charges for running on Diesel Generator Sets are Rs 15 per Kilowatt per hour.
i) The System shall work on State Electricity for 90% of the time and 10% on the Diesel Generating Sets

`

The below table summarizes the impact of the increase in the fan motor capacity on the Operational Costs.

Table 6. Operational Cost of Fan Motors

Fresh Air Analysis - Operational Cost (On Fan Motors)													
	Description	kW	% operation	Hours	Daily Usage (kW)	Days	Months	Annual Consumption	Diversity	Rate (on State Electricity)	Rate (on DG)	Annual Operating Cost (in Lakhs)
Base Case	TFA Unit with HRW	60	100%	4	240							
			75%	6	270							
			50%	6	180							
			25%	8	120							
					810	30	12	291600	0.75	7	15	**17**
Case 1	TFA Unit without HRW	38	100%	4	152							
			75%	6	171							
			50%	6	114							
			25%	8	76							
					513	30	12	184680	0.75	7	15	**11**

The below table summarizes the Total Payback Period of the Heat Recovery Wheel.

Table 7. Payback period of Heat Recovery Wheel.

Total Operation Cost	
	Amount (in lakhs)
Base Case	72
Case 1	93
	Payback Period (Years)
Base Case vs Case 1	2.1

`

The Payback period depends upon assessing the Final Capital and Operational Costs. As we have seen above that the Operational Costs are less of the System with Heat Recovery Wheel our next aim is to calculate the Payback period in years for the whole system to be economically viable. The above table summarizes our calculations and shows the Payback period of the System with Heat Recovery Wheel. It can be seen from the table that the Payback period is 2.1 years which is fast considering the magnitude of the Building. The office buildings have a significant less payback periods as their daily running hours are significant (24 hours also in cases of call centre buildings).

`

CHAPTER – 5
RESULTS AND DISCUSSION

It can be seen from the study that the Payback period of a Heat Recovery Wheel and its auxiliaries for the given building is 2.1 years. This shows the tremendous potential of Heat Recovery Wheels in mechanical ventilation of buildings. Comfort applications provide a better indoor environment that remains relatively constant with respect to the outdoor conditions. Areas that require air conditioning or heating throughout the year are best suited for Heat Recovery Wheels and the payback periods are much less.

The results show significant savings in the operation of the Heat Recovery Wheel. The Real time data shows that the Heat Recovery System is best suited for Tropical wet and dry climates or where throughout the year continuous heating or cooling is required. Also, the more dense the buildings are, the more ventilation rates shall be required and thus the payback period of the Wheel shall reduce. The difference in the initial investment costs is quickly made up by substantial reduction in the operating costs.

In winters, the heat Recovery Wheel can pre heat and pre humidify outdoor air by removing both sensible and latent heat from the exhaust air.

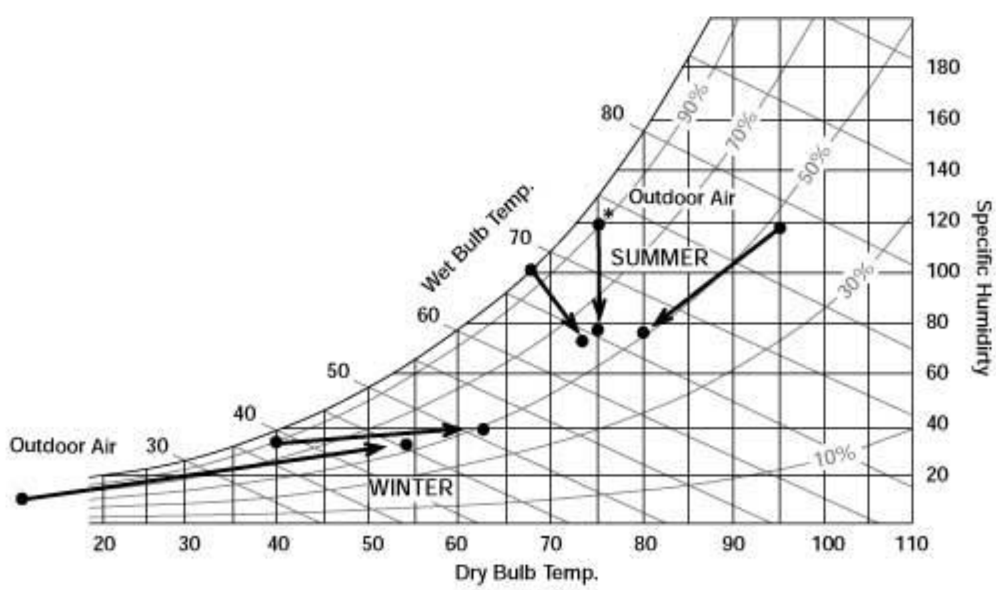

Figure 9- Energy Wheel function for Summers and Winters

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The following points have been observed while doing calculations for the savings in the load due to the use of Heat Recovery Wheel.

a) January has almost zero savings with the use of the Heat Recovery Wheel. This is due to the fact that the Dry Bulb temperature is so low that we do not need to air condition the outdoor air.
b) From February onwards, the outdoor air temperature starts to increase and thus we observe that marginal haphazard savings start to take place.
c) In March, since the ambient temperature has increased the savings also increase and are of the tune of around 20 – 40 btu/ hour.
d) Same trend continues in April and the hourly savings increase with maximum hourly savings reaching to 58 btu/ hour.
e) From May to mid October, the savings further increase with minimum hourly savings ranging around 30 btu/ hour.
f) From mid October to November, the savings start to decrease and are in the range of around 10 – 20 btu/ hour.
g) December is again a zero saving month as the ambient air is cold enough and does not need air conditioning.

The below graphs summarize the effect of revolution speed and face velocity on component effectiveness

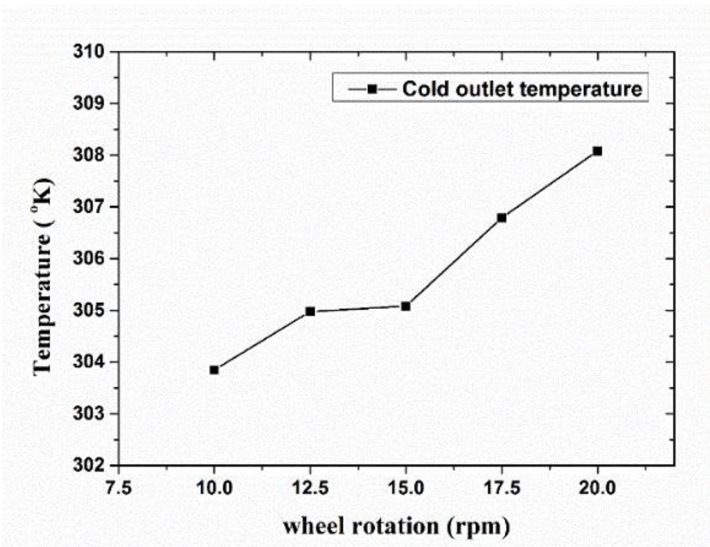

Figure 10. Effect of revolution speed, outlet temperature and constant face velocity in case of cold inlet

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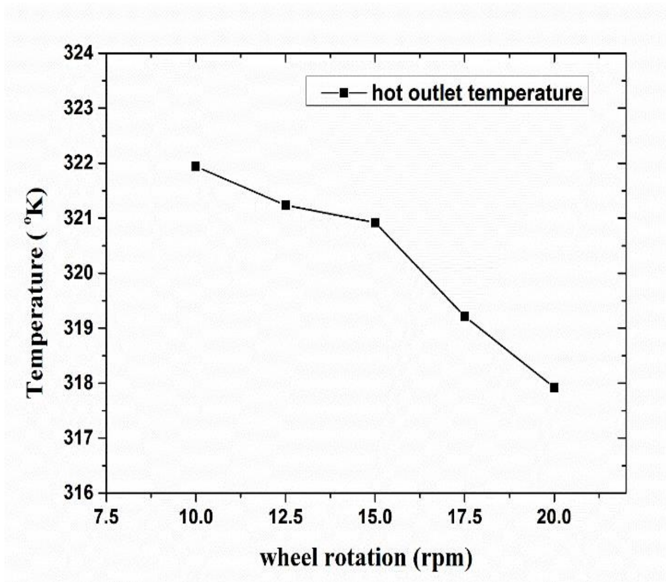

Figure 12. Effect of revolution speed, outlet temperature and constant face velocity in case of hot inlet.

If the heat wheel rotates too slowly, the matrix material average temperature becomes close to the air stream therefore, heat transfer decreases due to limited temperature difference. On the other side, if the wheel rotates too fast, the effect of carryover, i.e., the cross contamination between the two streams due to the amount of air trapped in the wheel channel, becomes relevant.

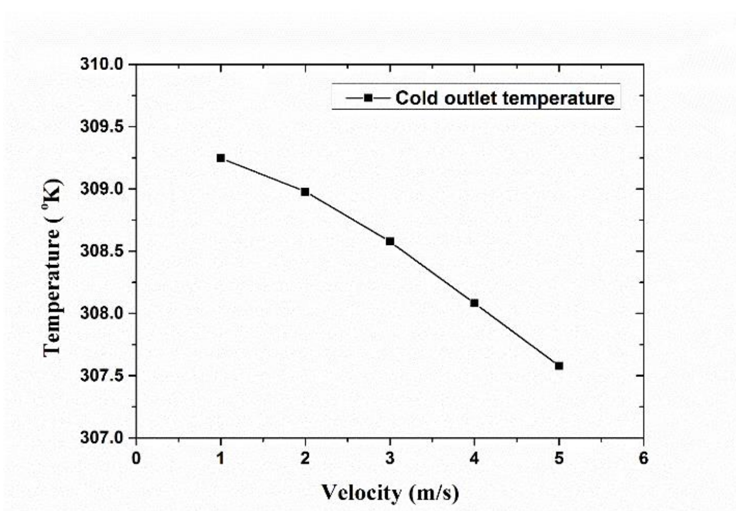

Figure 12. Effect of face velocity, outlet temperature and constant revolution of wheel in case of cold inlet

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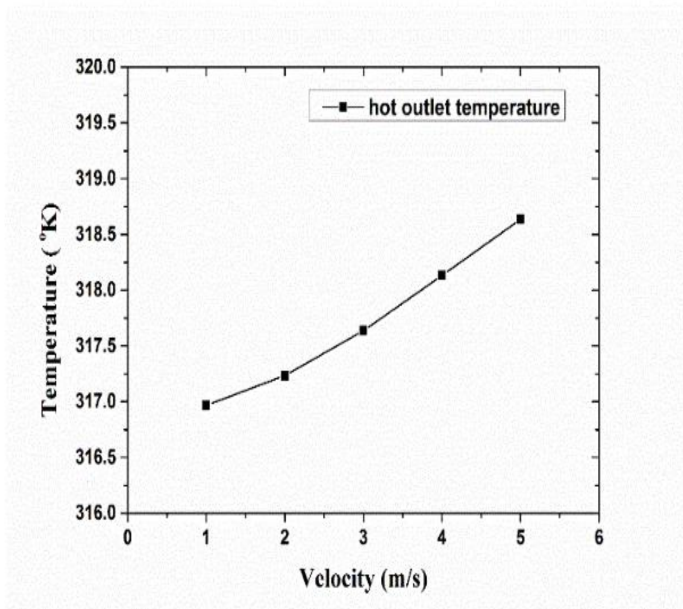

Figure 13. Effect of face velocity, outlet temperature and constant revolution of wheel in case of hot inlet

If the face air velocity of both streams increases the sensible effectiveness decreases because air heat capacity rate is bigger at constant heat transfer area.

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CHAPTER – 6

CONCLUSION

Heat Recovery Wheels are very promising technologies since they provide considerable energy savings in various sectors especially in domestic and industrial buildings. As clean energy generation becomes important day by day, efficient utilization of the energy appears much more significant due to the rising cost of energy production. A real time study is performed on the economical analysis of a Heat Recovery Unit used in the HVAC system used for ventilation of a building. The exhaust air is passed through a heat recovery unit and pre cooled. The suggested system not only preheats the air in winters.

The savings in energy consumption for a full year using weather bin of Kolkata comes to be 192339 Btu-hr/cfm. This comes out to be 36.39 % savings with respect to the conventional system of mechanical ventilation without Heat Recovery Wheel. The energy savings vary with the variation in the input parameters such as outdoor air quality, outdoor air temperature, return air temperature and no of operating hours. The heat transfer rate increases as the air flow rate increases

From this project we can conclude that, global shortage of energy sources can be overcome by using heat wheel as they provide cost-effective and environmentally friendly energy. Further the Heat Recovery Wheels will play an important role in the energy sector as a consequence of the developments in material sciences and growing significance of environmental issues.

Payback analysis is used to evaluate and compare systems. However, the payback analysis has a limitation—it considers only the initial cost of implementing a system and the recurring savings in energy. In addition, it ignores the cost of maintenance and financing, and any interactions a system may have with other building systems. Long term costs of the Heat Recovery Wheels shall have to be studied and analyzed to get a better understanding/ solution to this limitation.

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CHAPTER – 7

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