İSTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE ENGINEERING AND TECHNOLOGY

EFFECT OF DESIGN AND OPERATIONAL STRATEGIES ON THERMAL COMFORT AND PRODUCTIVITY

M.Sc. THESIS

Ergin KÜKRER

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Heat - Fluid Programme

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Thesis Advisor: Prof. Dr. Nurdil ESKİN

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TASARIM VE İŞLETME STRATEJİLERİN ISIL KONFOR VE ÜRETKENLİĞE ETKİSİ

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ABBREVIATIONS

CDD	: Cooling Degree Days
CFD	: Computational Fluid Dynamics
clo	: Clothing Insulation Value
COP	: Coefficient of Performance
EER	: Energy Efficiency Rate
FCU	: Fan-Coil Unit
HDD	: Heating Degree Days
HVAC	: Heating Ventilating and Cooling
IAQ	: Indoor Air Quality
IEA	: International Energy Agency
ISO	: International Standards Organsation
LTD	: Local Thermal Discomfort
MED	: Merkezi Derslik
MRT	: Mean Radiant Temperature
PFCU	: Personalised Fan-Coil Unit
PMV	: Predicted Mean Vote
PPD	: Percentage of People Dissatisfied
RH	: Relative Humidity
SBEM	: Simplified Building Energy Modelling
SHGC	: Solar Heat Gain Coefficient
VAV	: Variable Air Volume
VRF	: Variable Refigirant Flow

SYMBOLS

$\dot{V}$	: Pulmonary ventilation
A_{du}	: DuBois area
A_{eff}	: The effective radiation area of the clothed body
C	: Heat loss or gain by convection
C_p	: Spesific heat of dry air
dm	: Mass transfer
dQ	: Heat transfer
dW	: Heat production
E	: Evaporation on skin heat loss
E_d	: Evaporation on skin heat loss
E_{re}	: Latent respiration heat loss
f_{cl}	: The ratio of the surface area of the clothed body to the surface area
f_{eff}	: The effective radiation area factor
H	: Internal heat production
h_c	: Convective heat transfer coefficient
I_{cl}	: Clothing index
K	: Heat loss or gain by conduction
L	: Dry respiration heat loss
l	: Height
m	: Mass
M	: Metabolic heat production
m	: Permeance coefficient of the skin
η	: Mechanical efficiency
P_a	: Vapour pressure in ambient air
P_s	: Saturated vapour pressure at skin temperature
R	: Heat loss or gain by radiation
R_{cl}	: Total heat transfer resistance from skin to outer surface of the clothed body
R_{es}	: Dry and latent respiration heat loss or gain
S	: Stored or lost energy over time
t_a	: Ambient air temperature
t_{cl}	: Cloth surface temperature
t_{ex}	: Expiration air temperature
t_{mrt}	: Mean radiant surface temperature
t_s	: Skin surface temperature
W	: Mechanical Work
W_a	: Humidity ratio of inspiration air
W_{ex}	: Humidity ratio of expiration air
ε	: The emittance of the outer surface of the clothed body
λ	: Heat of vaporization of water
ν	: Air velocity
σ	: The Stefan-Boltzmann constant

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EFFECT OF DESIGN AND OPERATIONAL STRATEGIES ON THERMAL COMFORT AND PRODUCTIVITY

SUMMARY

As technology is advancing day by day, air-conditioning technology is also being expanded. Since the increasing energy consumption is reaching at an alarming rate, many researches emphasise the significance of energy efficiency. Yet, for a qualified engineer, the main purpose should be obtaining comfortable places for inhabitants as well as enhancing energy efficiency.

According to Environmental Protection Agency, people spend nearly 90% of their time indoors. Furthermore, there are vast of researches indicating the relativity between indoor comfort, productivity and health. In this sense, indoor environmental quality has an important place in human life. Thermal comfort is one the leading issues as far as indoor environmental quality is concerned.

Thermal comfort can be described as feeling comfortable in an environment based on the thermal conditions of the place. The most widely accepted thermal comfort model is “Fanger’s Thermal Comfort Method” and it relies on heat balance equations between the human body and its surroundings. According to Fanger Method, six parameters affect thermal comfort which are, air temperature, relative humidity, mean readiant temperature, air velocity, activity type/level and thermal resistance of clothes.

At this study, significance of thermal comfort parameters will be addressed. The main goal of the thesis is to assess the present thermal comfort scale of a selected building and investigate the ways of optimizing thermal comfort. In order to evaluate the thermal comfort, a case study is followed in a school building, Merkezi Derslik (MED). MED is a multi-functional school building placed in Istanbul Technical University Ayazaga Campus which has different usages of areas. To evaluate the building’s overall energy performance and thermal comfort analysis, simulation tools have been utilized. In the study, dynamic simulation model of MED is generated via DesignBuilder which is capable of modelling both energy and thermal comfort model of a given building. In the simulation tool, MED is defined with respect to its actual state of construction elements, HVAC systems and working schedules. After generation of the model, verification work is carried out. In order to verify the simulation model, one-week measurement is taken place in an amphitheater classroom in MED. After the measurement, measured air temperature and relative humidity have been compared to the simulation model; accordingly verification of the model has been done.

After the verification of the model; building’s existing loads, energy consumptions and thermal comfort status are calculated and investigated. Apart from common zones, MED consists of three different types of heated/cooled zone groups; North-oriented lecture halls, South-oriented seminar rooms and offices. Analysis showed

that especially in northern lecture halls, thermal sensation is rather cold and thermal comfort index is significantly poor. Also it has been noticed that there is thermal discomfort during some periods in the building. One of the main reasons for that is North-oriented lecture halls do not recieve any solar iraddiation. Besides, classrooms are only occupied between 07.30 and 17.00, accordingly mechanical systems of the refferred zones are only utilized in that time period. Consequently, till 17.00 to 07.00, the zones are not heated or cooled for a long time period which causes thermal discomfort especially during morning hours. In South-oriented zones, on the other hand, having floor-height glazings and no cooling serpantine in AHU, has resulted with increased temperatures during the summer months.

Taking into account all these information, a set of design and operational strategies developed and their affect on thermal comfort and thermal comfort parameters are evaluated with the purpose of optimizing the indoor environmental quality and productivity of the building. In the study, optimisation of AHU, optimisation of the shading element, pre-heating/cooling, set-point variation and effect of clothing strategies are evaluated and investigated. While AHU and shading element optimisation contributed office zones to be more comfortable during summer; set-point variation and pre-heating/cooling enhanced the thermal comfort sensation in northern-oriented lecture halls with the increase of annual energy consumption by 5 to 9%. On the other hand enhanced thermal environment has resulted with better productivities. It is presented that better indoor environmental quality corresponds to up to 10% productivity increase in both typing and thinking tasks.

In the study, each strategy is addressed separately, and as a conclusion, a final model that consists of combination of each strategy is generated. Combination of each strategy has resulted with obtaining a PMV value in the desired and reccommended interim which is -0,5 and +0,5 in both overall building and pre-defined zone groups. However, enhancing thermal comfort has lead the annual heating and cooling consumption of the building 8,5% and 14,2% respectively.

TASARIM VE İŞLETME STRATEJİLERİN ISIL KONFOR VE ÜRETKENLİĞE ETKİSİ

ÖZET

Gün geçtikçe artan enerji tüketimi, yeni teknolojilerin gelişimi ve enerji verimliliği ile alakalı pek çok çalışmayı da beraberinde getirmiştir. Günümüzde konutsal alanda tüketilen enerjinin büyük bir kısmını binalardaki iklimlendirme, ısıtma, soğutma ve havalandırma sistemleri oluşturmaktadır. İklimlendirme sistemleri ve sistemlerde enerjiyi etkin kullanabilmek son yıllarda en çok tartışılan konuların başında gelse de, kullanılan iklimlendirme sistemlerinin asıl amacı olan, ortamda bulunan insanlara kabul edilebilir seviyede, konforlu bir ortam sunmak da mühendislik alanındaki önemini konumaktadır. Literatürde insanların yaşamlarının büyük bir kısmını iç ortamlarda geçirdiğini ortaya koyan pek çok çalışma mevcuttur. İnsanoğlu bulunduğu ortamdaki çevresiyle sürekli bir etkileşim halindedir. Bu bakımdan üretkenlik, sağlık, yaşam kalitesi ile iç ortam kalitesi arasında önemli bir bağ vardır. Araştırmalara göre ısıl konforun sağlandığı ortamlarda, kişilerin algı ve iş yapma performansı en üst seviyeye ulaşmaktadır.

Isıl konfor, kişinin bulunduğu ortamda hissettiği ısıl rahatlık olarak tanımlanmaktadır. İnsan vücudu çevresiyle, ısı üretimi, iletimi ve kütle transferi olmak üzere 3 farklı mekanizma vasıtası ile etkileşim halindedir. Bu etkileşimi modelleyebilmek, uzun yıllardır araştırmacıların odak noktası olmuştur. Deri sıcaklığının 34°C'nin altına düştüğü ya da 37°C'nin üstüne çıktığı durumlarda, deri altındaki reseptörler tarafından beyine impuls gönderilerek vücuttaki ısıl dengenin korunması sağlanır. Herhangi bir impulsun oluşmadığı durumlarda ise ısıl dengenin oluştuğu ve dolayısıyla "ısıl tarafsızlık" ile beraber konforun sağlandığı ortaya konulmuştur. Isıl dengeye dayalı bu konfor modeli ilk defa P.O Fanger tarafından incelenmiş olup, daha sonra da kendi ismiyle "Fanger Isıl Konfor Modeli" olarak anılmıştır. Fanger modeli bu tezde de detaylı bir şekilde inceleneceği üzere ısıl denge üzerine kurulmuştur.

Fanger'in oluşturmuş olduğu ısıl konfor denklemini kullanarak, bir hacimdeki kişilerin ısıl konforunu tahmin edebilmek olasıdır. Isıl konforun tayini için fiziksel olgularla beraber kişilerin psikolojik faktörlerinin de değerlendirildiği Tahmini Ortalama Oy (Predicted Mean Vote – PMV) ve Tahmini Konforsuzluk Yüzdesi (Percentage of People Dissatisfied – PPD) İndeksleri geliştirilmiştir. Günümüzde bu geliştirilen PMV ve PPD'ye dayanan bu model, en yaygın kullanılan ısıl konfor değerlendirme metodudur. Araştırmalara göre PMV ve PPD indeksi, dolayısıyla ısıl duyum, çeşitli parametrelere dayanmaktadır. Bunlar: ortalama iç sıcaklık, ortalama iç bağıl nem, ortalama bağıl hava hızı, ortalama yüzey sıcaklığı, aktivite tipi ve seviyesi ve giysilerin ısıl değeridir.

Bu çalışmanın amacı yukarıda bahsi geçen ısıl konfor parametrelerini de göz önünde bulundurarak, bir binanın ısıl konfor durumunu daha önce tanımlanan Fanger'in ısıl denge metodu ile inceleyip, ısıl konforu hesaplamak; ve sonrasında geliştirilen tasarım ve işletme stratejileriyle var olan konforu ve buna ek olarak üretkenliği optimize etmeye çalışmaktır.

Tezde açık ve adım-adım bir metodoloji izlenmiştir. İlk kısımda, çalışmanın amaç, kapsam ve yönteminin belirtildiği giriş kısmı yer almaktadır.

Tezin ikinci kısmında, kaynak taraması yapılmış ve ısıl konfor ile ilgili daha önce yapılmış çalışmalar incelenmiştir. Bu kısımdaki kaynak taraması, ısıl konfor modelleri, ısıl konforun belirlenmesi, üretkenliğe olan etkisinin incelenmesi, parametrik analizi ve binalara uygulanan farklı stratejilerin ısıl konfora etkilerinin belirlenmesi olmak üzere değişik kapsamlarda değerlendirilmiş ve sunulmuştur. Kaynak taramasına ek olarak, günümüzde Amerika Birleşik Devletleri'nde ve Avrupa'da en sık kullanılan ısıl konfor standartları olan ASHRAE 55 ve ISO 7730 standartları da bu kısımda incelenmiştir.

Üçüncü kısımda ısıl konfor daha derinlemesine incelenerek, ısıl konforun geçmişten bugüne nasıl ele alındığı ortaya konulmuştur. Daha önce de bahsi geçtiği üzere, en önemli ısıl konfor modeli olan "Fanger Isıl Konfor Modeli" detaylı bir şekilde tanıtılmıştır. Fanger'in insan bedeninin ısıl dengesine dayanan konfor modeli ısıl dengeyi oluşturan her bir bileşen için tek tek irdelenmiştir. Isıl dengenin sağlanabilmesi için, metabolik iç ısı üretimi, deriden difüzyonla olan ısı geçişi (terleme), solunum yoluyla olan ısı geçişi, ışınım yoluyla meydana gelen ısı geçişi, taşınım yoluyla meydana gelen ısı geçişi, iletim yoluyla meydana gelen ısı geçişinden her biri detaylı bir şekilde Fanger yöntemine bağlı kalarak etüd edilmiştir.

Tezin dördüncü kısmında ısıl konfor göstergeleri ve ısıl konforu etkileyen değişkenler yer alır. Daha önce tanıtılan ısıl konfor parametreleri,ortalama iç sıcaklık, ortalama iç bağıl nem, ortalama bağıl hava hızı, ortalama yüzey sıcaklığı, aktivite tipi ve seviyesi ve giysilerin ısıl değeri ayrı ayrı incelenmiştir. Bu altı faktörden ilk dördü ortamın fiziksel durumu ile alakalıyken, diğer ikisi kişilere bağlıdır. Ayrıca bu bölümde ısıl konfor göstergesi olarak Tahmini Ortalama Oy (PMV), Tahmini Konforsuzluk Yüzdesi (PPD) ve Yerel Isıl Konforsuzluk (LTD) indekslerinden de bahsedilmiştir.

Beşinci bölüm ise, daha önce bahsi geçen ısıl konfor hesaplamalarının değerlendirildiği bölümden oluşmaktadır. Isıl konforu hesaplayabilmek ve örnek bir model üzerinde çalışabilmek için İstanbul Teknik Üniversitesi Ayazağa Yerleşkesi'nde yer alan Merkezi Derslik Binası (MED) seçilmiştir. MED, içerisinde amfi tipi sınıflar, seminer odaları, akademisyen ofisleri, bilgisayar laboratuvarları, kafeteryalar gibi pek çok farklı kullanıma sahip alanı bulunan bir yapıdır. Beşinci bölümde MED'in bu yapısı ile birlikte, tasarım özellikleri, yapı elemanlarının termo-fiziksel özellikleri, iklimlendirme sistemleri tanıtılmıştır. Binanın enerji ve ısıl analizini yapabilmek için dinamik simülasyon programlarından faydalanılmıştır. MED'in modeli, fonksiyonel bir analiz programı olan DesignBuilder'da yapılmıştır. DesignBuilder, EnergyPlus eklentisi ile birlikte tanımlanan, yapının hem enerji hesaplarını, hem de ısıl konfor hesaplarını yapabilen bir simülasyon aracıdır. Binanın simülasyon modeli DesignBuilder'da yapı elemanları, işletme takvimi ve tasarım durumları göz önünde bulundurularak aslına uygun olarak modellenmiştir. Modelin oluşturulmasından sonra, modelin gerçekliğinin ve tutarlılığının doğrulanmasının gerçekleştirilmesi amacıyla binadaki bir amfide bir haftalık süreyle sıcaklık ve nem ölçümü yapılmıştır ve elde edilen ölçüm sonuçları simülasyon verileriyle karşılaştırılarak modelin doğrulanması sağlanmıştır.

Modelin doğrulanmasının ardından, tezin altıncı kısımında binanın mevcut halinin ısıl yük, enerji ve ısıl konfor durumları irdelenmiş ve simülasyon sonuçları baz alınarak ortaya konmuştur. MED, ortak alanların dışında 3 farklı ısıtılıp soğutulan hacimden meydana gelmektedir: kuzey cephede yer alan amfiler, güney cephede yer alan seminer sınıfları ve ofisler. Bu üç zon grubunun PMV ısıl konfor indeksi incelendiğinde özellikle kuzey cephedeki amfi tipi sınıfların oldukça soğuk hissettirdiği ve zaman zaman ısıl konforsuzlukların meydana geldiği görülmüştür. Bunlara temel sebep olarak, kuzey cephedeki amfilerin güneş ışığı almaması ve amfilerin yalnızca 07.00-17.00 arasında ısıtılıp/soğutulması dolayısıyla binada ısıtmayan süreçte ısıl yığılmaların oluşması ve sabah saatlerinde ısıl konforun oldukça düşük olması gösterilebilir. Güney cephedeki zonlarda ise, boydan boya cam olması ve taze hava santralinde soğutucu serpantinin olmaması yaz aylarında sıcaklıkların artmasına sebep olup, konforun azalmasına yol açmıştır.

Tüm bunlar göz önünde bulundurularak farklı stratejiler geliştirilmiş, hem bu stratejilerin konfora olan etkisi gözlenmiş; hem de binada daha önce bahsi geçen "ısıl tarafsızlığa" ulaşmaya ve üretkenliğin artırılmasına çalışılmıştır. Çalışmada, taze hava santralinin optimizasyonu, gölgeleme elemanının konumlandırılmasının tayini, sıcaklık ayar noktasının (set-point) optimizasyonu, ön-ısıtma/soğutma ve kıyafetlerin ortamdaki konfora aylık etkisinin incelenmesi stratejileri hesaplanmış ve değerlendirilmiştir. Taze hava santralinin optimizasyonu ve gölgeleme elemanı stratejisi özellikle ofislerin yaz koşullarındaki şartlarını iyileştirirken; sıcaklık ayarı optimizasyonu ve ön ısıtma/soğutma ise %5 ila 9 bir enerji tüketimi artışına yol açsa da, amfi sınıflardaki ısıl duyumu çok soğuktan, "nötr – biraz soğuk" arasına çekmiş ve standartlara yakın bir hale getirmiştir.

Çalışmanın son bölümünde sonuç olarak tüm stratejilere tek tek değinilmiş ve en son tüm senaryoların birlikte uygulandığı bir model oluşturulmuştur. Bu modelde bina genelinde ve zon gruplarında PMV değeri istenen -0,5 ve +0,5 aralığında kalmış, ancak stratejiler yıllık ısıtma tüketiminde toplam %8,5, soğutma tüketiminde ise %14,2'lik bir artışa yol açmıştır. Binada ısıl konforla birlikte, üretkenlik de yakından araştırlmıştır. Özellikle binanın bir okul binası olduğu gözönünde bulundurulacak olursa, iç ortam konforunun üretkenliğe olan etkisinin ne denli önemli olduğu da ortadadır. Bu açıdan iç ortam kalitesinin içeride bulunan kişilerin düşünme ve yazma, okuma gibi faaliyetlerine etkisi ayrı ayrı denklemler ile hesaplanmış ve en son oluşturulan stratejide ne kadarlık bir iyileştirme yapılabileceği ortaya konulmuştur. Sonuçlara göre bütün uygulanan senaryolar hem kullanıcıların ısıl konforu hem de üretkenliğine olumlu yönde etki etmiştir. Geliştirilen senaryolar ile ortalama yazma ve düşünme faaliyetlerinde yaklaşık %10 artırım mümkün olmaktadır.

1. INTRODUCTION

The world’s dependence on energy has been increasing notably day by day. According to International Energy Agency (IEA) statistics, world-wide energy consumption has increased by 92% between the years of 1971 and 2014 [1]. As a result of this, increasing energy consumption has led fossil energy sources drain away in a rapid way which also has paved the way for searching new technologies and sources that do not depend on fossil fuels.

Taking pre-mentioned energy problems recently on the basis, energy efficiency is one of the most discussed topics in the field of air-conditioning design. However, it should be noted that a qualified engineer has to maintain thermal comfort for inhabitants as well as enhancing energy efficiency.

The interaction between human body and its environment directly affects the human life. Vast of researches have already presented that there is a link between indoor environmental quality, productivity and health [2]–[4]. Inhabitants’ comfort is both important in a psychological, health and production aspect. In this context, when designing the places we live, work and spend time, creating a comfortable habitat still remains one of the most challenging issues. In a place where optimum thermal comfort is obtained, occupants’ perception and performance usually reaches the highest level [5].

Human body is a subject to both heat production and transfer with its environment continuously. Consequently, researchers have carried out studies to solve this interaction and obtain more comfortable environments for humankind. When the skin temperature falls below 34˚C, cold censors placed in the skin start to send impulses to the brain. Similarly, heat censors step in when the skin temperature goes higher than 37 ˚C. If the signals on the both sides are equal, it can be said that thermal neutrality is obtained [6]. The most well-known thermal comfort model depends on the thermal neutrality and is developed by Fanger P.O, which is also named after him; “Fanger Thermal Comfort Model”. Fanger’s model relies on the heat balance and will be discussed in the thesis comprehensively. By using the Fanger's thermal comfort equation, it is possible to obtain predicted thermal comfort of the occupants in a place. In order to illustrate the thermal comfort, Fanger has developed Predicted Mean Vote (PMV) index, which basically predicts thermal comfort as a function of a set of parameters. Today, the PMV index is a widely-used indicator of thermal comfort. In addition to PMV index, Fanger has evaluated another index; Percentage People of Dissatisfied (PPD) which predicts the mean value of thermally dissatisfied people in a place.

In his research, Fanger has noted some significant parameters that affect thermal comfort which are;

- air temperature
- relative humidity
- air velocity
- mean radiant temperature
- activity type and level
- thermal insulation.

In this thesis, significance of the thermal comfort parameters will be addressed. The aim of this research is to evaluate the present thermal comfort scale of a building and investigate the ways of optimizing thermal comfort. The selected building for the thesis is a multi-functional school which has VRF systems added on subsequently to the building.

In the study, a step-by-step clear methodology is evaluated. Firstly, thermal comfort and its models are introduced. Since Fanger's method is going to be applied when evaluating the thermal comfort, Fanger method is explained comprehensively. As a second step, above-mentioned significant thermal comfort parameters are investigated in a broader aspect.

The study is followed by the case study. After introduction of the case study building and its actual conditions; simulation is carried out in order to evaluate the energy and comfort based results of the building. Subsequently, diverse operational and design strategies and their effect on the thermal comfort and productivity is evaluated.

The thesis puts emphasis on thermal comfort in the commercial buildings especially in school buildings since the relationship between productivity and thermal comfort .

2. LITERATURE REVIEW

The literature review is based on researches mainly concerning thermal comfort models, evaluation of thermal comfort, noteworthy parameters and thermal performance studies in buildings. Furthermore, thermal comfort standards will be identified in the present chapter.

2.1 Background

In the literature, there are a vast of researches regarding the indoor thermal comfort. According to Szokolay, even in antique ages, there were thoughts related to climatic suitability of the places where people lived [7]. Besides, Vitrivius who was a well known Roman author, architecture and civil engineer in the first century BC wrote in his book that "*all bodies are composed of the four elements, that is heat, moisture, the earthy and air, yet, there are mixtures according to natural temperament*." Moreover, according to Vitrivius, if heat becomes predominant in the body, it would destroy and dissolve all the other with its violence [5], [8]. Though, it is evident that, thermal balance of the body was considered in the past. Nevertheless, most of the studies did not start practically until the Industrial Revolution [7], [9]. As mentioned in the introduction of the thesis, development of the technology in the building climate techniques has resulted with better control of the HVAC systems. Consequently, a better environment for people has been a popular topic amongst the scientists and designers as well.

In 1923, Houghten and Yagloglou have developed the first studies on the comfort by simulating different conditions in the laboratories [10]. In the study, Houghten and Yagloglou made an effort to define control zone and proposed lines of comfort. As a result of their study, "effective temperature" is evaluated for human, which can be defined as correspondence between the temperature of the actual environment and the temperature of a notional environment [11]. The study is followed by Vernon in 1930, which is related to the measurement of the radiant heat in relation to human comfort [12]. In 1932, Vernon and Warner, have also conducted empirical studies among factory workers [13].

In 1970, after both mathematical models and experimental studies, Fanger has found out the numerical and more practical way to evaluate thermal comfort and defined it with predicted mean vote (PMV) and percentage of person dissatisfied (PPD) terms [14].

In the recent years, researches are mostly collected under two methods; laboratory studies and field studies. Researches have examined the parameters that concern thermal comfort and their impact on the thermal sensation via laboratory and field studies.

Toftum and Fanger have investigated thermal dissatisfaction based on the air temperature and relative humidity. In the study, it is found out that, optimum RH is below 36% for 26°C and below 57% for 23°C. Furthermore, the study refers the adverse effects of the high and low RH values on the both occupant health and building construction elements [15]. In a parallel research, Fang, Clausen and Fanger have studied the impact of temperature and humidity on perception of indoor air quality (IAQ) with different levels of air temperature and RH. In the research, which is carried out in climate chambers, temperature ranges vary between 18-28°C and RH between 30 - 70%. According to results, it is shown that temperature and humidity have a significant impact on perceived IAQ during whole-body exposure [16].

Fountain et al. have conducted climate chamber studies to investigate thermal comfort at higher humidities. Experiments are performed under conditions of 20-26 °C effective temperature and 60-90% relative humidity(RH) and according to results it was seen that PPD values never decrease under 25% in the activity levels which are defined as higher than 1,6 met [17].

Kaynaklı et al. studies the effect of temperature, RH, air velocity, metabolic activity and clothing resistance. Steady-state energy balance is evaluated and calculations are made via FORTRAN programming language. Each parameter is discussed comprehensively with relative graphics. In the study, not only the effect of thermal comfort variables on the human body is addressed, but also their effect on each other is examined [18].

Atmaca and Yiğit have analysed thermal comfort standards and energy balance models. Thermal comfort parameters are presented and the effects of personal and environmental indicators are investigated with the experiments. In the study, thermal sensation is evaluated for different values of air temperature, air velocity, mean radiant temperature, RH and metabolic activity level. Researchers suggest that 20 – 22 °C air temperatures and 0,4 – 0,6 m/s relative air velocity levels might cause discomfort for occupants, although the values are given as acceptable in the present standards [19].

Havenith, Holmer and Parsons argue the significance of clothing parameters and metabolic rate in terms of PMV indices. For clothing insulation, it is presented that effects of body motion and air movement have an important role and must be taken into account when assessing the thermal comfort. The study also emphasises the importance of measurement of metabolic rate in order to obtain a precise comfort assessment [20].

As well as, significance of thermal comfort parameters, conducted experimental studies in the literature also address the psychological aspect of the thermal comfort. In the literature, numerous researches can be found that remark the significance of thermal comfort and its impact on the productivity.

In an experimental study, Tanabe S. et al have taken a Japanese office building into consideration by the terms of indoor temperature and productivity [4]. In the experiments, exposure of various temperatures (25.5 ˚C, 28.0 ˚C and 33 ˚C) is studied both as short term (30 minutes) and long term (6 hours). Results revealed that the impact of thermal environment on task performance in short term was not consonant; however, in long term, increasing indoor air temperature had effect on workers' performance in a negative way.

Parallel studies showed that in the range of 25-32 ˚C temperatures, per ˚C increase leads 2% decrease in performance, whereas between 21-25 ˚C there is no substantive change in workers performance [3]. Besides, another study by Tham et al. (2003) resulted with a similar finding. Lowering the temperature 2K from 24.5 ˚C has resulted with an increase of 5% performance in a call centre. Kobayashi et al. (2005) also conducted an experimental study in a call centre [21]. One year's observation has showed the correlation between indoor air temperature and the average call response. Likewise to Seppänen’s and Tham’s studies, 1 ˚C increase in the air temperature has resulted with 2.1% decrease in workers’ response rate. Clement and Baizhan also suggest that productivity can be improved by 4 to 10% by improving the environmental conditions. [22] Additonally in covering IEQ in an economic aspect, Skaret Fisk and Rosenfield carried out diverse experiments indicating that the yearly potential gain of productivity due to the reduction of respitory infection would equal to 6-14$ billion [23],[24]. Whereas, Skaret points out that increased productivity due to better IEQ is greater that operation and maintenance costs at least 10 to 100 times [25].

On the other hand, Kosonen and Tan suggest a theoretical way to assess the productivity loss in air-conditioned office buildings using the PMV approach [26]. Productivity loss is a function of room temperature at different tasks. Mainly tasks are investigated into two titles; thinking tasks and typing tasks. The results reveal that task-related performance is important associated with the human perception on indoor environmental quality. Temperatures higher that 27°C lead to nealy 30% reduce in thinking tasks and typing tasks. In a similar research, Wyon investigated the impact of thermal comfort zone on human performance by the terms of reading, thinking and performing arithmetic [27]. According to Wyon, thermal environment can affect efficiency and productivity of mentioned activities by 5 to 15%.

Lastly, in the literature review, the examination of different type of strategies and their relation with the comfort issues is assessed as it follows.

Sekhar and Ching examine the IAQ and thermal comfort of an under-floor air-conditioning system in an office building placed in hot and humid climate. Thermal comfort parameters are measured from a set of grid points and Fanger’s PMV index is computed to find out occupants’ thermal sensation. The findings of the study revealed that under-floor air conditioning system offers a reasonable IAQ for inhabitants, however, local thermal discomfort, especially cold feet, is an issue in referred conditioning system [28].

Pan et al. evaluate the thermal comfort and energy saving of a personalized fan-coil unit air conditioning system (PFCU). The study is carried out in a climate chamber and a thermal manikin has been used in the study. The result of PFCU system is compared to central air-conditioning system and PMV values of PFCU are always founded higher than central air-conditioning system. Pan et al. also indicate that the time span required for reaching the desired air conditions is shorter in PFCU systems [29].

Cheong et al. assess the present thermal comfort conditions of an air-conditioned lecture theatre in a tropical climate based on measurement, CFD and subjective assessment. It is found out that measured air temperatures, velocities and RH are within the limits of recommended standards, yet, PMV and PPD indices have shown that occupants might be slightly dissatisfied. Study asserts that VAV air conditioning system is unable to cope with the peak occupancy load. Consequently, an additional demand controller ventilation system is proposed to cover peak loads and reduce the carbon dioxide rate in the lecture theatre. Also, new set-points for optimum thermal comfort is stated in the research [30].

Atılgan and Ekici have compared floor heating and radiator heating systems in an office building in terms of thermal comfort. Thermal comfort analysis has been evaluated in an experiment room located in a university and according to experimental results, floor heating is indicated to be better in creating uniform thermal environment compare to radiator system [31].

Mazzei, Minichiello and Palma study HVAC dehumidification systems in summer for thermal comfort. The study represents how mechanical dehumidification field offers a proper control of ambient temperature and humidity. In the study, various possible air handling unit configurations and their effects are evaluated. By analysing HVAC systems for a supermarket and a theatre, it was found out that hybrid HVAC systems with dehumidification offers remarkable energy savings and better IAQ for inhabitants [32].

Fong et al. have investigated the evaluation of thermal comfort conditions in a classroom with three different ventilation methods. The study is conducted experimentally within the inclusion of 48 participants under the same conditions but various ventilation methods. The thermal comfort analysis is carried out based on supply airflow rate, room temperature and RH. In the research, not only thermal comfort level, but also energy saving of the each system is evaluated [33].

Chirarattananon, Memon and Vangtook have studied application of radiant cooling and its impact on thermal comfort via a case study. The thermal comfort assessment is conducted in a university placed in Pakistan. Naturally ventilated classrooms and air-conditioned offices are simulated using a simulation tool (TRNSYS) for two different cases which are conventional air-conditioning and radiant cooling. The results have revealed that obtaining thermal comfort in the building is achievable for most of the time of the year. Besides, nearly 80% energy saving is possible in the case that thermal comfort is obtained through radiant cooling instead of conventional air conditioning [34].

2.2 Thermal Comfort Standards

The most well-known and practical thermal comfort standards are established by ASHRAE (American Society of Heating, Refrigerating and Air-conditioning Engineers) and ISO (International Standards Organisation); which are

- ASHRAE Standard 55
- ISO 7730

According to Olesen, both standards are well established and in agreement with each other.[35] Hereafter, definition of each standard will be evaluated.

2.2.1 ASHRAE standard 55

ASHRAE Standard 55 is established for thermal environmental conditions for human occupancy. Its aim is given in the standard as “to specify the combinations of indoor thermal environmental factors and personal factors that will produce thermal environmental conditions acceptable to a majority of the occupants within the space.”[36]. It is intended to use for analyzing thermal environment for building design, commissioning and existing buildings specifications.

The standard specifies the human thermal sensation based on four environmental factors and two personal factors as it has been described in the Fanger’s method. It uses steady state conditions as base. ASHRAE Standard 55 predicts thermal comfort with the pre-defined charts as it is given in Figure 2.1.

In Figure 2.1, acceptable range is given in terms of different clothing indices. According to ASHRAE 55 standards, 1.0 clo for heating season, 0.5 clo for cooling season is recommended and represented in the figure respectively.

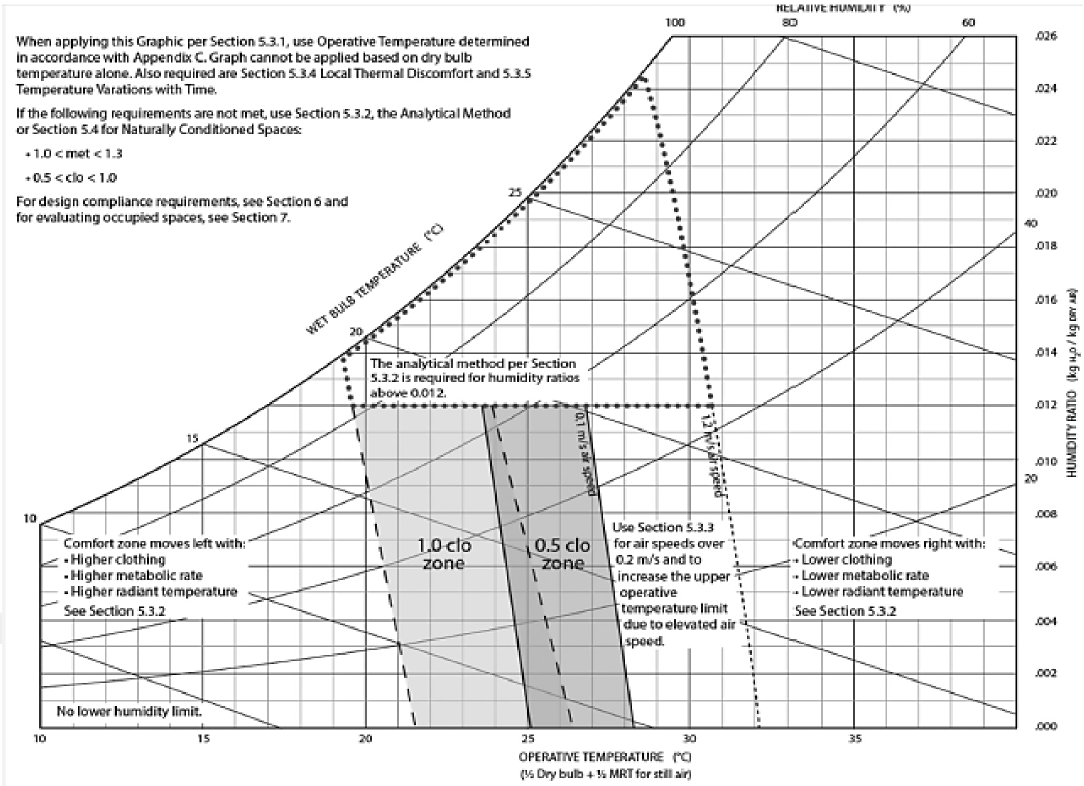


Figure 2.1 : Acceptable range of operative temperature and humidity for spaces[36].

Table 2.1 : Acceptable PPD and PMV ranges given in ASHRAE 55 [36].

PPD%	PMV Range
< 10	-0,5 < PMV < +0,5

Table 2.1 corresponds to the recommended PPD and PMV ranges for typical applications. In a convenient design, PMV ranges should be kept between the values of -0,5 and +0,5, accordingly predicted percentage of dissatisfied could be retained under 10% which means the majority of occupants are satisfied with the present conditions.

2.2.2 ISO 7730 Standard

The European Standard, ISO 7730 has been established to evaluate thermal comfort of indoor environment. ISO 7730 is an international standard that presents an approach to predict the general thermal comfort degree and degree of thermal dissatisfaction degrees of people by using PMV and PPD indices [37]. ISO 7730 Standard is quite similar and compatible with ASHRAE 55 Standards.

In ISO 7730 standards, thermal categories and recommended PMV and PPV values are given in Table 2.2.

Table 2.2 : Acceptable PPD and PMV ranges given in ISO 7730 [37].

Category	PPD (%)	PMV
A	< 6	-0,2 < PMV < +0,2
B	< 10	-0,5 < PMV < +0,5
C	< 15	-0,7 < PMV < +0,7

According to Table 2.2, alike ASHRAE standard 55, recommended PMV and PPD in a space is divided into categories of A, B and C. Based on the use of the building and existing conditions, building category is defined.

ISO 7730 also provides methods for the evaluation of local thermal discomfort such as draughts, asymmetric radiation and temperature gradients. As well as assessment of local thermal discomfort components, the standard includes recommendations in order to avoid discomfort for the occupants.

Along with ISO 7730, there are other subsidiary standards which are ISO 8896, ISO 9920 and ISO 10551 [38]. ISO 8996 corresponds to the estimation of metabolic heat production, whereas in ISO 9920, estimation of clothing properties are defined comprehensively. ISO 10551, subjective assessment methods, presents the principles and the methodology behind the subjective evaluation of the thermal comfort.

3. THERMAL COMFORT

Starting from the ancient times to present day, humankind has been struggling continuously to obtain thermally comfortable environment. In the last decades, advancing technology has enabled the invention of new climatization techniques that offer better indoor environmental quality for inhabitants. Yet, it should be mentioned that, as well as today, people have tried to reach thermally comfortable indoors by applying passive techniques such as natural ventilation, wind-catchers or appropriate building materials even in ancient times where the technology was far less advanced [39], [40]. Therefore, it is obvious that indoor quality of where people live has been one of the most challenging issues in human life, correspondingly, thermal comfort takes an important place in this context. In this chapter, description of thermal comfort and thermal comfort models will be investigated.

3.1 Description of Thermal Comfort

As far as thermal comfort is concerned, there are different definitions in the literature. Since comfort is a term which can be varied person to person, it is true that the term is subjective and could be assessed by subjective evaluation. In most of the studies, thermal comfort definition is often referred to ASHRAE 55 description which defines thermal comfort as "*the condition of mind that expresses satisfaction with the thermal environment and is assessed by subjective evaluation*" [36]. Similarly, another standard ISO 7730 explains thermal comfort as "*that condition of mind which expresses satisfaction with the thermal environment*" [37]. From another point of view, temperature sensations mainly depend on the activity of thermoreceptors placed in the skin, therefore, the condition of thermal comfort can be identified as "a state in which there are no driving impulses to correct the environment by behaviour" according to Hensen [41]. Parsons states that thermal comfort is a state that people strive for when they feel discomfort and refers the term as a 'psychological phenomenon' [9]. Fanger, who is widely regarded as one of the foremost scientists in the field of thermal comfort, extends the term with "thermal neutrality” which can be thought as the condition that the subject would prefer neighter warmer nor cooler surroundings [14]. Thus, thermal neutrality is a necessary condition for thermal comfort according to Fanger.

The reason for generating thermally comfortable environment is to meet the thermal demands of human and make man feel comfortable in where they spend time. Therefore, in order to acquire the thermal requirements of inhabitants, buildings are employed with diverse mechanical systems. Working schedule and typology of systems may vary from building to building according to occupants’ need, building typology or weather conditions. For instance, in Turkey, most of the residential buildings do not have active cooling systems, yet, in the Southern parts, it is common to have single air-conditioner units in apartments separately. In that way, each single thermal zone is treated different from each other. Consequently, it has led to building of thermal comfort standards and models such as ISO7730 and ASHRAE 55 which are described in the previous chapter. In the next section, thermal comfort models will be discussed.

3.2 Thermal Comfort Models

On designer perspective, determination of thermal comfort means obtaining indoor environmental quality for inhabitants by applying thermal comfort models which have been found through scientific methods. Thermal comfort models referring to human body are being discussed and developed since 1960s [42]. Most widely accepted thermal comfort standards ASHRAE 55 and ISO 7730 are both based on “Fanger Method”, which relies on heat balance equations between human body and its surroundings.

3.2.1 Fanger method

The thermal comfort model developed by P.O. Fanger between the years of 1966 and 1970 is regarded as one of the primary studies to determine thermal comfort. In 1967, to derive the correlation between activity level and sweat rate, Fanger used data from a study by McNall, Jaax, Rohles, Nevins and Springer and conducted a study in which college-age participants were exposed to different thermal conditions with standard clothing [43]. In the study, participants were made to vote on their thermal sensation and the linear relationship was formed from the ones which stated that they felt thermally neutral [44]. In addition to this experiment, Fanger has held another study which aims to obtain a linear relationship between activity level and mean skin temperature. In this study, college-age participants were made to conduct different type of activities that have different levels. On the basis of these studies, Fanger has developed a thermal comfort equation which predicts conditions where occupants will feel “thermally neutral”. According to Fanger, necessary condition for thermal comfort depends on thermal neutrality which can be described as the state of condition in which a person feels neither hot, nor cold.

Fanger noted that the most significant parameters for thermal comfort are:

- Ambient temperature
- Relative humidity
- Relative air velocity in the air
- Mean radiant temperature
- Metabolic rate
- Thermal resistance of the clothing

At the present time, Fanger’s parameters are thought as mainstream variables for determining thermal comfort and will be discussed in following chapters in the thesis.

The Fanger Model is based on the basic characteristics of thermodynamics: the first law. First law of the thermodynamics, which is also known as the law of conservation of energy, states that the total energy of isolated system is constant and energy can neither be created nor be destroyed. Applying the same logic to the human body, Fanger has developed the equations for thermal comfort. According to Fanger, the first condition necessary for thermal comfort of a person under long exposure is the existence of a heat balance. Conservation of energy in the human body can be investigated under three titles: heat production ($d\dot{W}$), mass transfer ($d\dot{m}$) and heat transfer (dQ) as it is shown in Figure 3.1 and equation 3.1.

$$\frac{\partial \mathrm{E}}{\partial t}=\partial\dot{w} \pm \partial\dot{m} \pm \partial\dot{Q} \tag{3.1}$$

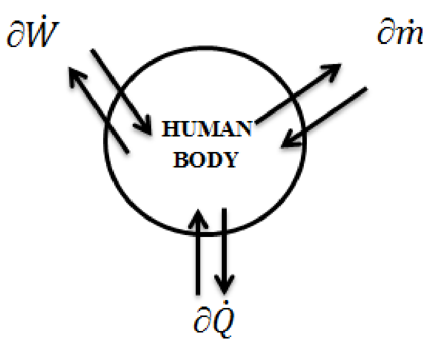


Figure 3.1 : Energy equilibrium of the human body.

In the following section, represented energy equilibrium of human body is going to be investigated further.

3.2.1.1 Thermal equilibrium of human body

As stated in the previous section, Fanger's model depends on the first law of thermodynamics and accordingly human body is a subject to heat transfer with its environment. Consequently, from heat balance calculations, stored or lost energy over time is investigated. According to Fanger, as this value approaches to zero, human internal temperature remains the same. Therefore, each unit of heat balance equation should be considered when thermal comfort of human body is being modelled.

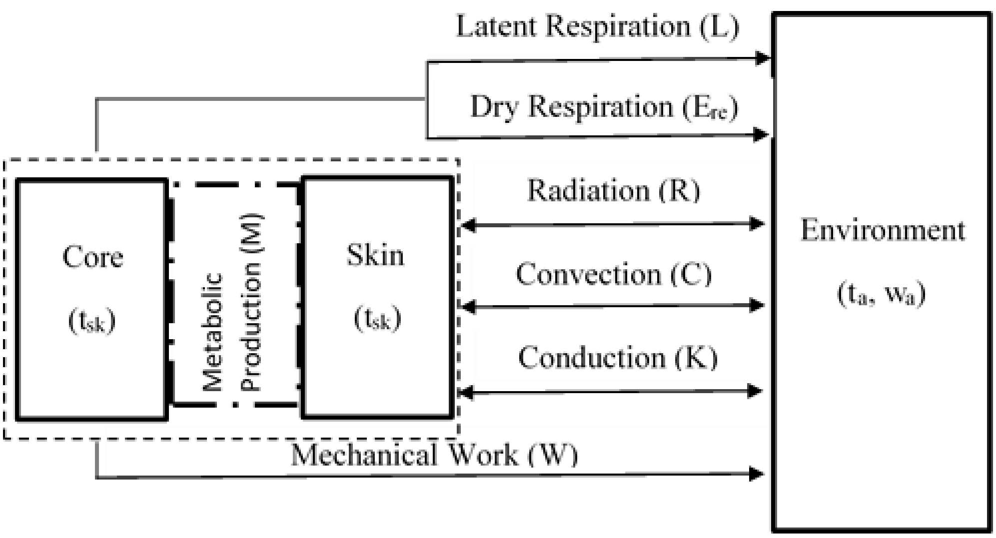


Figure 3.2 : Schematic representation of energy transfer of human body with the environment [5].

A schematic flow chart of the energy transfer of human body with its environment is represented in the following Figure 3.2. Thus, combining all mechanisms represented in Figure 3.2, Fanger has developed energy transfer in human body as given in the following equation 3.2.

$$S = M - W - E_d \pm R_{es} \pm C \pm K \pm R \tag{3.2}$$

where, S = Stored or lost energy iwn human body (W/m^2)

M = Metabolic rate (W/m^2)

E_d = Heat loss by skin diffusion (W/m^2)

R_{es} = Heat loss or gain by respiration (W/m^2)

C = Heat loss or gain by heat conduction (W/m^2)

K = Heat loss or gain by convection (W/m^2)

R = Heat loss or gain by radiation (W/m^2)

As in shown Figure 3.2 and Equation 3.2, energy transfer from human body is related to seven mechanisms: internal heat production, latent respiration, dry respiration, radiation, convection, conduction and mechanical work. In the following sections, each term is going to be assessed separately.

Internal heat production

Human body invariably operates to remain the internal body temperature in a constant value by the help of thermoregulatory system. In a healthy person, internal body temperature varies 36,5 °C to 37 °C [45]. In order to maintain this value, internal heat production takes place when there is lack of heat. The energy produced in a human body is released by the oxidation process and defined as metabolic rate (M) by Fanger. Although metabolic rate is mostly converted to the internal body heat (H), in some cases it can be also converted to external mechanical power (W) as given in the following equation 3.3.

$$M = H + W \tag{3.3}$$

External mechanical power relies on physical activities that require high energy such as climbing or working out. Concerning definition of external mechanical power, external mechanical efficiency term (ɳ) is introduced in equation 3.4 as follows;

$$\eta = \frac{W}{M} \tag{3.4}$$

thus, combining equation 3.3 and 3.4, internal heat production in the human body: equation 3.5 is derived;

$$H = M\,.\,(1 - \eta) \tag{3.5}$$

Metabolic rate and external mechanical efficiency values vary for different typical activities. Fanger has compiled a table for metabolic rate and external mechanical efficiency values however, more recent tables for metabolic rate can be found on ASHRAE 55 and EN ISO 8996 standards. ASHRAE and ISO standards neglect the effect of external mechanical efficiency; therefore, this term is not given in their tables. The following Table 3.1 includes values and comparison of recent thermal standards with Fanger's table.

Table 3.1 : Metabolic rate with reference to ASHRAE 55, Fanger and EN ISO 8996.

Activity	Metabolic Rate (W/m^2)		
	ASHRAE 55:2010	Fanger (1970)	EN ISO 8996:2004
Resting			
Sleeping	40	35	40
Reclining	45	40	45
Seated, quiet	60	50	55
Seated, relaxed	70	60	70
Walking (on level surface)			
up to 2,5 km/h	-	-	70 ÷ 130
3,2 km/h	115	100	130 ÷ 200
4,3 km/h	150	120 ÷ 130	130 ÷ 200
6,8 km/h	220	190 ÷ 290	200 ÷ 260
8 km/h	-	290	>260
Office activities			
Reading, seated	55	50 ÷ 60	70 ÷ 130
Writing	60	50 ÷ 60	70 ÷ 130
Typing	65	45 ÷ 60	70 ÷ 130
Filing, seated	70	50 ÷ 60	70 ÷ 130
Filing, standing	80	50 ÷ 60	70 ÷ 130
Walking about	100	-	-
Lifting/packing	120	-	-
Misc. occupational activities			
Cooking	95 ÷ 115	80 ÷ 100	130 ÷ 200
House cleaning	115 ÷ 200	100 ÷ 170	130 ÷ 200
Machine work	105 ÷ 235	100 ÷ 200	130 ÷ 200

Table 3.2 (continued): Metabolic rate with reference to ASHRAE 55, Fanger and EN ISO 8896.

Activity	Metabolic Rate (W/m^2)		
	ASHRAE 55:2010	Fanger (1970)	EN ISO 8996:2004
Misc. leisure activities			
Dancing	140 ÷ 255	120 ÷ 220	200 ÷ 260
Gymnastics	175 ÷ 235	150 ÷ 220	200 ÷ 260
Basketball	290 ÷ 440	380	200 ÷ 260

Table 3.1 corresponds to comparison of different metabolic rates. Complete versions of metabolic rate values for ASHRAE 55(2010), Fanger(1970) and EN ISO 8996 which are summarized in Table 3.1 can be found on Appendix A.

It is stated in equation 3.5 that mechanical efficiency term is also should be taken into account when considering metabolic heat production. Fanger has depicted different external mechanical efficiency rates in his study, yet in ASHRAE 55 and EN ISO 8896, this term is often neglected. EN ISO 8896 states that the mechanical efficiency of muscular work is so low that in most types of industrial work it is assumed to be equal to nil. Correspondingly, the metabolic rate is assumed to be equal to the rate of heat production. In following Table 3.2, external mechanical efficiency rates are given from Fanger's study.

Table 3.2 : Mechanical efficiencies at different typical activities[14].

Activity	Mechical Efficiency (η)
Resting	
Sleeping	0
Reclining	0
Seated, quiet	0
Seated, relaxed	0
Walking on the level surface	0
Walking up a grade	
%5 Grade	
1,6 km/h	0,07
3,2 km/h	0,10
4,8 km/h	0,11
6,4 km/h	0,10

Table 3.2 (continued): Mechanical efficiencies at different typical activities [14].

Activity	Mechical Efficiency (η)
%15 Grade	
1,6 km/h	0,15
3,2 km/h	0,19
4,8 km/h	0,19
%25 Grade	
1,6 km/h	0,2
3,2 km/h	0,21
Office activities (i.e Reading, writing, typing)	0
Misc. occupations	
Bakery	0 – 0,1
Brewery	0 – 0,2
Carpentry	0 – 0,2
Foundry work	0 – 0,2
Garage work	0 – 0,1
Laboratory work	0
Locksmith	0 – 0,1
Machine work	0 – 0,1
Manufacture of tins	0 – 0,1
Shoemaker	0 – 0,1
Shop assistant	0 – 0,1
Teacher	0
Watch repairer	0
Vehicle driving	0 – 0,1
Heavy work	
Pushing wheelbarrow	0,2
Handling 50 kg bags	0,2
Pick and shovel work	0,1 – 0,2
Digging trenches	0,2
Domestic work	
House cleaning	0 – 0,1
Cooking	0
Washing dishes	0
Ironing	0,1

As represented in Table 3.2, for most of the daily activities estimated mechanical activities equal to zero in the Fanger’s study, which is parallel to ASHRAE 55 and EN ISO 8894. Yet, in some special types of activities such as walking upstairs or walk uphill η can take on values up to 0,20-0,25.

Heat loss and gain by mass transfer

Mass transfer is one of the three mechanisms as far as energy equilibrium of the human body is concerned as given in the equation 3.1. In human body, mass transfer occurs in two ways: skin diffusion and respiration. Hereafter, these two terms will be evaluated.

Heat loss by skin diffusion

Heat loss by skin diffusion takes place by the water vapour diffusion through the skin, in other words by sweating. Evaporation losses in sweating are subject to consider when evaluating the mass transfer in a human body. In the main energy balance equation 3.2 this term is shown with E_d indices and expansion of it is given in the equation 3.6 as it follows.

$$E_d = \lambda \,.\, m \,.\, A_{Du} .\, (P_s - P_a) \tag{3.6}$$

where, E = Evaporation on skin heat loss (W)

λ = Heat of vaporization of water (Wh/kg)

m = Permeance coefficient of the skin (kg/ hr m^2 mmHg)

A_{Du} = DuBois area (m^2)

Ps = Saturated vapour pressure at skin temperature (mmHg)

Pa = Vapour pressure in ambient air (mmHg)

This type of heat loss depends on the amount of moisture on the skin, as well as the difference between water vapour pressure at the skin and in ambient air. Between 27°C and 37°C temperatures, saturated vapour pressure at skin temperature can be linearized with following approximation in equation 3.7 as stated by Fanger. [14]

$$p_s = 1{,}92 \,.\, t_s - 25{,}3 \tag{3.7}$$

In equation 3.7 t_s refers to the skin temperature.

Additionally, in order to determine the amount of vapour reaching the skin, permeance coefficient of the skin must be found. Within the studies of Fanger, permeance coefficient is found out to be 0.00061 kg/h m2 mmHg for human skin.

Skin diffusion is calculated by taking surface area of the nude body. In this regard, the most useful measure is proposed by DuBois in 1916 and described by as given in following equation 3.8 in ASHRAE handbook [46].

$$A_{Du} = 0{,}202 \; . \; m^{0{,}425} . \; l^{0{,}725} \tag{3.8}$$

where, m = mass (kg)

l = height (m)

Heat loss or gain by respiration

With heat loss by skin diffusion, other mechanism that is subject to mass transfer of human body with its environment is respiration mechanism. Description of heat loss through respiration can be divided into two: latent respiration heat losses (E_{re}) and dry respiration heat losses (L) as shown in the following equation 3.9.

$$R_{es} = E_{re} + L \tag{3.9}$$

In the following part, latent and dry respiration will be evaluated separately.

Latent respiration heat loss

Latent respiration of heat loss occurs in respiratory system of human. Heat and water vapour are transferred to aspirated air through respiratory system by convection and evaporation mainly. According to Fanger, the latent respiration heat loss relies on the humidity ratio of expiration air and inspiration air and pulmonary ventilation as it is given in equation 3.10.

$$E_{re} = \dot{V} \, . \, (W_{ex} - W_a) \, . \, \lambda \tag{3.10}$$

where, E_{re} = Latent respiration heat loss (W)

$\dot{V}$ = Pulmonary ventilation (kg/hr)

W_{ex} = Humidity ratio of expiration air (kg water/kg dryair)

Wa = Humidity ratio of inspiration air (kg water/kg dryair)

Asmussen and Nielsen [47] have presented that pulmonary ventilation of human depends on metabolic rate and with the review of Liddel [48], following equation 3.11 is found as an useful approach for pulmonary ventilation rate:

$$\dot{V} = 0{,}006 \, M \tag{3.11}$$

where M stands for metabolic rate. As well as former researches, equation 3.8 is taken as literal method in newer publications such as Miller's in 2010 [49].

Dry respiration heat loss

Similar to latent respiration heat loss, dry respiration heat loss occurs in human respiratory system. However, conversely, only temperature difference causes sensible heat transfer between expired and inspired air. The equation for dry respiration heat loss is given in equation 3.12 as follows;

$$L = \dot{V} \,.\, C_p \,.\, (t_{ex} - t_a) \tag{3.12}$$

where, L = Dry respiration heat loss (W)

C_p = Spesific heat of dry air at constant pressure (W/kg°C)

t_{ex} = Expiration air temperature (°C)

t_a = Ambient air temperature (°C)

Heat loss or gain by heat conduction through the clothing

Heat loss or gain by heat conduction through the clothing depends on simple conduction principles which states that in the presence of a temperature gradient, heat transfer by conduction occur in the direction of decreasing temperature [50]. Therefore, the main equation for this mechanism can be states as it is given in equation 3.13;

$$K = \frac{1}{R_{Cl}} \,.\, A_{Du} \,.\, (t_s - t_{Cl}) \tag{3.13}$$

where, K = Heat loss or gain by conduction (W)

R_{cl} = Total heat transfer resistance from skin to outer surface of the clothed body (m^2°C/W)

t_s = Skin surface temperature (°C)

t_{cl} = Cloth surface temperature (°C)

Although heat loss or gain by heat conduction through the clothing depends on basic conductive heat transfer principles, transfer of dry heat between the skin and outer surface of the clothed body is quite complicated and heat transfer resistance should be determined. Yet, to simplify the calculations, in the literature, R_{cl} is given in the tables in terms of clothing index I_{cl} .According to studies by Fanger and Gagge, [14], [51] the correlation between total heat transfer resistance and clothing index is found as;

$$I_{Cl} = \frac{R_{Cl}}{0{,}18} \tag{3.14}$$

Here, I_{cl} is a dimensionless expression for the total thermal resistance from the skin to the outer surface of the clothed body. Fanger has estimated clothing values as given in the following table 3.3, for differ]ent clothing ensembles [14].

Table 3.3 : Data for different clothing ensembles with reference to Fanger.

Clothing ensembles	I_{cl} (clo) Fanger (1970)
Nude	0
Shorts	0,1
Typical tropical clothing ensemble	0,3 – 0,4
Apollo constant wear garment	0,35
Light summer clothing	0,5
Light working ensemble	0,6
U.S Army “Fatigues” Man’s	0,7
Combat tropical uniform	0,8
Typical business suit	1,0
Typical business suit + cotton coat	1,5
Light outdoor sportswear	0,9
Heavy traditional European business suit	1,5
U.S army standard cold-wet uniform	1,5 – 2,0
Heavy wool pile ensemble	3 – 4

As it can be seen from table 3.3, although given clothing values by Fanger are have a large scale including different clothing ensembles, it is not practical to use for typical clothing. Thus, withing new studies and experiments, ASHRAE 55 gives clothing insulation values for typical ensembles as it is given in table 3.4 [36].

According to ASHRAE 55 standards, clothing insulation values are classified taking the daily clothings on the basis. Accordingly different clothing combinations are presented in the standard.

Table 3.4 : Clothing insulation values for typical ensembles [36].

Clothing description	Garments Included	I_{clo} (clo)
Trousers	1) Trousers, short-sleeve shirt	0,57
	2) Trousers, long-sleeve shirt	0,61
	3) Trousers, long-sleeve shirt, suit jacket	0,96
	4) Trousers, long-sleeve shirt, suit jacket, vest, T-shirt	1,14
	5) Trousers, long-sleeve shirt, long-sleeve sweater, T-shirt	1,01
	6) Trousers, long-sleeve shirt, long-sleeve sweater, T-shirt, long underwear bottoms	1,30
Skirts/dresses	7) Knee-length skirt, short-sleeve shirt (sandals)	0,54
	8) Knee-length skirt, long-sleeve shirt, full slip	0,67
	9) Knee-length skirt, long-sleeve shirt, half-slip, long-sleeve sweater	1,10
	10) Knee-length skirt, long-sleeve shirt, half-slip, suit jacket	1,04
	11) Ankle-length skirt, long-sleeve shirt, suit jacket 1.10	1,10
Shorts	12) Walking shorts, short-sleeve shirt	0,36
Overall/coveralls	13) Long-sleeve coveralls, T-shirt	0,72
	14) Overalls, long-sleeve shirt, T-shirt	0,89
	15) Insulated coveralls, long-sleeve thermal underwear tops and bottoms	1,37
Atletic	16) Sweat pants, long-sleeve sweatshirt	0,74
Sleepwear	17) Long-sleeve pajama tops, long pajama trousers, short 3/4 length robe (slippers, no socks)	0,96

Heat loss/gain by radiation

Human body is subject to heat transfer by radiation with surroundings constantly. Since human body is mostly warmer than surrounding objects, it can be said that heat is lost from the body in most of the cases. The amount of heat lost is calculated by Stefan-Boltzmann law as given in the equation 3.15.

$$R = A_{eff} \,.\, \varepsilon \,.\, \sigma \, [(t_{cl} + 273)^4 - (t_{mrt} + 273)^4] \tag{3.15}$$

where, R = Heat loss or gain by radiation (W)

A_{eff} = The effective radiation area of the clothed body (m^2)

ε = The emittance of the outer surface of the clothed body

σ = The Stefan-Boltzmann constant ($W/m^{2}{}^{\circ}K^4$)

t_{mrt} = Mean radiant surface temperature (°C)

In equation 3.15 A_{eff} is defined as the effective radiation area of the clothed body and can be calculated as given in the following equation 3.16;

$$A_{eff} = f_{eff} \,.\, f_{cl} \,.\, A_{Du} \tag{3.16}$$

where, f_{eff} = The effective radiation area factor

f_{cl} = The ratio of the surface area of the clothed body to the surface area of nude body

In equation 3.16, the effective radiation area factor is another parameter that is difficult to obtain precisely. However, within the experimental studies, Fanger has found out that the effective radiation area factor is 0,696 for sedentary body posture, whereas, 0,725 for standing posture [14].

Heat loss/gain by convection

Heat loss or gain by convection mechanism is expressed with convection heat transfer formula as it is given in the equation 3.17, as it follows;

$$C = h_c \,.\, A_{Du} \,.\, f_{cl} \,.\, (t_{cl} - t_a) \tag{3.17}$$

where, C = Heat loss or gain by convection (W)

h_c = Convective heat transfer coefficient (W/m^2 °C)

In equation 3.17, convective heat transfer coefficient depends on air velocity. For lower air velocities, the heat transfer occurs via free convection. In this case, according to Nielsen and Pedersen's investigations h_{cl} can be taken as a function of the temperature difference between cloth surface temperature and ambient temperature [14], [52]. The correlation between convective heat transfer coefficient and mentioned temperatures is given in equation 3.18:

$$h_c = 2{,}05 \,.\, (t_{cl} - t_a)^{0{,}25} \tag{3.18}$$

Yet, when relative velocity in the subject place exceeds 2,6 m/s, convection turns into forced convection rather than free convection. In such a case, equation 3.17 is invalid and cannot be used. Therefore, another approach to obtain convection heat transfer coefficient by Winslow, Gagge and Herrington [53] is developed as given in equation 3.19;

$$\mathrm{h_c} = 10{,}4\,.\sqrt{v} \tag{3.19}$$

Where, v the relative velocity (m/s) and higher than the value of 2,6 m/s.

4. THERMAL COMFORT INDICATORS AND VARIABLES THAT AFFECT THERMAL COMFORT

Presented in the thermal comfort models and equations in more detail, there are several variables that affect thermal comfort. Researches indicate that internal body temperature of man is between 36,5 °C to 37 °C [45], [54]. Also skin temperature is normally varies from 31 to 34°C [54], [55]. Therefore, if heat is supposed to be lost from the human skin, then the temperature of the environment must be less than skin temperature. When the conditions are on the contrary state, it means that heat will be transferred to human body from the surrounding environment. In that context, body and skin temperature depend on air temperature and it is one of the most significant indicator in terms of thermal comfort. However, it is not the sole indicator. A vast number of researches have presented that, thermal sensation relies on a set of variables and can be represented as diverse indicators [7], [9], [14].

Thermal comfort indicators can be investigated in two common groups; internal environment variables and human related variables. Whereas, thermal comfort indicators are addressed in three titles; Predicted Mean Vote (PMV), Percentage of People Dissatisfied (PPD), Local Thermal Discomfort (LTD). These variables and indicators will be investigated hereafter comprehensively.

4.1 Internal Environment Variables

In the previous sections, it has been mentioned that internal environment plays an important role in thermal sensation of people living inside. Consequently, today's HVAC systems are designed according to thermal sensation of occupants. Researches and Fanger's pre-defined model have revealed that, thermal comfort depends on mainly four environment variables which are, indoor air temperature, relative humidity, air velocity and mean radiant temperature. Each term will be evaluated in the following sections one by one.

4.1.1 Mean indoor air temperature

As stated before, mean indoor air temperature is one of the key indicator and variable in the terms of thermal comfort. Mean indoor air temperature can be defined as the temperature of the air surrounding the human body. The average is with respect to location and time [36]. The air temperature in space is often the most significant environmental variable which effects thermal comfort of its occupants [56]. In addition, since it affects directly the convective dissipation, air temperature is a major factor in heat transfer [7].

4.1.2 Mean indoor relative humidity

Relative humidity is the ratio of the partial pressure of the water vapour in the air saturation pressure of water vapour at the same temperature and total pressure [36]. In other words, it is calculated as a ratio of the prevailing partial pressure of water vapour to the saturated water vapour pressure. The reason that mean relative humidity is a significant indicator in thermal comfort is when a liquid such as water or sweat in human body is heated it evaporates to a vapour which causes heat loss to the surrounding environment [9]. In that way, relative humidity takes an important role. Especially in hot summer days, thermo-regular system uses sweating as an evaporative cooling mechanism to regulate the body temperature and sweating is dependent on mean indoor relative humidity of the indoor environment. According to ISO 7730 standards, in temperatures lower than 26°C and moderate activity levels, every 10% increasing relative humidity is assumed to be warm as a 0,3°C rise in the operative temperature [37]. Yet, for higher temperatures, it is effect is thought to be greater.

4.1.3 Mean indoor air velocity

Another significant parameter for determination of thermal comfort is mean indoor air velocity or air speed which can be defined as the mean rate of the air movement in a zone. Air movement across the body might affect the heat transfer of body via convection mechanism. According to Szokolay, subjective reactions to air movement is given as it follows [7].

<0.1 m/s stuffy

to 0.2 unnoticed

to 0.5 pleasant

to 1 awareness

to 1.5 draughty

>1.5 annoying.

Air velocity is often increased by opening windows or use of fans in a space. In summer days, higher air velocities might be preferred due to its cooler effect in higher temperatures. With reference to ISO 7730 standards, Figure 4.1, represents the air velocity and its effects on the thermal sensation.

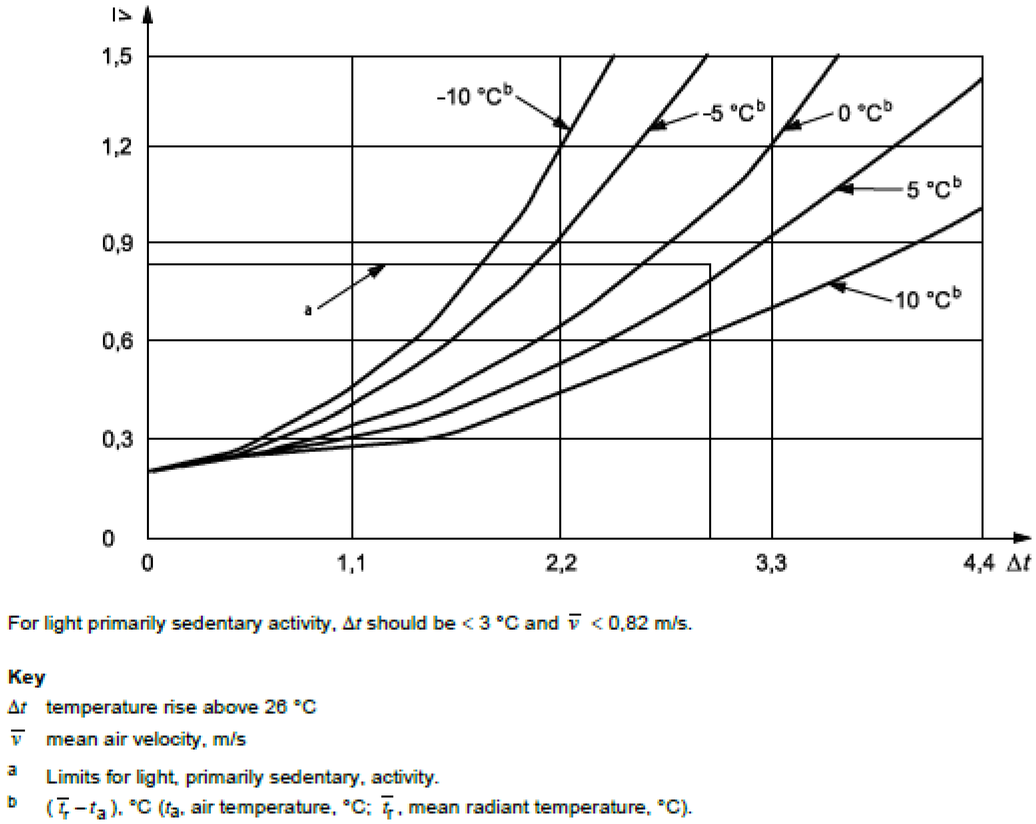


Figure 4.1 : Required air velocity to offset higher temperatures [37].

It should be noted that summer conditions are taken into account in Figure 4.1 with clothing level of 0,5 clo and 1,2 met units. On the basis of 26°C and 0,2 m/s air velocities, figure depicts how increased velocity offsets the effect of room temperature.

4.1.4 Mean radiant temperature

As well as indoor air temperature, Mean Radiant Temperature (MRT) should be taken into consideration in thermal comfort calculations. ASHRAE 55 defines MRT as "*the temperature of a uniform, black enclosure that exchanges the same amount of heat by radiation with the occupant as the actual enclosure."*[36] which can be simply interpreted as the temperature of surrounding surfaces that shapes radiation exchange between human body and its surroundings. MRT relatively affects all the radiant heat transfers from the surfaces and objects in space such as the walls, ceiling, windows, heaters, lights, equipment [56]. MRT cannot be measured directly and only can be obtained by globe thermometer.

4.2 Human Related Variables

As well as indoor environmental variables, human related variables also should be mentioned when addressing thermal comfort. Human related variables are examined under two titles; activity type and level and effect of clothing. Unlike environment conditions, these variables might change from person to person, even when they all stay in the same exact place. Thus, when assessing the total thermal comfort a large group, activity types of each person should be taken into account precisely.

4.2.1 Activity type and level

In the previous chapter, Fanger's heat balance equation was explained and it was shown that internal heat production also counts in thermal comfort calculations. Consequently, the indicator for internal heat production is activity type and level. Activity level is measured in terms of metabolic rate or 'met' [51]. One met unit is equal to 58,2 W/m2 and met rates are given in various tables for different type and level of works as it is previously represented in Table 3.1. Metabolic rate is the value of produced energy from human body and substantially dependant on activity.

4.2.2 Effect of clothing

The effect of clothing is another point in terms of heat transfer between human body and its environment. In the literature, clothing insulation can be found in units of 'clo' [53]. Mostly, it is not easy and practical to directly measure the value of clothing insulation, and consequently, formerly constituted tables are used for estimation of clothing value for occupants. Clothing value tables can be found on previous chapter, Tables 3.3 and Table 3.4.

4.3 Thermal Comfort Indicators

In the previous chapter, variables that affect thermal comfort is explained. According to pre-mentioned variables, a set of thermal comfort indicators are established. In this section, thermal comfort indicators will be evaluated.

4.3.1 Predicted mean vote (PMV) indicator

Predicted Mean Vote (PMV) term is developed by Fanger in the 1970's from several laboratory and climate chamber studies. In these studies, participants are exposed to different thermal environments with exactly the same level of activity and clothing insulation. Participants are made to vote on how they feel within the use of ASHRAE thermal sensation scale shown in the Figure 4.2.

-3	-2	-1	0	1	2	3
cold	cool	slightly cool	neutral	slightly warm	warm	hot

Figure 4.2 : ASHRAE Thermal sensation scale [36].

Shown in the Figure 4.2, ASHRAE thermal sensation scale has seven points; in which "0" is considered as ideal conditions for person (neither cold nor hot).

Using the heat balance equations, which are presented in Chapter 3.2.1.1, Fanger has built a mathematical model of the relationship between all the environmental and psychological factors as given in the following equation 4.1 and 4.2

$$\text{PMV} = (0{,}303\ e^{-0{,}036*M} + 0{,}028)[(M - W) - R - C - K - E - R_{es}] \tag{4.1}$$

or

$$\text{PMV} = (0{,}303\ e^{-0{,}036*M} + 0{,}028)\ S \tag{4.2}$$

Where, M = Metabolic rate (W/m^2)

W = Mechanical work (W/m^2)

R = Heat loss or gain by radiation

C = Heat loss or gain by heat conduction (W/m^2)

K = Heat loss or gain by convection (W/m^2)

E = Heat loss by skin diffusion (W/m^2)

R_{es} = Heat loss or gain by respiration (W/m^2)

S = Stored or lost energy in human body (W/m^2)

The PMV index evaluates the thermal comfort as a function of activity, clothing and four environmental parameters which are defined above. Based on the steady-state heat transfer between the body and environment, the PMV index is applied to predict the average value of the thermal comfort equation. The average value is given in the terms of ASHRAE sensation scale represented in the Figure 4.2.

4.3.2 Percentage of people dissatisfied (PPD) indicator

Based on PMV index, Fanger has proposed another indicator which estimates the predicted percentage of people dissatisfied (PPD). The PPD index is totally dependent on PMV and can be calculated using the following equation 4.3;

$$\mathrm{PPD} = 100 - e^{-(0{,}03353.PMV^4 + 0{,}2179.\ PMV^2)} \tag{4.3}$$

Equation 4.3, namely the relation between PMV and PPD index is also expressed in Figure 4.3 graphically.

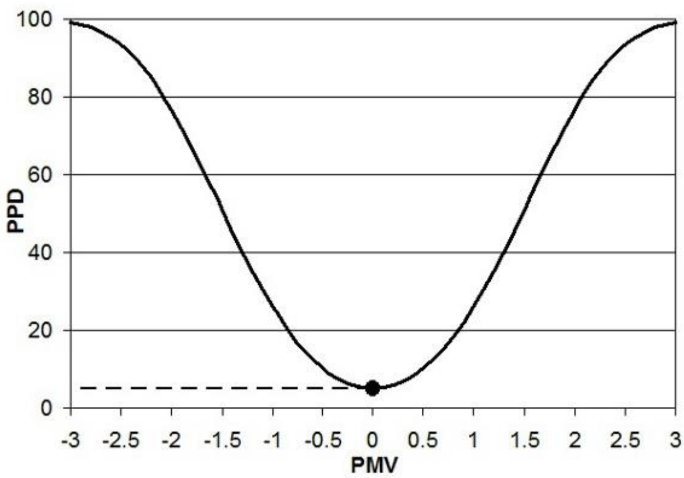


Figure 4.3 : Evolution of PPD on the basis of PMV index [57].

As it can be seen from Figure 4.3, the PPD index cannot reach zero value. Fanger, in his study mentions that issue as it is not possible to satisfy all people in a large group which share the same climatic conditions. Even with the perfect environmental system and conditions, it is impractical to attain a PPD value lower than 5% in mentioned group [14]. Therefore, although zero value of the PMV index is achieved, minimum obtainable value for PPD index is 5% as it is represented in Figure 4.3. Consequently, in a decent design, designer's aim should be keeping all occupants in a thermal comfort zone, yet, with an exception of 5% dissatisfied occupant.

4.3.3 Local thermal discomfort indicator

Despite a person's provided thermal neutrality, parts of the body can still be exposed to conditions which lead to thermal discomfort. Therefore, in conditions such as draught, high vertical temperature differences between head and ankles, too warm or too cool floors or too high radiant temperature asymmetry local thermal discomfort arises. ASHRAE defines local thermal discomfort as *"A person may feel neutral as a whole but still feel if one or more parts of the body are too warm or too cold."*[46].

To prevent local thermal discomfort, there are a set of recommendations provided with respect to radiant temperature asymmetry, vertical temperature differences and draught. In Table 4.1, recommended radiant temperature asymmetry is represented on the basis of ASHRAE standards.

Table 4.1 : Recommended radiant temperature asymmetry [36].

Radiant Temperature Asymmetry (°C)			
Ceiling warmer than floor	Ceiling cooler than floor	Wall warmer than air	Wall cooler than air
< 5	< 14	< 23	< 10

As it can be seen in Table 4.1, radiant temperature difference in ceiling should be less than 5°C in ceilings warmer than floor; whereas, 14°C is recommended for ceilings cooler. In walls, 23°C and 10°C is suggested for wall warmer and cooler than air respectively.

With radiant temperature asymmetry, warm and cool floors are also should be mentioned. Figure 4.4 shows the expected dissatisfaction percentage of occupants with respect to the floor temperature.

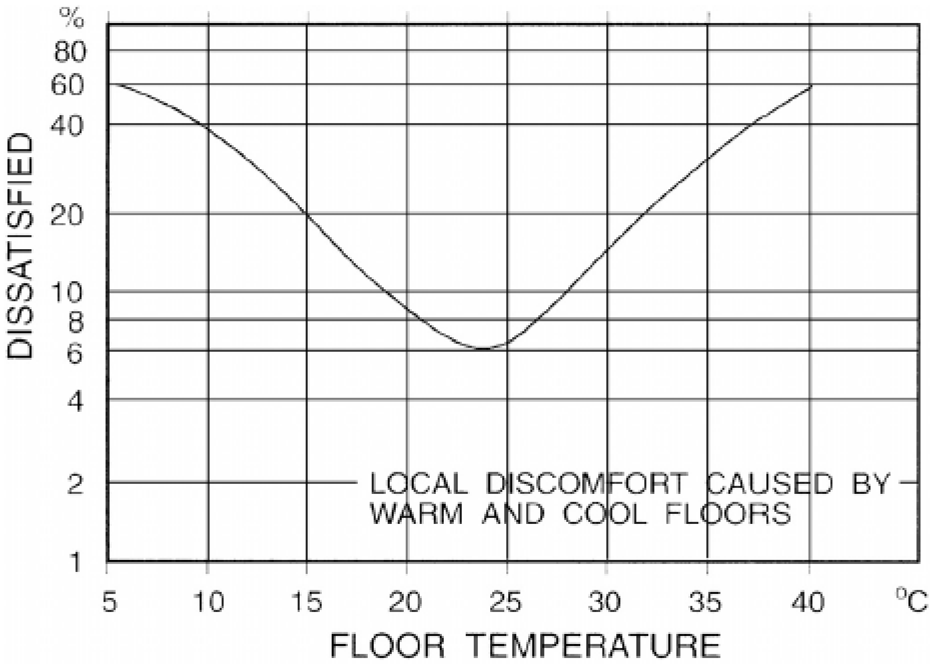


Figure 4.4 : Local discomfort and dissatisfaction percentages due to floor temperature [36].

In Figure 4.4, it can be seen that the optimum floor temperature is between 23 and 24°C; yet, with the acceptance of maximum 10% dissatisfied occupants, 19 – 29 °Cs are suggested in order to prevent local thermal discomfort.

Similarly, air temperature difference between head and feet and its relation with the local discomfort is given in Figure 4.5.

Figure 4.5 depicts the expected percentage of occupants who are dissatisfied due to the air temperature difference where the head level is warmer than ankle level. ASHRAE 55 assumes maximum 5% of occupants should be dissatisfied by the vertical temperature difference; therefore, allowed range is up to 3°C.

As represented in the figure, local thermal discomfort problems occur by local air temperature difference and accordingly, percentage of people dissatisfied is significantly increases. In 6 ˚Cs temperature difference percentage of people dissatisfied rises to 40% which indicates that almost half of the occupants are not satisfied with the present thermal conditions.

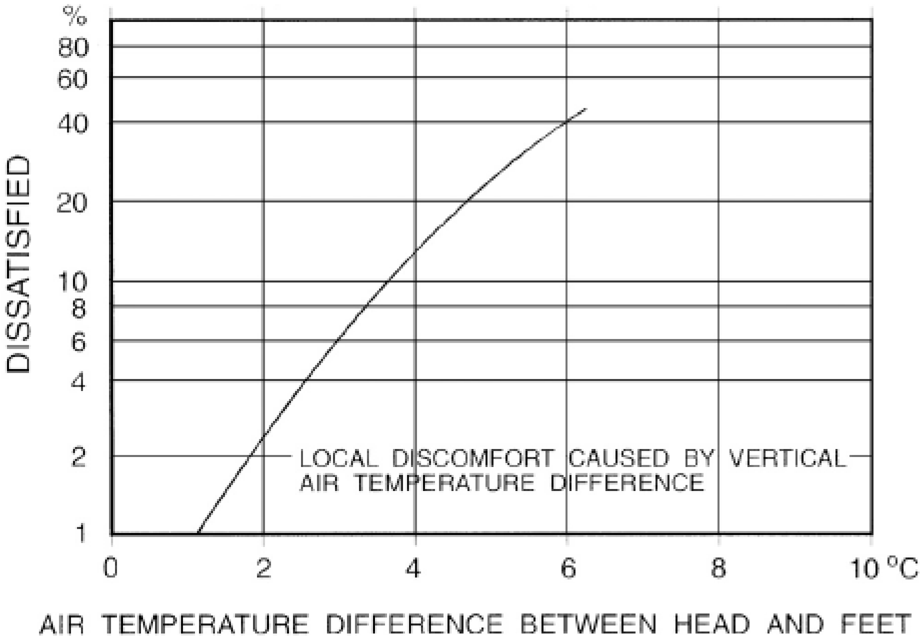


Figure 4.5 : Local discomfort and dissatisfaction percentages due to vertical air temperature difference [36].

Summarizing mentioned local thermal discomfort indicators and recommendations, Figure 4.6 represents all the reference values given in ASHRAE standards.

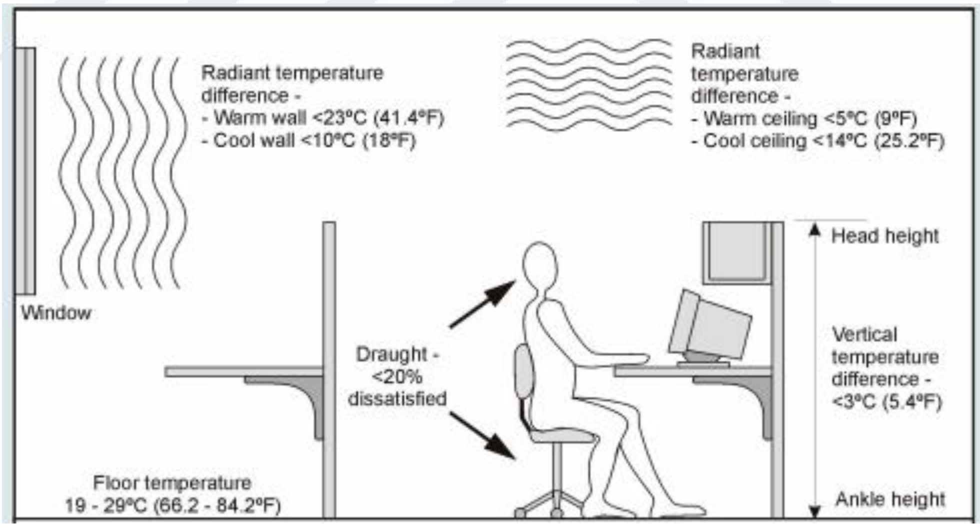


Figure 4.6 : Recommended local thermal conditions in ASHRAE 55 [44], [58].

5. EVALUATION OF THERMAL COMFORT: CASE STUDY

In this thesis, in order to verify previously explained thermal comfort calculation method and analyze the effect of energy efficiency strategies on thermal comfort, Merkezi Derslik (MED) building is selected. In the current chapter, specifications of the chosen building will be evaluated.

5.1 General Overview of the Building

MED is a newly constructed building in the Istanbul Technical University Ayazaga Campus and hosts a variety of areas such as lecture theatres which are used by multidisciplinary students with common lectures, seminar rooms, computer labs, academician rooms and common zones like cafeterias and stationaries. The building consists of 5 levels: two floors with two mezzanine floors and the basement. This multifunctional building is located at the junction point of pedestrian roads and vehicle roads in the campus. Location of the building in the campus is given in the following Figure 5.1.

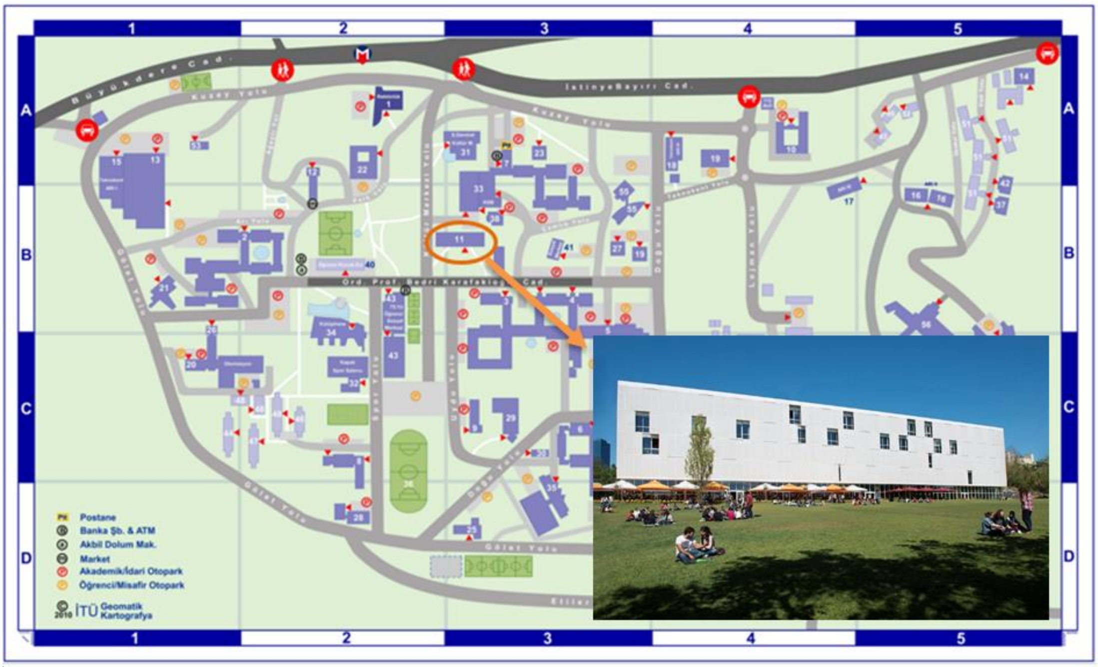


Figure 5.1 : Location of MED in Ayazaga Campus.

As shown in the Figure 5.1 and 5.2, it is oriented in the Northwest-Southeast direction and any façade of the building is not attached to another building. In the southeastern frontage, green area for students takes place.

Figure 5.2 : Location and Cross Section of MED.

MED consists of 128 different rooms, in which 16 of them are used as lecture hall for students. As aforementioned, besides common spaces, there are distribution zones such as corridors and residential places such as office rooms are located in the building. Typical usage of a floor and its mezzanine floor is depicted in the Figure 5.3.

Represented in the Figure 5.3, amphitheatre type classrooms mainly takes place in the northern side of the building and ceiling height of the mentioned classrooms reaches up to 8.1 meters. Namely, the floor height of the amphitheathre type classrooms covers two floors.

Whereas in the southern part of the building, seminar rooms and academician rooms in the mezzanine floor are located. Ceiling height of seminar and academician rooms is 3,6 meters.

In addition to classrooms and offices, there are other rooms in MED as it is shown in Figure 5.3. In the middle part of each floor, toilets, storage rooms and mechanical rooms are placed, yet, these zones have no systems with respect to heating and cooling purposes.

Figure 5.3 : Typical usage distribution of floors.

Building is located in the 2nd climate zone according to TS 825, Turkish Thermal Insulation Standards [59] and 4th according to ANSI/ASHRAE/IESNA Standard 90.1-2007 [60]. Istanbul has a mild climate, which summer months are generally warm and humid, yet, very little rain drops especially during July and August [61]. However, in winter months, the weather is mostly cold and wet, but in comparison to Turkey's other cities it can be referred as mild. According to Turkish State Meteorological Service, average temperatures and rainy days between years 1950 and 2014 is given in the Table 5.1. As it is seen from the following Table 5.1, in Istanbul, air temperature hardly decreases down minus temperatures. Whereas, average temperature is between 5 to 8 ˚Cs in winter and this value corresponds to 20-23˚Cs in summer months.

Table 5.1 : Climatic conditions of Istanbul [61].

ISTANBUL	Jan	Feb	Mar	Apr	May	June	July	Aug	Sep	Oct	Nov	Dec
	Average Temperatures Between Years 1950 – 2014											
Average Temperature (°C)	5,6	5,7	7,0	11,1	15,7	20,4	22,8	23,0	19,7	15,6	11,4	8,0
Average Highest Temperature (°C)	8,5	9,0	10,8	15,4	20,0	24,5	26,5	26,7	23,6	19,1	14,7	10,8
Average Lowest Temperature (°C)	3,2	3,1	4,2	7,7	12,1	16,5	19,5	20,0	16,8	13,0	8,9	5,5
Monthly Average Rainy Day	17.5	15.2	13.8	10.4	8.1	6.0	4.2	4.9	7.3	11.2	13.3	17.3

In addition to this long-term evaluation for air temperatures, another study is conducted regarding Heating Degree Days (HDD) and Cooling Degree Days (CDD) for Istanbul. HDD is a number of degrees that a day's average temperature is below 18°C. For CDD this value equals to higher than 22 °C [62]. According to the study by Sensoy, which had been held between 1975 and 2005, , HDD is 1937, whereas CDD is 44 in Istanbul Sarıyer province [62].

5.2 Design Conditions

In MED, design conditions are chosen mainly in accordance with thermal comfort standards. According to TS EN 15251 standard, firstly building category should be defined as it is given in Table 5.2.

Table 5.2 : Building categories according to EN 15251 [63].

Category	Explanation
I	High level of expectation and is recommended for spaces occupied by very sensitive and fragile persons with special requirements like handicapped, sick, very young children and elderly persons
II	Normal level of expectation and should be used for new buildings and renovations
III	Acceptable, moderate level of expectation and may be used for existing buildings
IV	Values outside the criteria for the above categories. This category should only be accepted for a limited part of the year

From defined categories in Table 5.2, MED can be regarded as a new building with normal level of expectation, therefore, it is suitable to recall “Category II”. In TS EN 15251, according to building category different design conditions are referred. Recommended indoor temperatures for energy calculations are given in Table 5.3 with reference to TS EN 15251.

Table 5.3 : Temperature ranges for hourly calculation of cooling and heating energy in three categories of indoor environment [63].

Type of Building or space	**Category**	**Temperature range for heating (°C)**	**Temperature range for cooling (°C)**
Offices and spaces with similar activity (single offices, open plan offices, conference rooms, auditorium, cafeteria, restaurants, classrooms)	I	21,0 – 23,0	23,5 – 25,5
	II	20,0 – 24,0	23,0 – 26,0
	III	19,0 – 25,0	22,0 – 27,0

Since MED is regarded as “Category II”, temperature range can be chosen 20 – 24° C for heating season and 23 – 26 ° C for cooling season. In this regard, set-point is selected 22° C for heating and 24° C for cooling in MED. Also it should be noted that, taking EN 15251 and ISO 7730 as a reference clothing values are considered as “1,0 clo and 0,5 clo” for heating season and cooling season respectively.

As well as heating, ventilation is also addressed in afore-mentioned standards. Basic required airflow values are given in following Table 5.4.

Table 5.4 : Recommended ventilation rates with respect to building categories [63].

Type of space	**Category**	**Airflow per person ($l/s/m^2$)**	**Very low polluting building ($l/s/m^2$)**	**Low polluting building ($l/s/m^2$)**	**Non low polluting building ($l/s/m^2$)**
Class room	**I**	5	5,5	6	7
	II	3,5	3,8	4,2	4,9
	III	2	2,2	2,4	2,8

According to standards, supply air recommended to be 3,5 l/s per space area. In MED, ventilation system is designed in construction phase so in modelling process real ventilation rates will be used. Actual ventilation rates vary from space to space; yet, mostly it is highly consonant, even better compare to the referred standards. For instance, for amphitheater classrooms, ventilation rate is 2,4 l/s/m^2. Actual ventilation rates for each zone will be given in Appendix B comprehensively. Along with ventilation rates, actual occupancy rates can be found on Appendix B for each different zone in MED.

As far as lighting is concerned, two different types of lighting systems have been used in MED. First one is the main lighting system with florescent lamp units with 58W power and used broadly in the building. The second type is hanging light systems which are used in the halls and corridors. Each type can be seen in Figure 5.4.

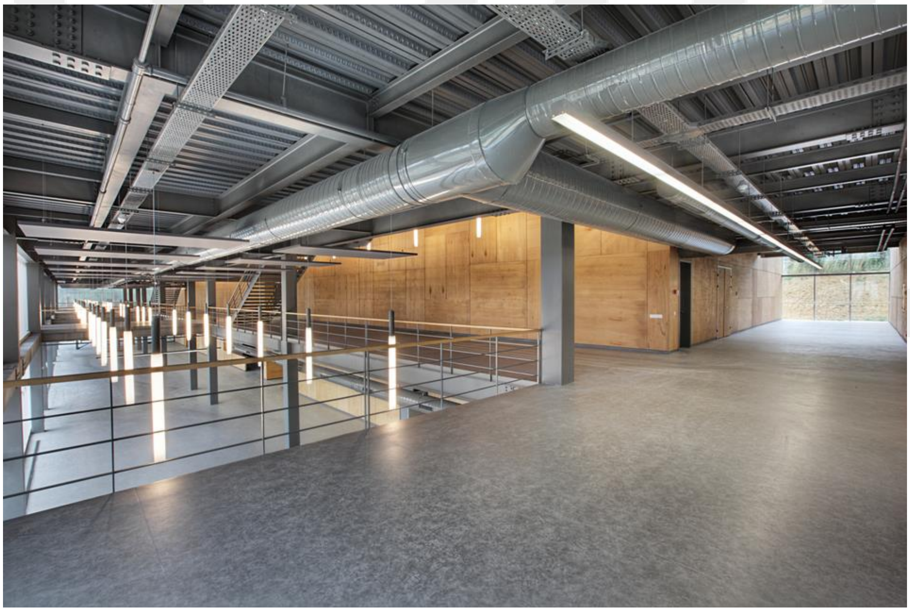


Figure 5.4 : Lighting systems in MED.

Each room has different numbers of lighting units; hence, each room is evaluated separately based on the actual lighting units. Lighting rates for each zone can be found in Appendix B.

5.3 Thermo-physical Properties of Building Materials

The building is constructed as steel construction and covers nearly 10.000 m^2 area. Steel construction of MED during construction phase is given in the following Figure 5.5.

Figure 5.5 : Steel construction of MED.

MED has various types of construction elements regarding external, internal and curtain walls and windows. The thermal properties of wall constructions are given in Table 5.5.

Table 5.5 : Thermal properties of wall constuctions.

Construction Type	U-value (W/m^2K)
Internal walls	0,37 ÷ 1,77
External walls	0,34
Floors	0,87
Earth-contracted floors	0,29
Roof	0,20

In MED, there are 28 different types of internal and external walls, thus, overall heat transfer coefficient (U-value) of the internal wall constructions differs between 0,37 and 1,77 W/m^2K.

Along with solid constructions, thermal property of glazing is significant in a building. For MED, specifications of glazing is given in Table 5.6 as it follows.

Table 5.6 : Thermal properties of glazings.

Construction Type	U-value (W/m^{2}K)	Total Solar Transmission (SHGC)	Light Transmission
Windows and curtain walls	1,6	0,54	0,77

Although internal and external walls differs from façade to façade, box profile construction is used in general. In terms of insulation, stone wool is prefferred in the façades. Based on the orientation of the building, western, southern and eastern façade consist of curtain walls which let sunshine inside through the surface and provide daylighting for classrooms. Besides, for southern, western and a portion of eastern façade, a mobile shading element is constructed as seen in the Figure 5.6.

Figure 5.6 : South-eastern façade and shading element.

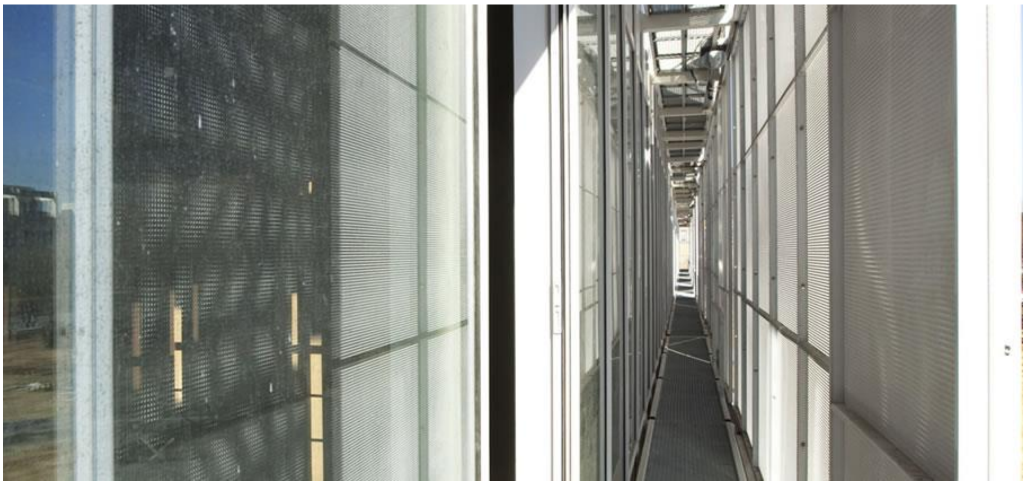


Figure 5.7 : Double-layer façade of MED.

As shading element, movable perforated metal panel is used for both blocking sun radiation and reducing cooling loads for hot summer days as well as letting daylight in with its transperant contexture through double-layer façade. Double-layer façade and the space between curtain walls and perforated metal panel can be seen in Figure 5.7.

5.4 Building HVAC System

For MED's heating, cooling and ventilating needs various types of mechanical systems are present. Heating is provided by both Fan-Coil Units (FCU) which are fed by two natural gas boilers and Variable Refigirant Flow (VRF) air conditioning system. Natural gas boilers are placed in the mechanical room in the basement floor and composed of two 500 kW Viessman Vitoplex 100 Modulating Natural Gas Boiler. Spesifications of natural gas boilers are given in the following Table 5.7.

Table 5.7 : Technical properties of boiler [64].

Technical data of boiler			
Rated output	from	kW	460
	to	kW	500
Rated thermal load	from	kW	500
	to	kW	546
Max. flow temperature		˚C	120
Max. operating pressure		bar	4
Standard efficiency		%	94
Standby loss		%	0,13

Regarding cooling needs, mainly VRF air conditioning systems are installed in the building. VRF is an air conditioning system developed by Daikin in which refrigerant flow can be adjusted through copper pipes. Unlike split air-conditioning units, multiple VRF air-conditioners can be coupled to single condenser unit. Given in the Figure 5.8, in MED, condenser units are placed in both backside and roof of the building.

Figure 5.8 : VRF outdoor units in MED garden.

VRF capacities vary according to thermal need of the zone. Indoor units are selected as compact 4 way casette types. Specification of VRF system units are given in the following Table 5.8.

Table 5.8 : Technical spesifications of VRF system[65].

Technical data of VRF system units			
Total cooling capacity	from	kW	14
	to	kW	45
EER	from	W/W	3,84
	to	W/W	3,28
Total heating capacity	from	kW	16
	to	kW	50
COP	from	W/W	4,17
	to	W/W	3,52
Air flow	from	m^3/h	9000
	to	m^3/h	13000
Operating range	˚C		-5÷43

As far as ventilation is concerned, plenums that located in each floor in mechanical rooms take place. Intake air is provided by suction nozzle located in the roof and used air is exhausted through another line from offtake channel. Percantage of outdoor air is set via an automation system. Intake air is first taken to plenums where through a heat exhanger, exhaust air's heat can be transffered to intake air without blending. Then, heating serpantines which are fed by natural gas boilers heat the air till service temperature. A representation of this system is given in Figure 5.9.

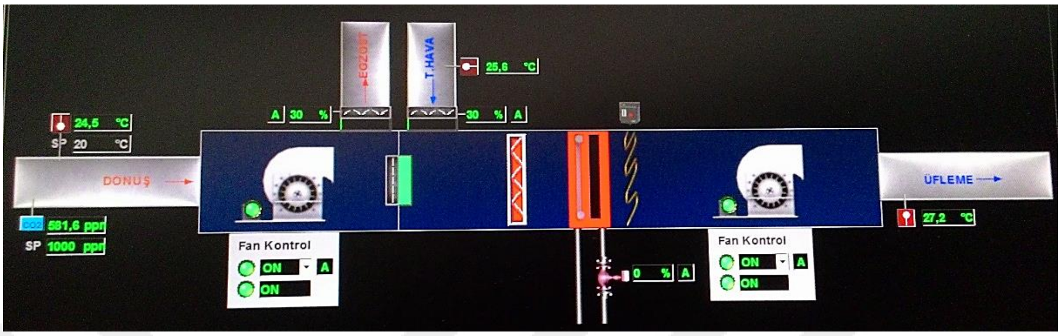


Figure 5.9 : Scheme of the ventilation system.

5.5 Simulation tools and selection of applicable simulation tool

Building modelling and simulation is an effective way to examine a given building in terms of architectural, mechanical and civil engineering aspects. As a part of this, recently, energy simulation tools are widely have been used in order to assess building's performance, determine energy consumption and demands. These simulation tools are not only can be used during life-cycle time of the building, but also during pre-construction phases. By this way, energy simulation tools may show which system would be the most efficient solution. Before construction works, in the preliminary design stage, from passive solutions (e.g. orientation of the building, window to wall ratio) to active solutions (e.g. mechanical system selection, control strategies) course of action can be determined thanks to results of generated energy model. Simulation programs enable users to create dynamic interactions between people, equipment, environment, HVAC system and the building as indicated in the Figure 5.10.

Developing technology in building information modelling area has led better understanding of how building operates or will operate in a given circumstance. As well as consumption and demand-side calculation of the building, simulation tools can also predict the thermal comfort parameters such as ambient temperature or mean radiant temperatures for a given zone dynamically.

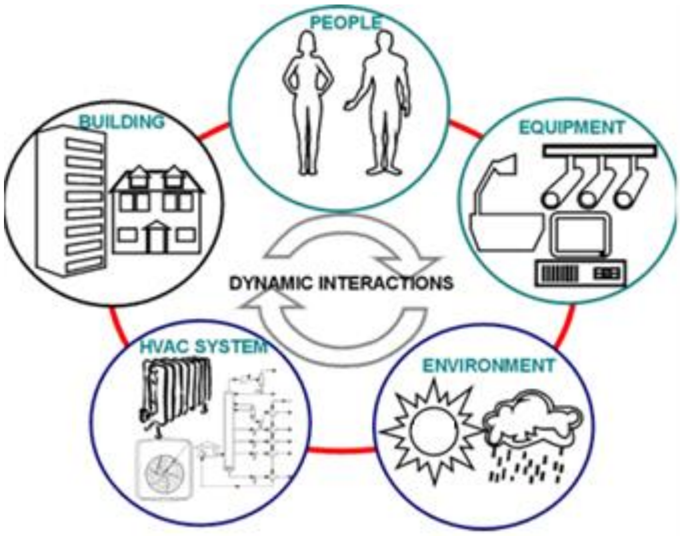


Figure 5.10 : Dynamic interactions via energy simulation tools [66].

These values are calculated with the help of a simulation engine. Energy consumption of the building depends on numerous parameters which are embedded into the engine, yet, these parameters can be summarized in general as follows [67].

- Building geometry and orientation.
- Weather condition.
- Internal loads.
- HVAC systems and configuration.
- Operating strategies and schedules.
- Simulation specific parameters. (i.e. numeric convergence tolerances, workflow).

These six bullet-points summarized above can be thought as main parameters for performing simulation. Building geometry and orientation, weather condition and internal loads are significant when determining heating and cooling loads precisely. Load calculation is followed by plant sizing and peak design loads for equipment. Consequently, HVAC system can be designed and configured.

Simulation engines work with respect to mathematical and thermodynamic algorithms and use above summarized inputs. Based on these inputs, the engine work flow is represented in the Figure 5.11.

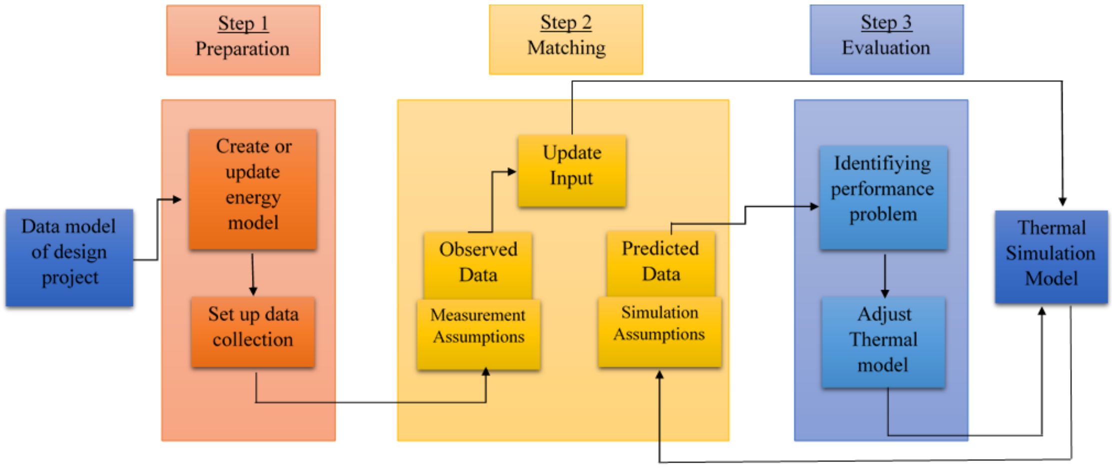


Figure 5.11 : Workflow to produce a thermal simulation model [67].

Data model of design project is derived from internal and external parts of the building. Each simulation programme has climatic data embedded to tool for particular places which plays a key role when assessing external loads. Embedded data of a region is often taken from meteorological services statically for a long period of time. If data is not given in the default program, it can be found on the web and uploaded by user as well. In addition, during project's commissioning, measuring and verification phase, weather data can be interchanged to measured real-time weather data by users in particular some software. Besides external loads, internal loads must be taken into consideration in the model. Internal loads cover a large portion, ranging from lighting or equipment loads to loads that rely on occupation and behaviour of occupants. Internal loads might occur both as latent loads and sensible loads.

Recently, there are substantial amount of simulation tools which are capable of performing energy and thermal simulations. Each simulation tool has its own capabilities and spesifications.

Simulation tools mainly depend on simulation engines and according to application areas, different engine models are developed. As an example, while COMSOL has capable of simulating the heat transfer via finite element approach, eQuest uses DOE-2 engine which enables users to simulate the dynamic energy flow.

Widely known and most used tools according to their application areas and spesifications can be found in the following Table 5.9.

Table 5.9 : A list of simulation tools and application areas.

Tool	Application
COMSOL	Multiphysics, heat transfer and finite element simulation and modelling
DesignBuilder	Building energy simulation and environmental design, 3-Dimensional Modelling, solar shading, thermal design and analysis, heating and cooling loads, natural and artificial lighting, Internal air, mean radiant and operative temperatures, humidity, CO2 emissions, solar shading, heat transmission, comfort studies
Ecotect	Environmental design and analysis, thermal design and analysis, 3D modelling; Solar control, overshadowing, natural and artificial lighting prevailing, winds & air Flow, life cycle assessment, life cycle costing, scheduling, geometric and statistical acoustic analysis.
eQUEST	Energy performance, simulation, energy use analysis, conceptual design performance analysis, 3D Modelling, thermal design and analysis, heating and cooling loads, Solar control, overshadowing, Lighting system, life cycle assessment, life cycle costing, Scheduling.
EnergyPlus	Energy Simulation, thermal design and analysis, Heating and cooling loads, Validation; Solar control, Overshadowing, Natural and artificial lighting, Life cycle assessment, Life cycle costing, Scheduling.
ESP-r	Environmental Design, 3D Design, thermal design and analysis, heating and cooling loads, Solar control, lighting, natural ventilation, combined heat and electrical power generation and photovoltaic facades, acoustic analysis, life cycle and environmental impacts assessments.
Green Building Studio	Environmental Design, thermal analysis, annual energy consumption (electric and gas), Carbon emissions, day lighting, water usage and cost, Life cycle costing, natural ventilation.
IES VE	Thermal design and analysis, heating and cooling loads, CO2, Validation; Solar, Shading, Lighting, Airflow, Life cycle costing, Scheduling, fire evacuation.
TRACE 700	Environmental design, 3D Model (3D Design), thermal design and analysis, heating and cooling, life cycle costing, plants system.
TRNSYS	Environmental design, 3D Model (3D Design), thermal design and analysis, heating and cooling loads, Solar control, overshadowing, prevailing winds & air Flow, electrical, photovoltaic, hydrogen systems, Life cycle costing.
Riuska	Environmental design, 3D Model, thermal design and analysis, heating and cooling loads, validation; Solar control, overshadowing, lighting, life cycle assessment, life cycle costing, scheduling.

Capabilities of each programme are given in the table above. Each programme has its own advantages. DesignBuilder/EnergyPlus and eQuest are compared in the Table 5.10 according to their specifications [68].

Table 5.10 : Comparison of DesignBuilder/EnergyPlus and DOE-2 engine [68].

Specification	DesignBuilder/EnergyPlus	eQuest (DOE-2 engine)
HVAC Loads	Works with reference to heat balance method which is more accurate. Capable of performing radiant and convective calculations for each surface.	Works with reference to transfer function method which is an approximation of the heat balance method and includes specific weighing factors that can be prone to user error.
Integrated simulation of loads and systems	Building response to thermal loads is calculated simultaneously with system operation	Building response to thermal loads is calculated independent to system operation.
Thermal Comfort	With EnergyPlus tool, capable of calculating thermal comfort parameters. Enables user to define activity levels as well as clothing rates for thermal comfort calculations. Also calculates surface temperatures and thermal comfort can be evaluated for each single zone.	Gives mean air temperature for each zone hourly, yet, cannot model thermal comfort.
HVAC Systems	Offers two types of HVAC system design; basic and detailed. Allow users to model mechanical system type of each thermal zone.	Mechanical system must be designed by selecting one of the pre-defined systems.
Natural Ventilation	Natural ventilation can be modelled with airflow network which enables wind and buoyancy-driven airflow calculations to be performed simultaneously.	Simplified natural ventilation with operable windows can be modelled.
Shading control	DesignBuilder offers more specific and various types of shading and shading controls.	Only limited shading controls.
Visual comfort	Capable of assessing visual comfort for each zone.	Cannot evaluate visual comfort.

In this study, due to remarkable features in thermal comfort assessment and ease of use, DesignBuilder which relies on EnergyPlus module when assessing the calculations will be used. In the following section, DesignBuilder, EnergyPlus and their features will be explained.

5.5.1 Design Builder

DesignBuilder is a functional simulation tool which enables modelling environment and accordingly calculates various environmental performance data such as annual energy consumption, HVAC component analysis and details. The software has been released as first version in United Kingdom as a Graphical User Interface to the EnergyPlus simulation engine in 2005.

With the help of tools, Design Builder has a great variety of features. Some typical uses of Design Builder are given as it follows[69];

- Assessing and calculating building’s annual consumption and demand.
- Thermal simulation for naturally ventilated buildings.
- Solar gains and irradiation assessment.
- Thermal gains and losses of the building.
- Visualisation of site layouts and evaluating façade options.
- Dynamic thermal comfort analysis.
- Daylighting analysis and solar shading.
- Calculating equipment sizes for heating and cooling systems.

Design builder has a user-friendly interface. With “Learning mode” definition of each tab can be seen on the right hand side of the screen. Interface of the program is represented in the Figure 5.12.

Main screen is composed of a variety of sections which are basically; menu tab, toolbar, model data tabs, edit screen and 3D visualisation of the model, navigator panel, status bar, screen tabs and info panel.

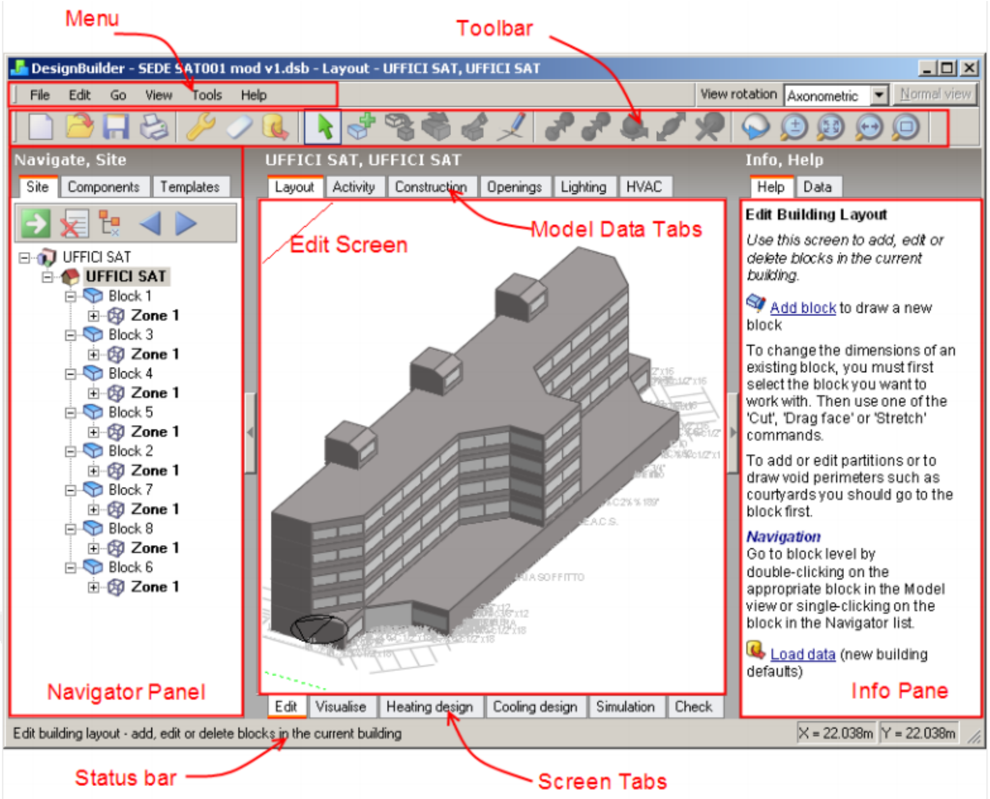


Figure 5.12 : Interface of DesignBuilder.

DesignBuilder has 4 different modules as shown in the Figure 5.13, which needs a licence for each model intended to be used.

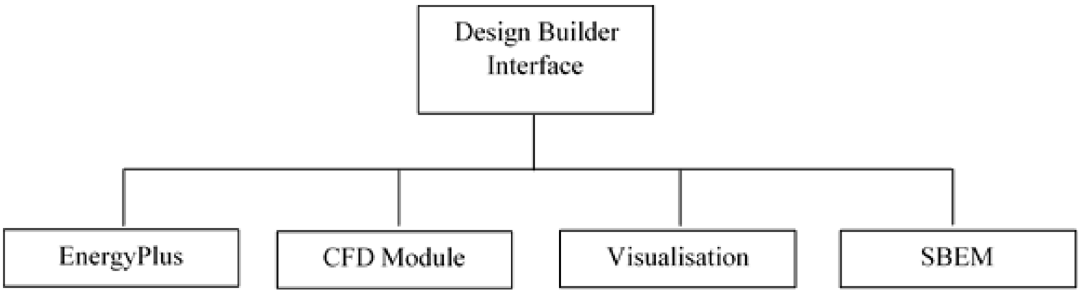


Figure 5.13 : Design Builder Modules.

These four modules are EnergyPlus, Computational Fluid Dynamics (CFD) Module, Visualisation and Simplified Building Energy Modelling (SBEM). EnergyPlus, which will be covered comprehensively in following part, is a tool that performs thermal and energy related simulations. Computational Fluid Dynamics module as the name suggests performs CFD analysis with a family of methods to calculate temperature, air velocity or specify other kind of fluids' behaviour. As well as thermal analysis, Design Builder can be used for visualisation by architects by elaboration of 3D model, rendered images and site shading analysis. Another module of the software is SBEM and it is mostly operated by energy assessors in order to get an insight and practical results of building's energy consumption.

In this thesis, since building's overall consumption, loads and thermal comfort parameters will be taken into consideration in detail, EnergyPlus module will be used. Next section stands for general overview of EnergyPlus and its structural and working principles.

5.5.2 EnergyPlus

The EnergyPlus is a software that consists of a collection of many program modules working together to calculate the energy demand for heating and cooling purposes. It is established by U.S Department of Energy and can be used as a stand-alone programme as well. However, due to ease of use and lack of graphical interface, EnergyPlus is often used as an embedded tool with pre-defined DesignBuilder software. The tool does calculation by simulating the building and concerning thermal and energy systems in changing environmental and operating conditions. The basis of the simulation relies on fundamental heat balance principles [70]. Schematic modules of the tool are represented in the Figure 5.14.

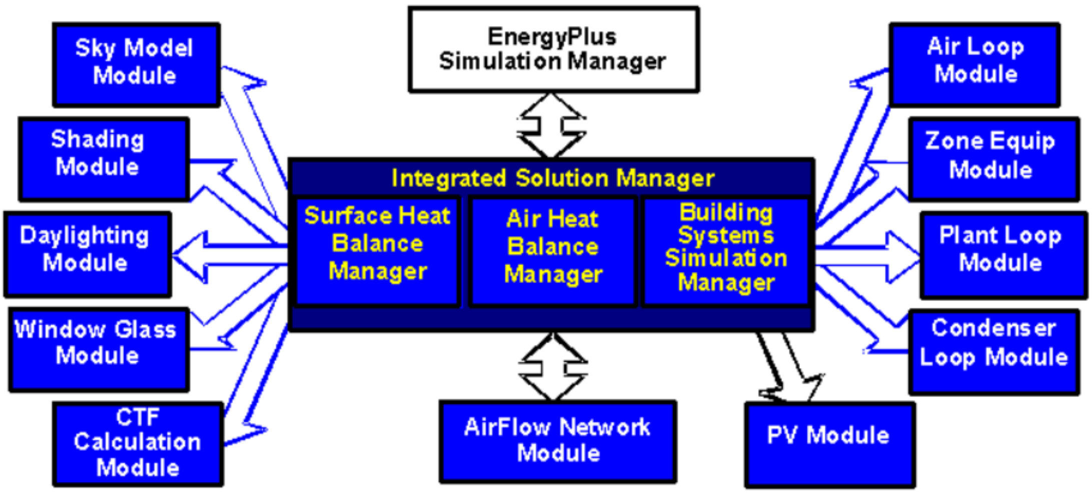


Figure 5.14 : Schematic Modules of EnergyPlus.

Similar to other simulation programs, it performs energy analysis based on user's description of a building from the perspective of the building's physical make-up and associated thermal systems. EnergyPlus calculates the heating and cooling loads necessary to maintain thermal control set points.

EnergyPlus has considerable amount of features in which some of them are listed as follows and given in the EnergyPlus Manual and in the webpage[70].

- Integrated, simultaneous solution where the building response and the primary and secondary systems are tightly coupled.
- Sub-hourly, user-definable time steps for the interaction between the thermal zones and the environment and variable time steps for interactions between the thermal zones and the HVAC systems.
- ASCII text based weather, input, and output files that include hourly or sub-hourly environmental conditions, and standard and user definable reports, respectively.
- Heat balance based solution technique for building thermal loads that allows for simultaneous calculation of radiant and convective effects at both in the interior and exterior surface during each time step.
- Transient heat conduction through building elements such as walls, roofs, floors, etc. using conduction transfer functions.
- Improved ground heat transfer modelling through links to three-dimensional finite difference ground models and simplified analytical techniques.
- Combined heat and mass transfer model that accounts for moisture adsorption/desorption either as a layer-by-layer integration into the conduction transfer function.
- Thermal comfort models based on activity, inside dry bulb, humidity, etc.
- Anisotropic sky model for improved calculation of diffuse solar on tilted surfaces.
- Daylighting controls including interior illuminance calculations, glare simulation and control, luminaire controls, and the effect of reduced artificial lighting on heating and cooling.
- Loop based configurable HVAC systems (conventional and radiant) that allow users to model typical systems and slightly modified systems without recompiling the program source code.

- Atmospheric pollution calculations that predict CO2, SOx, NOx, CO, particulate matter, and hydrocarbon production for both on site and remote energy conversion.

5.6 Generation of Simulation Model

In this section, generation of simulation model will be evaluated with pre-defined simulation tool DesignBuilder. MED is generated in DesignBuilder with following six footsteps as it follows;

1. Adjusting building's site location: In the first phase, building's site location and orientation is introduced to the programme. DesignBuilder has Istanbul's default site location and hourly weather data.
2. Introducing building footprints: As a second step, building's floor plans are imported from the CAD file to DesignBuilder, based on the actual state of the building.
3. Thermal zoning: After introduction of building floor plans, zoning takes place. In MED each thermal zone is created separately.
4. Definition of construction elements: Once floor plans and spaces are introduced to the tool, definition of construction elements must be done. According to actual conditions and thermal properties of the construction elements which are given in previous chapter, each construction element is defined uniquely.
5. Adjusting each thermal zone specifications: In step three each zone was already created separately. Yet, in terms of usage or another point, each zone might have different specifications. (i.e. different occupancy rates, set-points, lighting levels) In this stage, each thermal zone is adjusted according to real case which is also mentioned in previous chapter in design conditions.
6. Defining HVAC system: In this point, building's mechanical system is introduced in DesignBuilder with reference to the actual HVAC system placed in the building.

Following above-mentioned methodology, MED is modelled in DesignBuilder which is also represented in Figure 5.15.

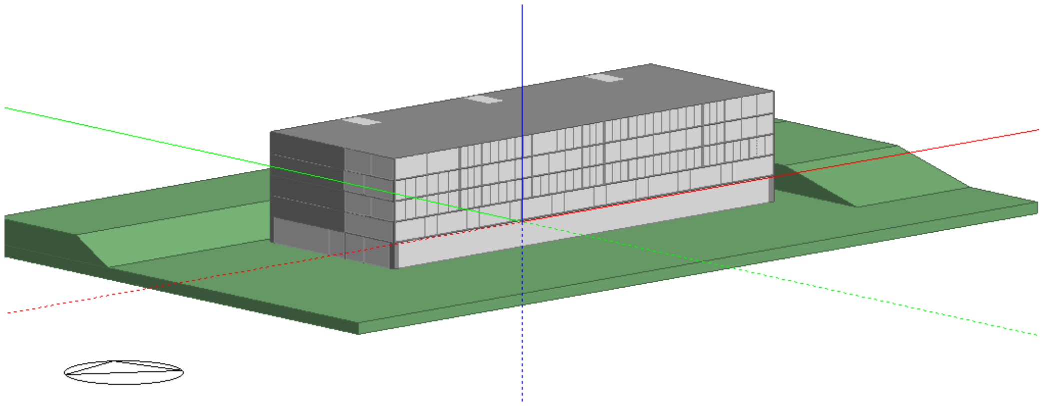


Figure 5.15 : DesignBuilder model of MED.

5.7 Measurement

In order to check the validation of DesignBuilder model of MED, temperature and relative humidity measurement have been carried out in one of the amphitheater classrooms in MED. Measurement is conducted in “AMFI 12” which is located in northern part of the building and second floor. Its location and floor plan can be seen in the following Figure 5.16.

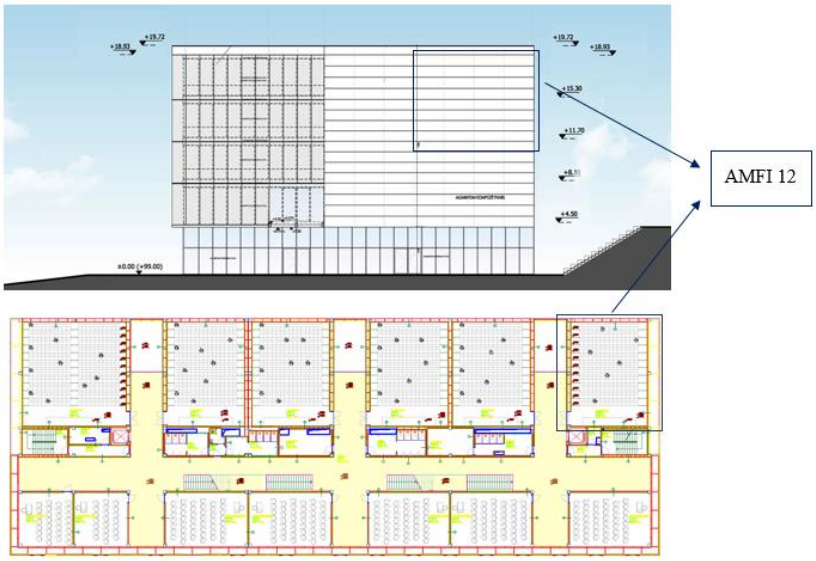


Figure 5.16 : Location of the measured zone,“AMFI 12”.

Measurement is conducted with a “RHT 20 Humidity and Temperature Data Logger” which is represented in Figure 5.17 and capable of measuring and recording relative humidity and temperature of the room.

Figure 5.17 : RHT20 humidity and temperature data logger.

RHT 20 measuring instrument records up to 16,000 temperature and humidity readings with a user programmable sample rate [71]. Specifications of RHT20 are given in the Table 5.11 as it follows.

Table 5.11 : Specifications of the measuring instrument [71].

Specifications	Range	Resolution	Accuracy
Temperature	-40 to 70 °C	0,1°C	±1,0 (-10 to 40°C) ±2,0 (all other ranges)
Humidity	0 to 100% RH	0,1% RH	±3%RH (40 to 60%) ±3,5%RH (20 to 40% & 40 to 80%) ±5%RH (0 to 20% & 80 to 100%)
Data logging interval	1 seconds to 24 hours		

In AMFI 12, RHT20 is placed to the back side of the classroom in order to prevent air streams and user interaction. The measurement point in AMFI 12 can be observed in Figure 5.18.

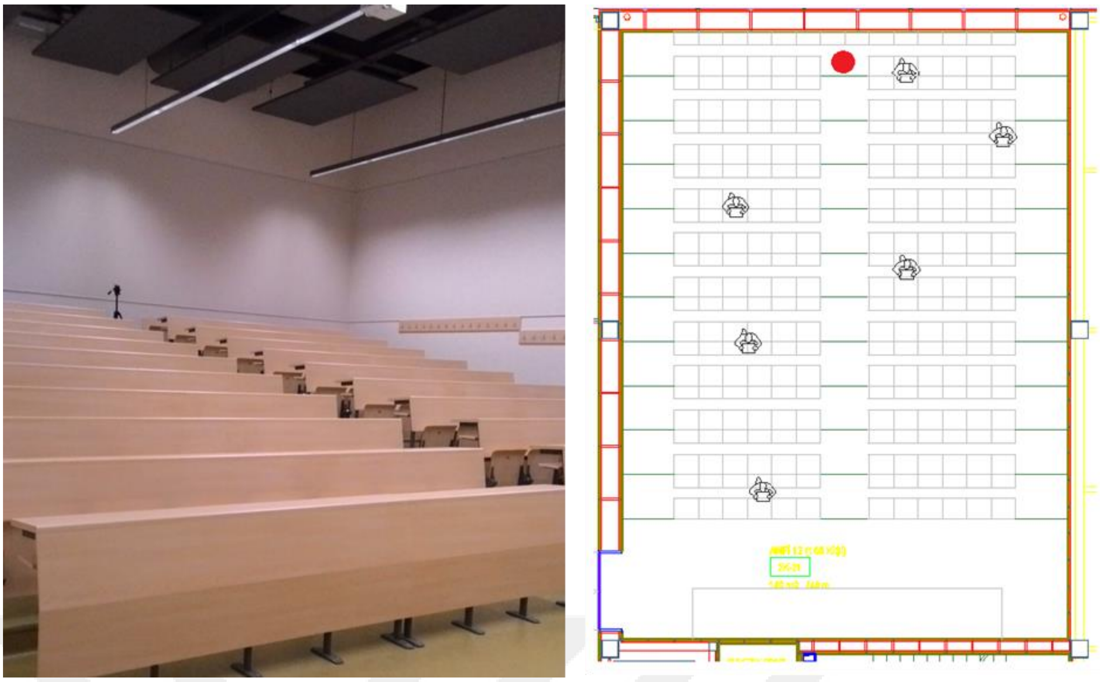


Figure 5.18 : Representation of the measurement point in AMFI 12.

One week measurement is taken place between 15th and 21st of February 2016 and covers 24 hours in a day. Since MED has a multifunctional use and hosts different lectures; its occupancy rates are also changeable. During measurement occupancy periods and rates are summarized in the following Table 5.12 and Table 5.13.

Table 5.12 : Occupied periods, course start and end times during measurement.

Date	**Morning**		**Afternoon**	
	Start	End	Start	End
15th Feb - Monday	08.30	11.30	13.30	15.45
16th Feb - Tuesday	08.30	11.10	13.30	16.05
17th Feb - Wednesday	08.40	11.40	13.30	15.30
18th Feb – Thursday	10.35	13.00	13.35	16.24
19th Feb - Friday	08.45	12.08	13.47	16.00
20th Feb - Saturday	-	-	-	-
21st Feb - Sunday	-	-	-	-

Table 5.13 : Occupancy rates in AMFI 12 during measurement.

Date	Morning		Afternoon	
	Avg. Occupancy (person)	Max. Occupancy (person)	Avg. Occupancy (person)	Max. Occupancy (person)
15th Feb - Monday	60	63	58	60
16th Feb - Tuesday	30	32	56	62
17th Feb - Wednesday	106	112	37	40
18th Feb – Thursday	24	29	43	45
19th Feb - Friday	20	22	68	80
20th Feb - Saturday	0	0	0	0
21st Feb - Sunday	0	0	0	0

Temperature and relative humidity of AMFI 12 have been measured and recorded between 15th of February and 21st of February and results are given in the Figure 5.18 and Figure 5.19 on an hourly basis.

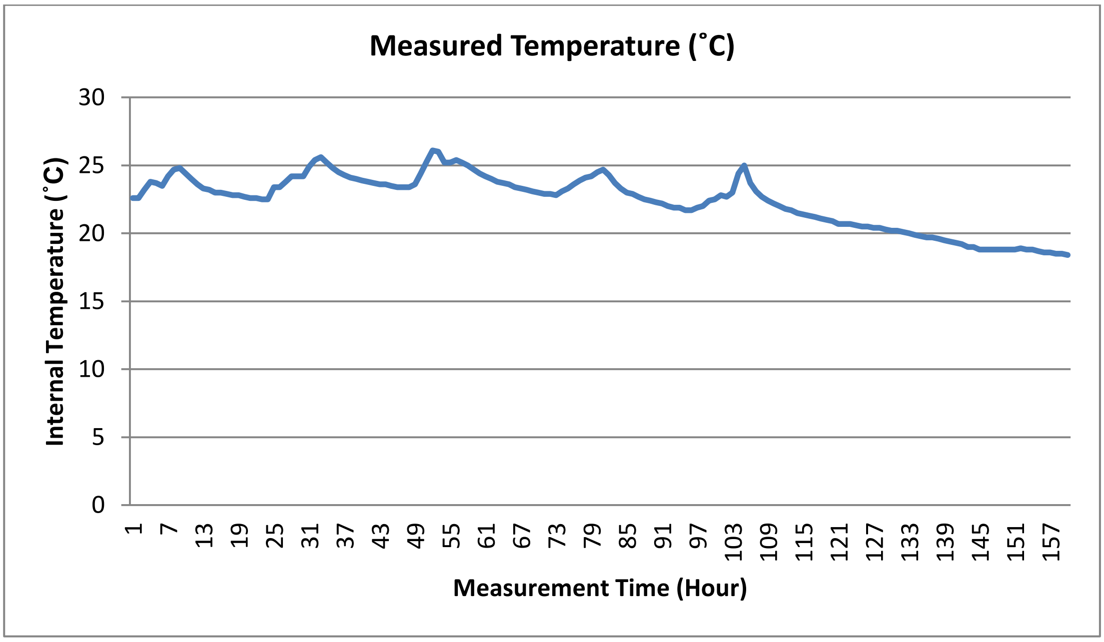


Figure 5.19 : Temperature distribution in AMFI 12 during measurement.

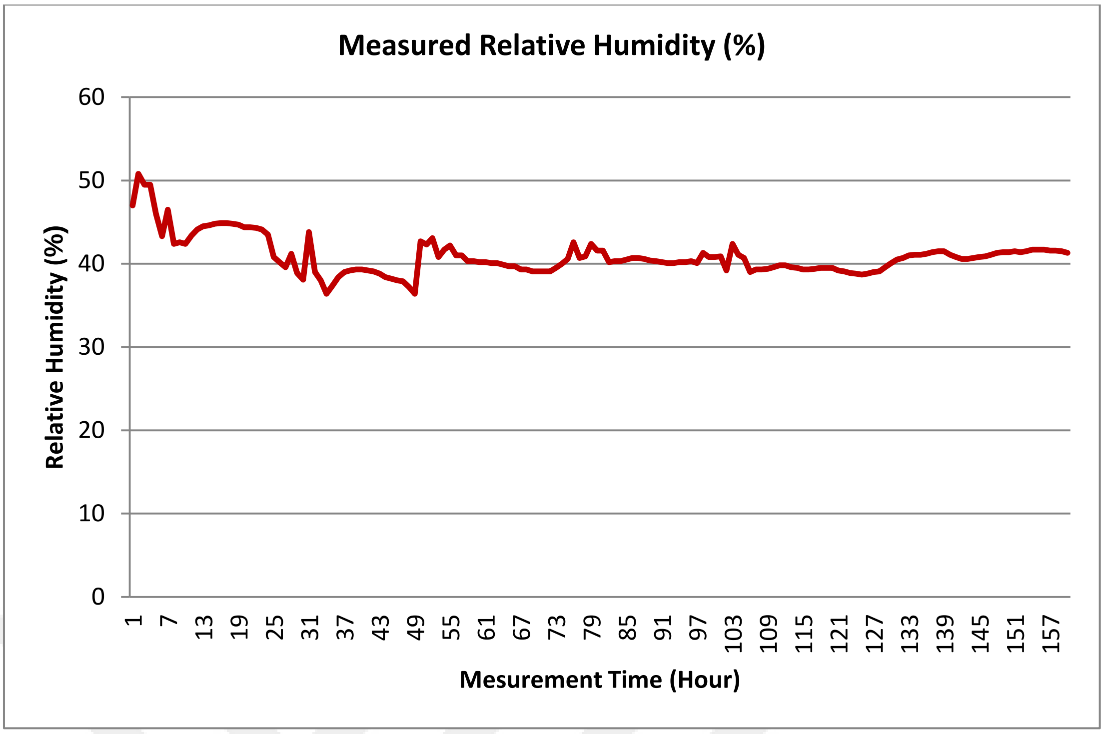


Figure 5.20 : Relative humidity distribution in AMFI 12 during measurement.

5.8 Validation of the Simulation Model

Validation of the simulation model has been fulfilled by comparing the measured values with the values taken from DesignBuilder. In the previous chapter, measured mean air temperature and relative humidity values were already presented. In this point, generated energy model of the building was simulated and accordingly mean air temperature and relative humidity distribution are found. Comparison of simulated and measured temperature distribution are given in Figure 5.21.

As it is given in Figure 5.21, difference between simulation and measured temperature hardly exceeds 1°C which is also equal to the accuracy rate of the measuring equipment. In total 160 measurement hours, 31 hours exceeds the value of 1°C. Errors, especially during Wednesday and Friday, can be explained by the occupancy rates. Since classroom had more than 100 occupants in Wednesday; measured temperature reaches temperatures higher than 25°C. Conversely, in Friday morning only 22 students were present, which takes the measured temperature down to lower degrees compare to the simulation case. In addition, it should be noted that the temperature is measured in the backstage of the room (Figure 5.18) in order not to obstruct the classroom ways, which might also led the measured temperatures to reach higher degrees in average. Yet, still overall error percentage is 3% and stays in the acceptable interim.

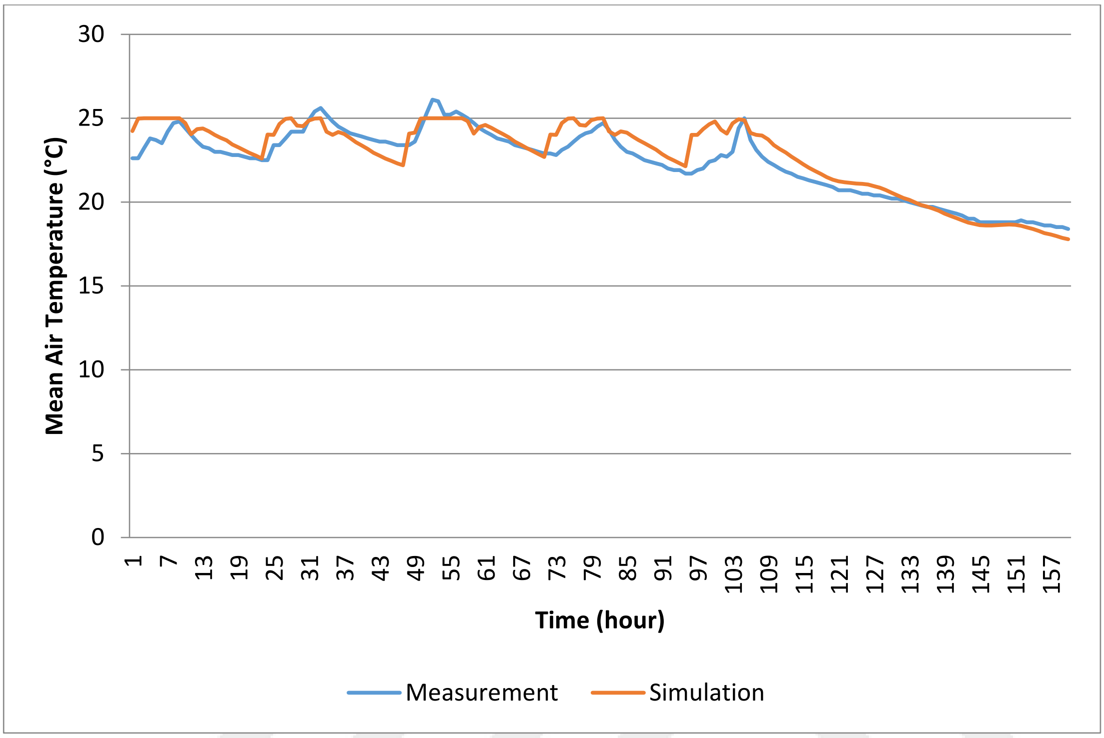


Figure 5.21 : Comparison of simulated and measured temperature distribution.

As well as temperature, measured relative humidity was also compared with the simulation case. Comparison of simulated and measured relative humidity can be depicted in Figure 5.22.

As seen in Figure 5.22 relative humidity does not change considerably, yet, it is only affected by the occupancy. The classroom was closed during the weekend and also regarding HVAC system did not operate. Consequently, RH stays nearly the same at the weekend. As a comparison between simulation and the measurement, the overall measurement error is %3 whereas the accuracy of the measuring equipment was %3,5.

In addition to mean air temperature and relative humidity comparisons, also installed capacities are compared with the simulation sizing. While in the building 1000 kW natural gas boilers are present, design heating capacity found in simulation is 935 kW which is close to the real case.

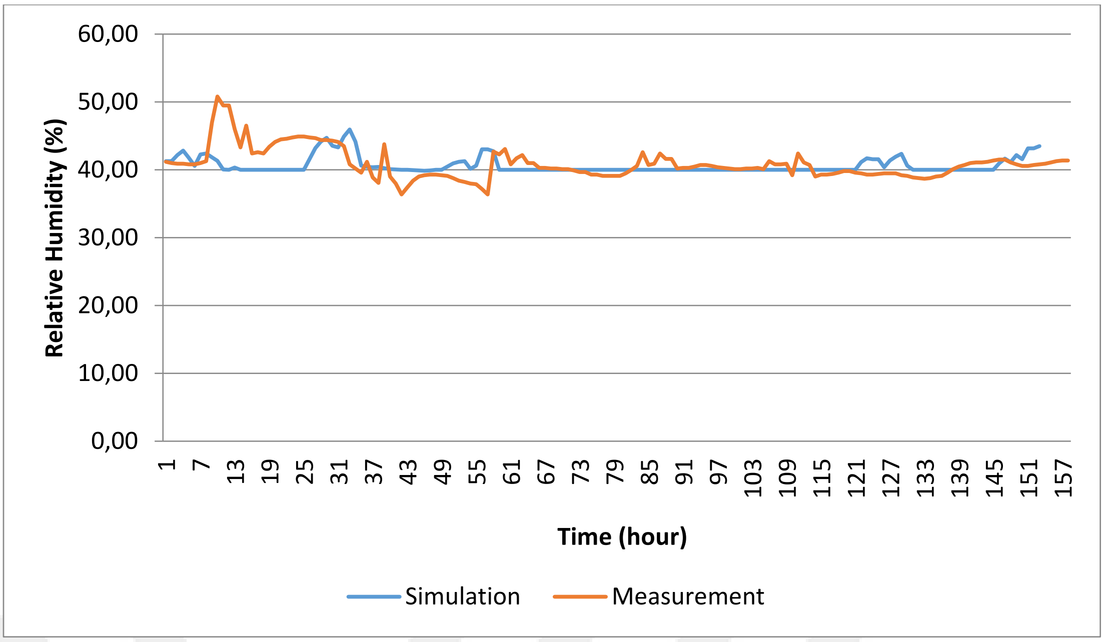


Figure 5.22 : Comparison of simulated and measured relative humidity distribution.

6. ANALYSIS

After validation of the simulation model, detailed analysis has been carried out. As a first step, analysis of the existing building has been completed taking the actual conditions on the basis. In the next section, results of analysis of existing building will be evaluated.

6.1 Existing Building

Existing building definitions and specifications were already introduced in Chapter 5. According to thermo-physical conditions of the materials and pre-defined design conditions, existing building is simulated in DesignBuilder. Results will be given into three titles;

- Loads and energy consumption of the building
- Thermal comfort analysis of the building
- Productivity analysis of the building

Hereafter, these two titles will be evaluated for existing status.

6.1.1 Load and energy consumption of the existing building

Energy simulation of MED is assessed as baseline model for the existing case. Monthly internal gains of the building is evaluated in terms of different application areas; general lighting, solar gains from interior and exterior windows, zone sensible cooling, computer and equipment, total latent loads, occupancy and zone sensible cooling.

All results are evaluated for each month in simulation tool and given in Figure 6.1 and Table 6.1 as it follows.

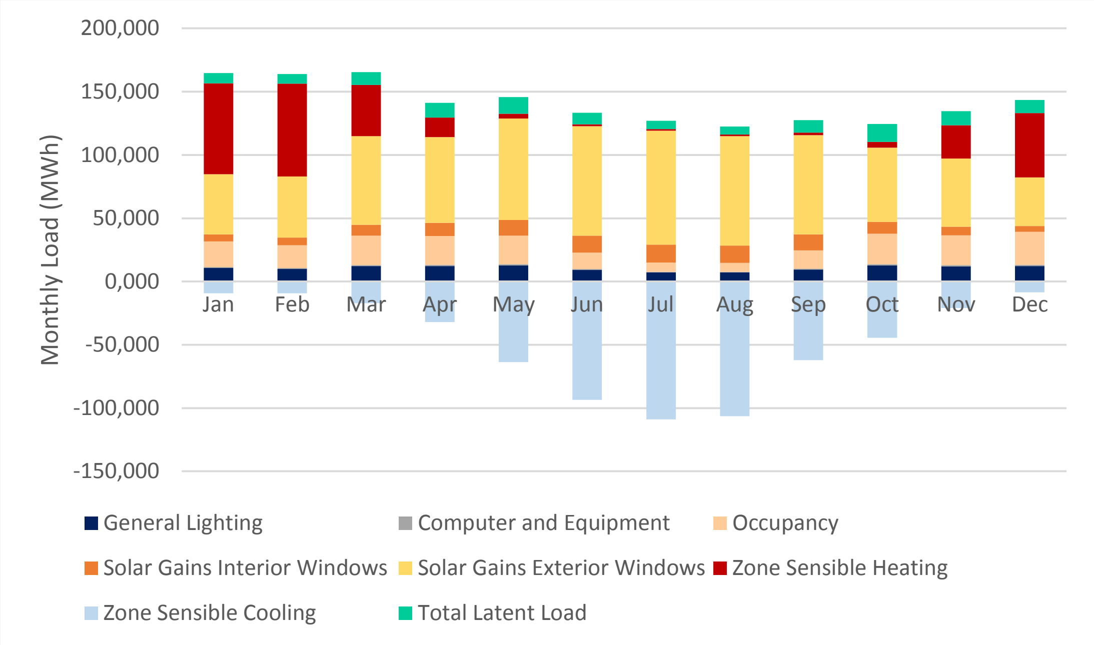


Figure 6.1 : Montly internal gains of the existing building.

Corresponding to Figure 6.1, contribution of each component is given in detail in Table 6.1.

Table 6.1 : Detailed monthly loads of the existing building.

	General Lighting (MWh)	Computer and Equipment (MWh)	Occupancy (MWh)	Solar Gains Interior Windows (MWh)	Solar Gains Exterior Windows (MWh)	Zone Sensible Heating (MWh)	Zone Sensible Cooling (MWh)	Total Latent Load (MWh)
Jan	10,697	0,796	20,100	5,751	47,435	71,76	-9,21	8,26
Feb	9,951	0,715	17,990	5,968	48,511	73,20	-9,26	7,57
Mar	12,124	0,915	23,090	8,777	70,083	40,49	-17,15	10,12
Apr	12,255	0,950	22,860	10,359	67,619	15,61	-32,04	11,46
May	12,760	0,990	22,520	12,256	80,203	3,78	-63,68	13,33
Jun	9,310	0,630	12,920	13,409	86,521	1,45	-93,46	9,17
Jul	7,280	0,420	7,360	14,123	90,072	1,32	-108,9	6,38
Aug	7,240	0,410	7,130	13,679	86,507	1,25	-106,5	6,26
Sep	9,530	0,690	14,430	12,660	78,265	2,13	-62,05	9,92
Oct	12,768	1,040	23,840	9,414	58,832	4,54	-44,53	13,96
Nov	12,000	0,960	23,610	6,690	53,965	26,24	-20,89	11,16
Dec	12,330	1,000	25,840	4,700	38,494	50,64	-8,49	10,58

As it can be seen from the Figure 6.1 and Table 6.1, solar gains from the exterior windows cover the largest portion of the monthly loads of the building. Solar gains from the exterior windows nearly cover the 62% of the annual internal gains of MED. Gains from occupancy follows it with 17% and gains from lighting covers 10%.

Building is south oriented and has more than 30% window to wall ratio which makes the solar gains from exterior windows one of the highest indicators in monthly load analysis. Also note that, occupancy rates change from month to month as the building's classrooms are only occupied during the teaching period; so during July and August season it reaches the lowest values. In the university, building occupancies are adjusted according to academic calendar.

Besides internal gains, internal losses of the existing building are investigated. Internal losses consist of heat loss from glazing, walls, ceilings, floors, partitions, roofs and via infiltration and ventilation. Heat losses are given in Figure 6.2 as it follows.

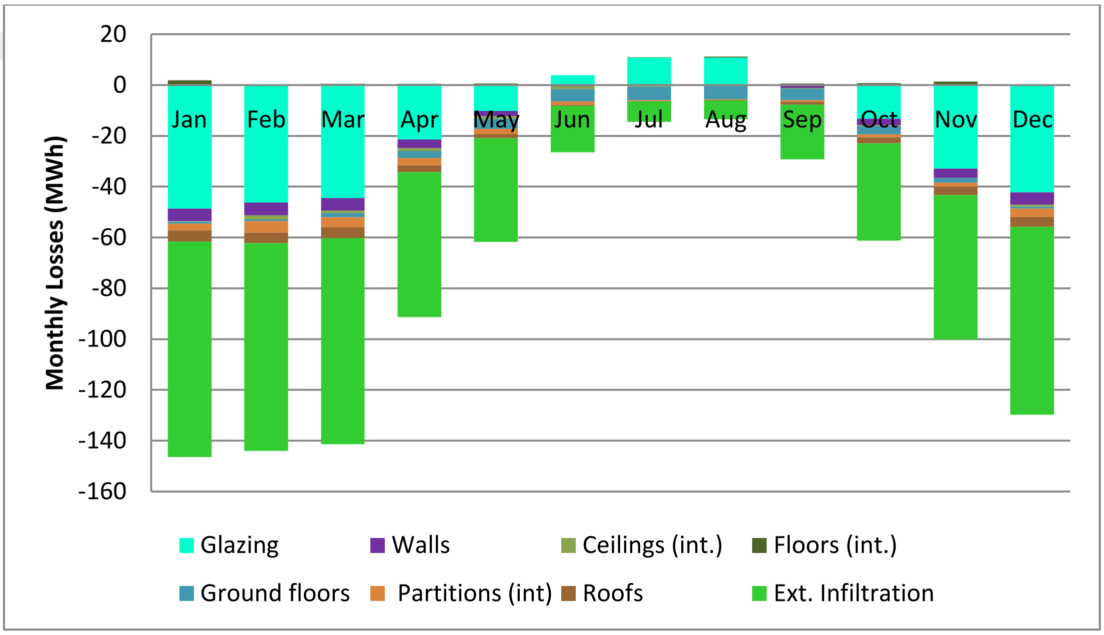


Figure 6.2 : Montly internal heat losses of the existing building.

Given in Figure 6.2, most of the heat losses occur due to losses from external infiltration which are composed of ventilation and infiltration losses through walls or windows. In addition, losses from glazing cover a large portion.

In addition to monthly loads of the building, building's energy consumption is evaluated. Existing building's annual energy consumption can be seen in Figure 6.3 and Table 6.2 as it follows.

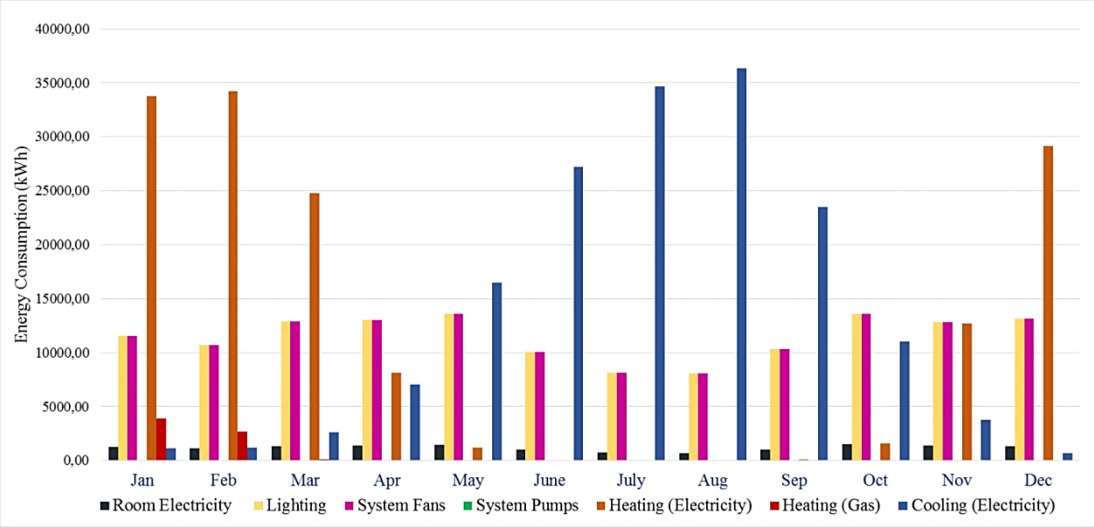


Figure 6.3 : Annual energy consumption figure of the existing building.

Corresponding to Figure 6.2, contribution of each component to the annual energy consumption of the building is represented comprehensively in Table 6.2.

Table 6.2 : Annual energy consumption values of the existing building.

	Room Electricity (kWh)	Lighting (kWh)	System Fans (kWh)	System Pumps (kWh)	Heating (Electricity) (kWh)	Heating (Gas) (kWh)	Cooling (Electricity) (kWh)
Jan	1246,28	11524,81	11524,00	0,52	33763,00	3931,00	1131,00
Feb	1137,65	10687,36	10687,00	0,32	34251,00	2693,00	1172,40
Mar	1319,05	12921,09	12921,00	0,02	24750,00	133,00	2637,00
Apr	1378,16	13052,40	13052,00	0,00	8150,00	15,00	7052,00
May	1459,40	13595,97	13595,00	0,00	1216,00	0,00	16486,00
June	977,60	10076,16	10076,00	0,00	28,00	0,00	27235,00
July	720,06	8111,92	8111,00	0,00	12,82	0,00	34673,00
Aug	708,82	8064,25	8064,00	0,00	12,17	0,00	36325,00
Sep	998,81	10294,73	10294,00	0,00	95,60	0,00	23473,00
Oct	1509,10	13595,97	13595,00	0,00	1599,60	0,00	11002,00
Nov	1403,12	12800,80	12800,00	0,00	12731,00	22,51	3743,00
Dec	1356,37	13125,82	13125,00	0,01	29160,00	75,92	700,07

As it depicted in the Table 6.2 cooling consumption of the building remains one of largest parts of the total consumption of MED. There might be several reasons for this occasion. First, building has high window-to-wall ratios and accordingly higher solar gains, as it is explained before. Secondly, in the building ventilation system does not have any cooling serpentine. Yet, even though there is no cooling system present, fresh air is supplied to the building in the cooling season. In that way, cooling consumption increases drastically in summer months. In the building, cooling is only provided by VRF systems and it is seen on the electricity consumption. However, for heating purposes, along with the VRF air-conditioning system, boiler is also utilized in order to heat ventilation air as a Air Handling Unit (AHU). Consequently, building's heating consumption is given under two titles in Table 6.2, heating gas and heating electricity which corresponds to boiler and VRF system consumption respectively.

6.1.2 Thermal comfort analysis of the existing building

As well as energy results, thermal comfort analysis study has been conducted which is one of the main purposes of the thesis. Overall building's monthly PMV values are calculated by using Fanger Method and results can be seen in Figure 6.4.

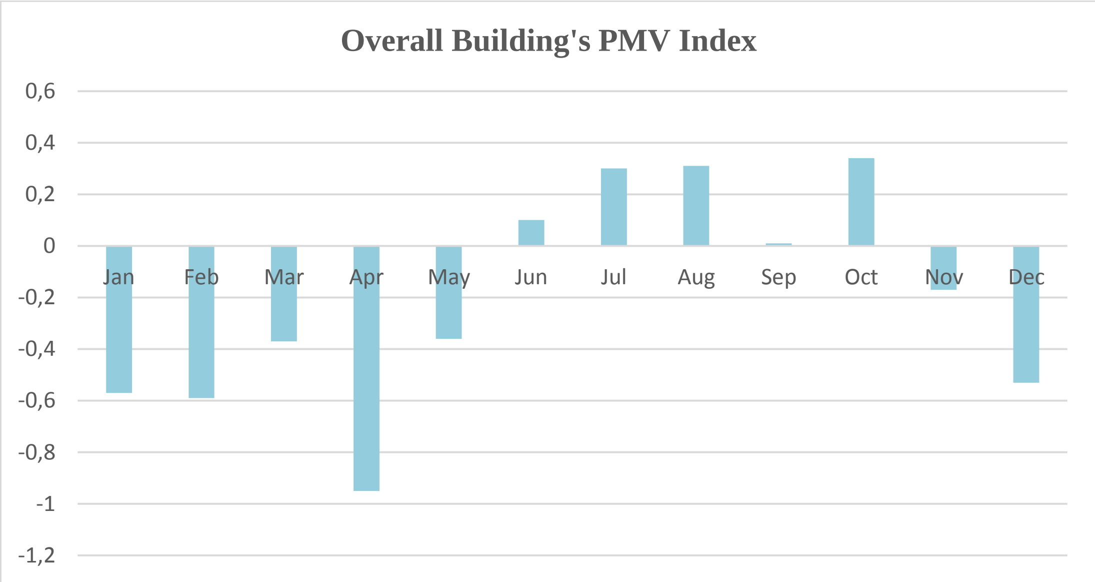


Figure 6.4 : Fanger PMV values of the existing building.

As represented in Figure 6.4, PMV index has negative values between November and January; however, during June and October it has positive value which means that the building is slightly hot. According to simulation results, the overall building reaches overall peak PMV value in month April with -0,9. It can be explained by the clothing value assumption of the simulation tool. In DesignBuilder, "clo" values are taken as fixed values in according to ASHRAE 55 and ISO 7730 standards; 0,5 for cooling season and 1 for heating season. Accordingly, it causes sharp reduces and increases in PMV index in the month April and October as it can be seen from the Figure 6.4. This issue is going to be investigated further in the following part of the thesis.

MED mainly consists of two different types of zones apart from common ones; offices and classrooms. Classrooms also divides into two, south-oriented seminar rooms and north-oriented amphitheatre type lecture halls. Besides building's overall PMV, PMV value of each zone should be investigated in order to get precise information regarding the building. Consequently, building's monthly PMV values are evaluated for south-oriented offices and seminar rooms along with north-oriented lecture halls (amphitheatre). Monthly average PMV indexes of referred zones are given in Figure 6.5 as it follows.

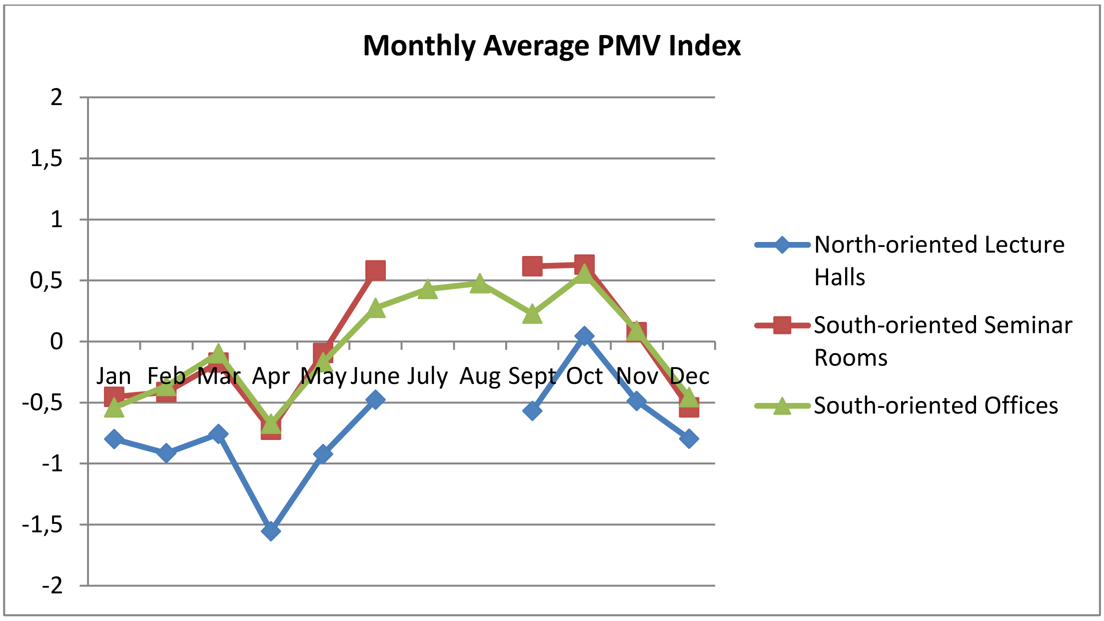


Figure 6.5 : Monthly average PMV index of different zones in existing case.

Given in Figure 6.5, south-oriented zone groups have higher PMV values compare to the nort-oriented lecture halls. Yet, mainly the trend in each zone group is similar. It should be noted that PMV values are calculated only during occupied times, consequently July and August months are not evaluated in classrooms.

In addition to PMV values, mean air temperature, radiant temperatures, outside dry-bulb temperature and the relative humidity is significant when addressing thermal comfort analysis of the building. Mentioned values can be seen in the Figure 6.6.

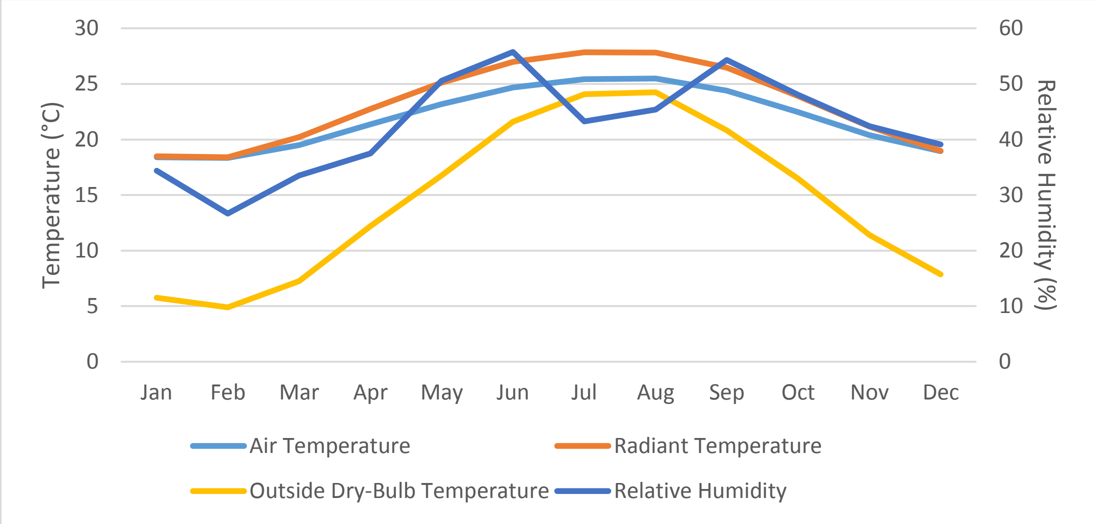


Figure 6.6 : Monthly thermal comfort indicators of the overall building.

Detailed values of each component for overall building in Figure 6.6 are represented in the Table 6.3 as it follows.

Table 6.3 : Monthly thermal comfort values of the overall building.

	Air Temperature	Radiant Temperature	Operative Temperature	Outside Dry-Bulb Temperature	Relative Humidity	Fanger PMV
Jan	18,41	18,5	18,45	5,76	34,36	-0,57
Feb	18,36	18,41	18,39	4,88	26,66	-0,59
Mar	19,5	20,22	19,86	7,25	33,53	-0,37
Apr	21,37	22,74	22,06	12,21	37,51	-0,95
May	23,2	25,13	24,17	16,77	50,55	-0,36
Jun	24,68	26,98	25,83	21,61	55,77	0,1
Jul	25,43	27,84	26,63	24,09	43,28	0,3
Aug	25,48	27,82	26,65	24,24	45,39	0,31
Sep	24,41	26,48	25,44	20,82	54,3	0,01
Oct	22,48	23,92	23,2	16,46	48,03	0,34
Nov	20,38	21,17	20,78	11,39	42,42	-0,17
Dec	18,96	18,98	18,97	7,87	39,09	-0,53

6.1.3 Productivity analysis of the existing building

In the literature section of the thesis, it was already presented that there is a strong link between productivity and thermal environment. Several researches emphasise the impact of indoor environment on the occupants productivity. In order to measure mentioned impact, Kosonen and Tan has suggested a theoretical approach which basically investigates productiviy loss in two groups; thinking and typing depending on the present PMV conditions of the environment [26]. Kosonen and Tan represent the mathematical models to calculate typing and thinking tasks and equations are depicted in equation 6.1 and 6.2 as it follows.

$$PL_{ty} = -60{,}543\, PMV^6 + 198{,}41\, PMV^5 - 183{,}75\, PMV^4 - 8{,}1178\, PMV^3 + 50{,}24\, PMV^2 + 32{,}123\, PMV + 4{,}8988 \qquad (6.1)$$

$$PL_{tp} = 1{,}5928\, PMV^5 - 1{,}5526\, PMV^4 - 10{,}401\, PMV^3 + 19{,}226\, PMV^2 + 13{,}389\, PMV + 1{,}8763 \qquad (6.2)$$

where, PL_{ty} = Typing Productivity Loss (%)

PL_{th} = Thinking Productivity Loss (%)

PMV = Predicted Mean Vote

Although it is clear that there is a bond between productivity and thermal environment, it should be also noted that optimal performance, or namely productivity, does not necessarily occur under on neutral thermal conditions. Accordingly, in equations 6.1 and 6.2, thermal neutrality value "0" of PMV does not correspond to optimum value of productivity. Instead, PMV value of "-0,211114" is evaluated to be the best value for minimizing both typing and thinking productvity loss.

Existing conditions' monthly productivity loss percentages are calculated using the above-mentioned approach for each month. Existing building's PMV values are already presented in the previous section, thus, according to the values overall productivity losses for thinking and typing activities are found and given in Figure 6.3 as it follows.

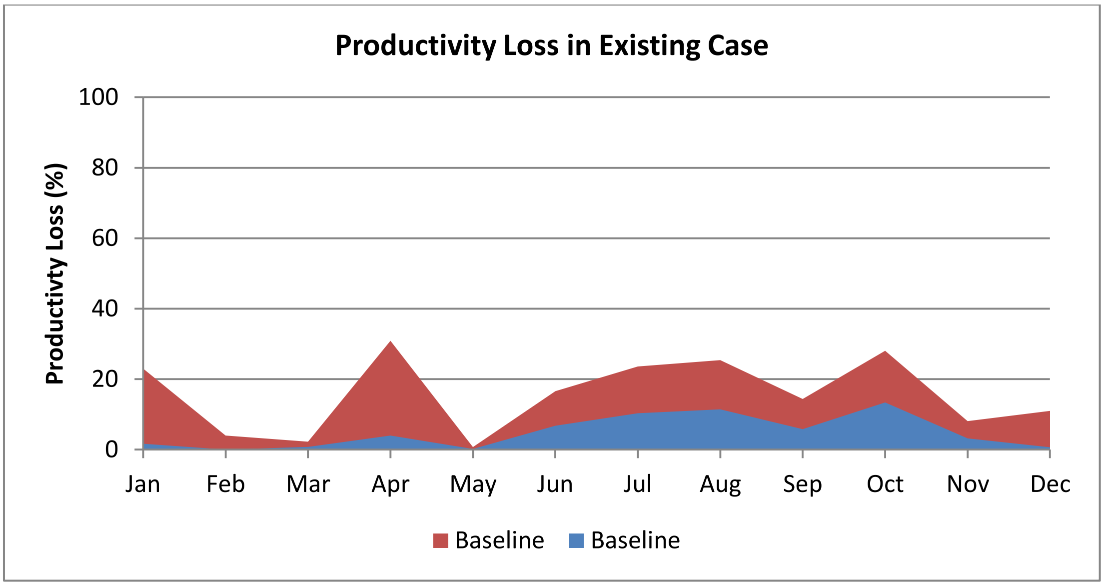


Figure 6.7 : Productivity loss for overall existing building.

As can be seen from the Figure 6.7, thingking productivity loss in overall building is relatively low, yet, for typing activities it reaches up to 30%.

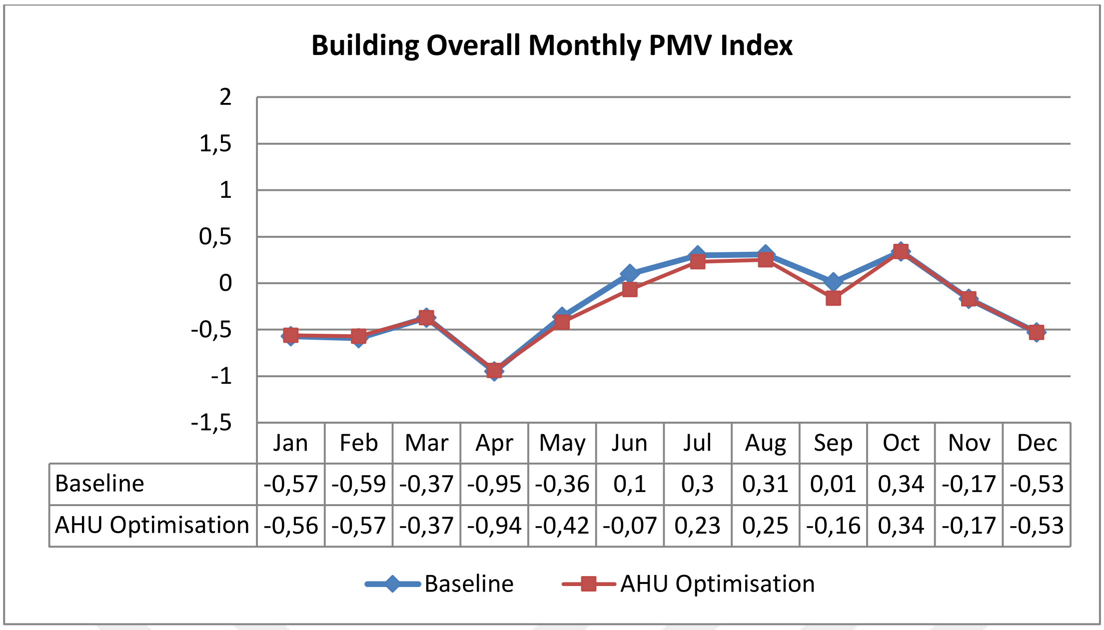
6.2 Evaluated Strategies

It was found out that in the existing building thermal comfort cannot be obtained with the present conditions during some periods. In order to investigate and optimize the thermal comfort, a set of strategies have been carried out including both design and operational strategies and their effect on thermal comfort, energy consumption and productivity are analyzed. Proposed strategies will be investigated in offices placed in the south-west direction and amphitheater classrooms placed in the north-east direction of the building.

Hereafter proposed strategies will be defined and evaluated.

6.2.1 Optimisation of AHU

As a first strategy, optimization of Air Handling Unit (AHU) is evaluated. As it is explained in the Chapter 5 in detail, building has only heating coils in AHU system. Additionally, main halls and corridors, which cover more than 3400 m^2, were not provided fresh air continuously even in occupied hours since the ventilation and AHU is disabled after 17.00. Consequently in optimisation of AHU strategy, cooling coils are also added to the AHU system. PMV comparison for whole building on an annual basis can be seen in Figure 6.7.



	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Baseline	-0,57	-0,59	-0,37	-0,95	-0,36	0,1	0,3	0,31	0,01	0,34	-0,17	-0,53
AHU Optimisation	-0,56	-0,57	-0,37	-0,94	-0,42	-0,07	0,23	0,25	-0,16	0,34	-0,17	-0,53

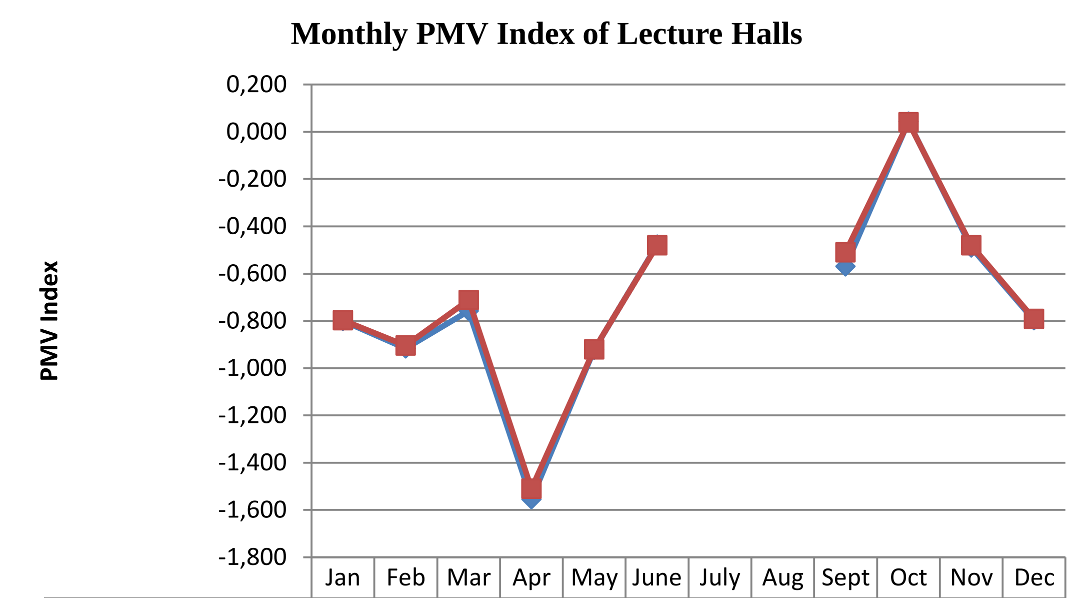
Figure 6.8 : PMV comparison of baseline and AHU optimsation strategy.

As represented in Figure 6.7, AHU optimisation strategy has impact mainly on cooling degree days. Within the introduction of cooling coils to the fresh air side, summer PMV values have slightly reduced and have remained between the recommended values of ±0.3. Yet still, there are room for improvement especially in winter months.

Buildings overall PMV value might be misleading in some cases if it is thought that each zone is assessed one by one in thermal comfort for a time interval. Therefore along with building's overall monthly PMV values, offices and amphitheatres will be evaluated and compared with the baseline case. As it is explained in previous chapters, offices and lecture halls constitute a large part of MED. Subsequently, north-oriented lecture halls and south oriented offices and seminar rooms will be investigated.

Comparison of baseline and AHU optimisation for North-oriented Lecture Halls, South-oriented offices and seminar rooms are given in Figure 6.8.

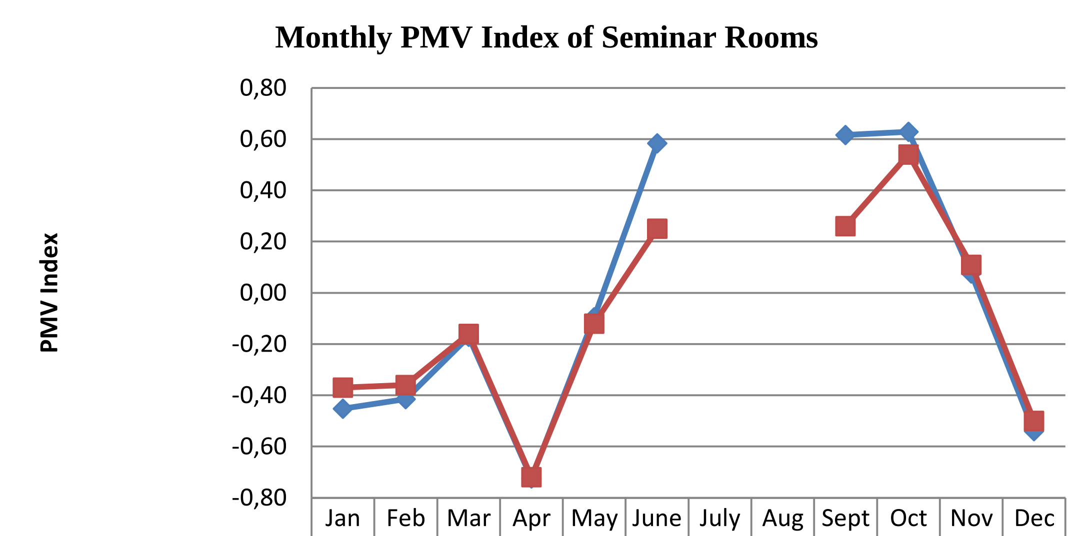
As given in the Figure 6.8, AHU optimisation strategy has barely effect on the north-oriented lecture halls since the lecture halls are mostly cold during each period of year.



	Jan	Feb	Mar	Apr	May	June	July	Aug	Sept	Oct	Nov	Dec
Baseline	-0,80	-0,91	-0,75	-1,55	-0,92	-0,47			-0,56	0,044	-0,48	-0,79
AHU Optimisation	-0,79	-0,90	-0,71	-1,51	-0,92	-0,48			-0,51	0,041	-0,48	-0,79

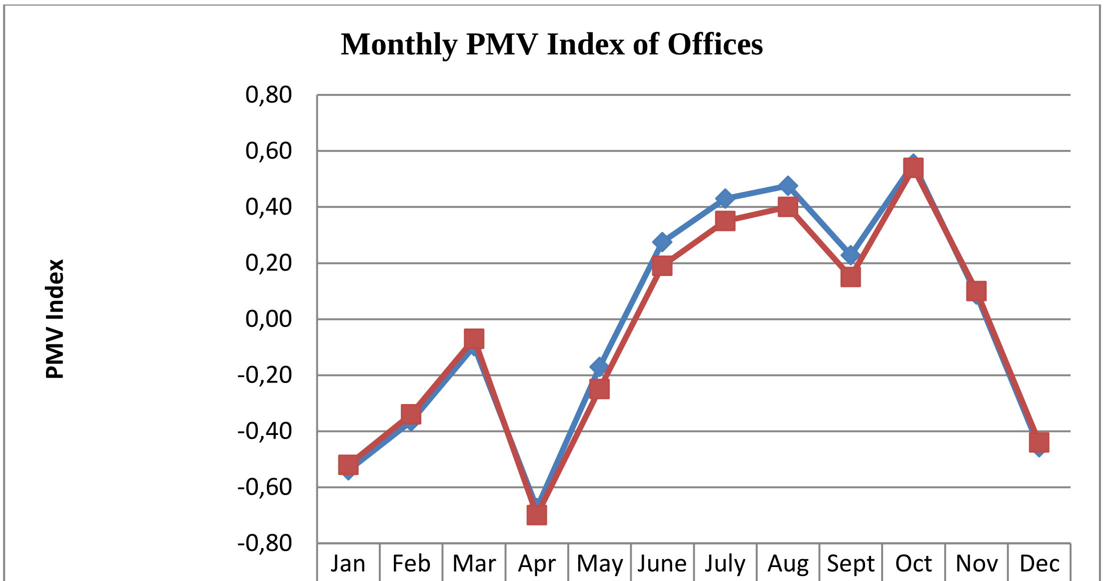
Figure 6.9 : PMV comparison of baseline and AHU optimsation strategy in lecture halls.

Besides lecture halls, effect of the AHU optimisation strategy is investigated in south-oriented zone groups; seminar rooms and offices. Comparison of the baseline and AHU optimisation in seminar room groups and offices can be seen in Figure 6.9 and 6.10 respectively.



	Jan	Feb	Mar	Apr	May	June	July	Aug	Sept	Oct	Nov	Dec
Baseline	-0,45	-0,41	-0,17	-0,73	-0,10	0,58			0,62	0,63	0,08	-0,54
AHU Optimisation	-0,37	-0,36	-0,16	-0,72	-0,12	0,25			0,26	0,54	0,11	-0,50

Figure 6.10 : PMV comparison of baseline and AHU optimsation strategy in seminar rooms.



	Jan	Feb	Mar	Apr	May	June	July	Aug	Sept	Oct	Nov	Dec
Baseline	-0,54	-0,37	-0,10	-0,67	-0,17	0,27	0,43	0,48	0,23	0,56	0,09	-0,46
AHU Optimisation	-0,52	-0,34	-0,07	-0,70	-0,25	0,19	0,35	0,40	0,15	0,54	0,10	-0,44

Figure 6.11 : PMV comparison of baseline and AHU optimsation strategy in offices.

As represented in Figure 6.9, AHU optimisation has a possitive effect on seminar rooms especially in June and September months. However, since the classrooms are not occupied during the summer period, PMV value is not calculated and accordingly no effect can be observed in July and August months in seminar room zone groups. On the other hand, offices are occupied even during the summer holiday, therefore impact of AHU optimisation during summer time can be depicted from Figure 6.10.

As it is represented in the figure, AHU optimsation has significant effect on the summer months. With the introduction of cooling coil to the ventilation system, summer PMV's of the south-oriented zone groups; especially in office zone groups which are occupied during the summer time. In offices PMV value approaches to the desired value in the summer months significantly. Though, it has hardly any effect on the amphitheater lecture halls which are located in the north perimeter of the building.

6.2.2 Pre-heating

In the baseline management of MED, heating and cooling systems start at 07.00 for classrooms. Consequently, it leads to accumulation of heat in the unoccupied hours. Especially during winter months, air temperature of lecture halls at midnight reduces down to 5-10°Cs and causes thermal discomfort in the morning hours. In order to prevent this situation, pre-heating strategy is evaluated. 1 hour earlier pre-heating strategy is considered Results of the mentioned strategy is given in Figure 6.11 as it follows.

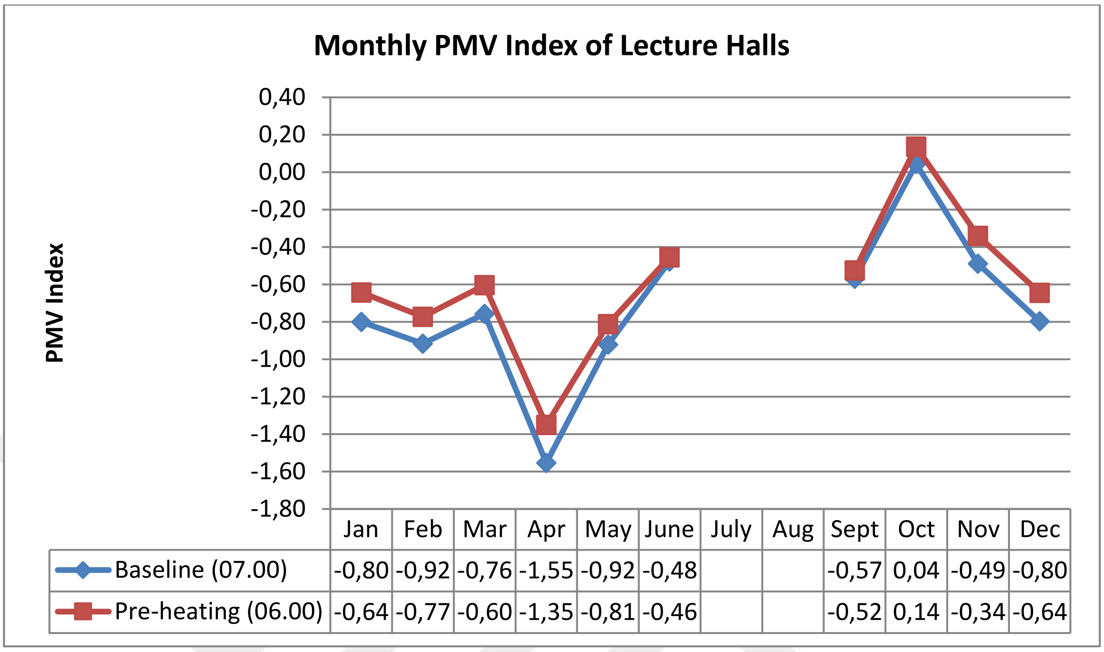


Figure 6.12 : PMV comparison of baseline and pre-heating strategy in lecture halls.

As given in Figure 6.11, pre-heating strategy has significant effect on the lecture halls. Yet, apart from thermal comfort side; its contribution to energy efficiency should be also addressed. Compared to the baseline, annual energy consumption of the pre-heating strategy is given in Figure 6.12.

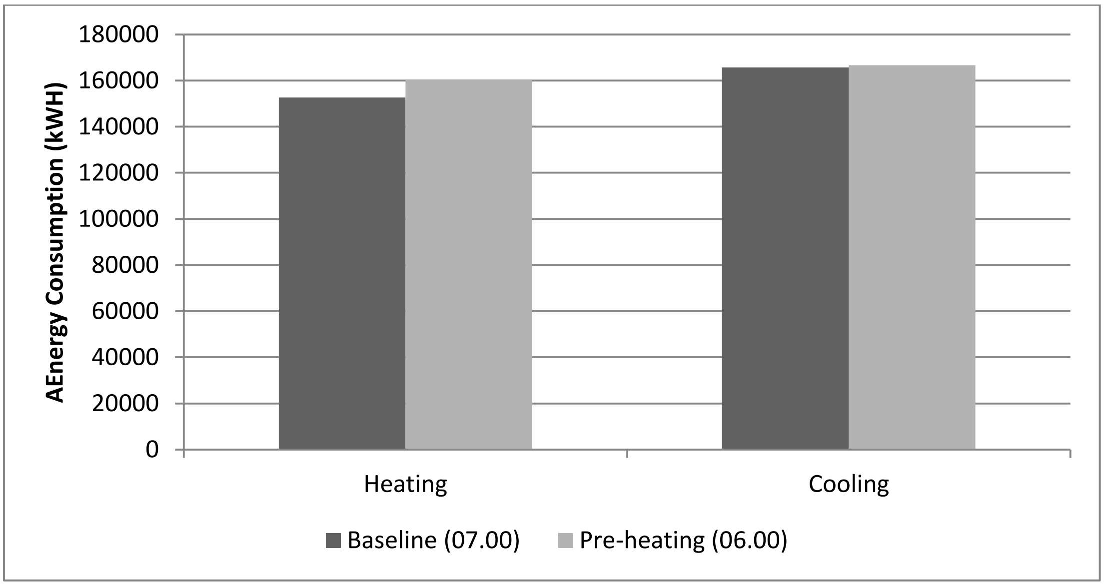
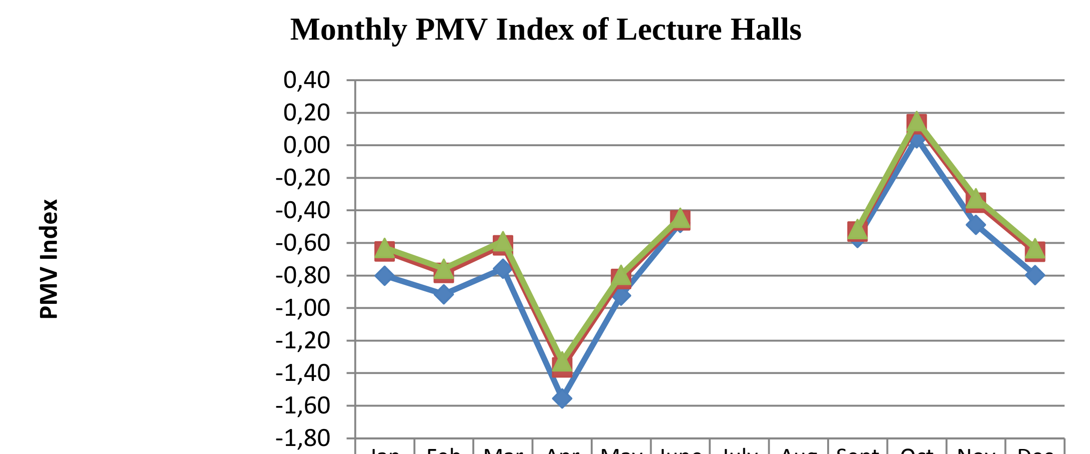


Figure 6.13 : Energy consumption comparison of baseline and pre-heating strategy.

In pre-heating strategy, annual heating consumption of the building is increased 8,5%, whereas, in annual cooling consumption it corresponds to 3,9%.

6.2.3 Set-point variation in lecture halls

In the baseline model, set-points are regarded as 22°C for heating season and 24°C for cooling season according to TS EN 15251 Standards [63]. Although building's set-points were determined based on the standards, preferring the same set-point for each zone might cause thermal discomfort for occupants. It is obvious that, since northern oriented lecture halls does not recieve solar radiation and have large amount of volumes, thermal sensation is cold. Accordingly, two different set-point variation strategy is evaluated in the lecture halls; 23°C and 24°C for heating. Comparison of these strategies in lecture halls is depicted in Figure 6.13 as it follows.



	Jan	Feb	Mar	Apr	May	June	July	Aug	Sept	Oct	Nov	Dec
22°C Set Point (Baseline)	-0,80	-0,92	-0,76	-1,55	-0,92	-0,48			-0,57	0,04	-0,49	-0,80
23°C Set Point	-0,65	-0,78	-0,61	-1,36	-0,82	-0,46			-0,53	0,13	-0,35	-0,65
24°C Set Point	-0,63	-0,76	-0,59	-1,33	-0,79	-0,44			-0,51	0,15	-0,32	-0,63

Figure 6.14 : PMV comparison of baseline and set-point variation in lecture halls.

In Figure 6.13, comparison of baseline PMV index of lecture halls between 23°C and 24°C set-point variation strategies are represented. Results revealed that, in winter months, increasing set-point is an effective solution to enhance thermal comfort of occupants in lecture halls. By applying 23°C and 24°C set-point variations, each month had an positive affect of thermal sensation.

Apart from thermal comfort aspect, energy consumption of the each strategy and baseline is also evaluated. Annual energy consumption of each strategy is given in Figure 6.14.

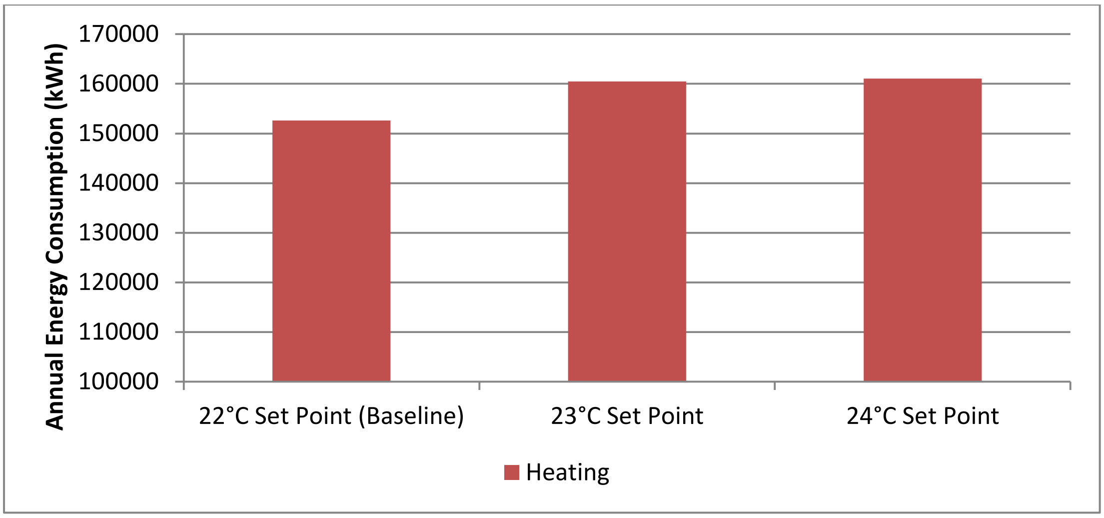


Figure 6.15 : Energy consumption comparison of baseline and set-point variations.

According to building's annual energy consumption analysis, 23°C set-point variation have resulted with the increase of 5,1% increase in annual heating consumption; yet, for 24°C consumption has increased by 5,6%. It should be mentioned that set-point raise is only applied to the 12 lecture halls in the building.

6.2.4 Optimisation of the shading system

In the south-western façade of the building, shading element which consists of movable perforated metal panel is constructed as it is seen in Figure 6.15.

Figure 6.16 : Moveable shading element of the building.

This moveable shading element allows occupants to open and close the shading element block according to their requirements. In this strategy, optimisation of the shading system is assessed with the comparison of different time periods to keep shading element on or off.

In the analysis, “off position” of the shading element is investigated in 5 different time-periods; 30 March (Baseline), 15 April, 30 April, 15 May and 30 May. Comparison of each strategy in a south-oriented office zone and a seminar room are given in the following Figure 6.16 and Figure 6.17 respectively.

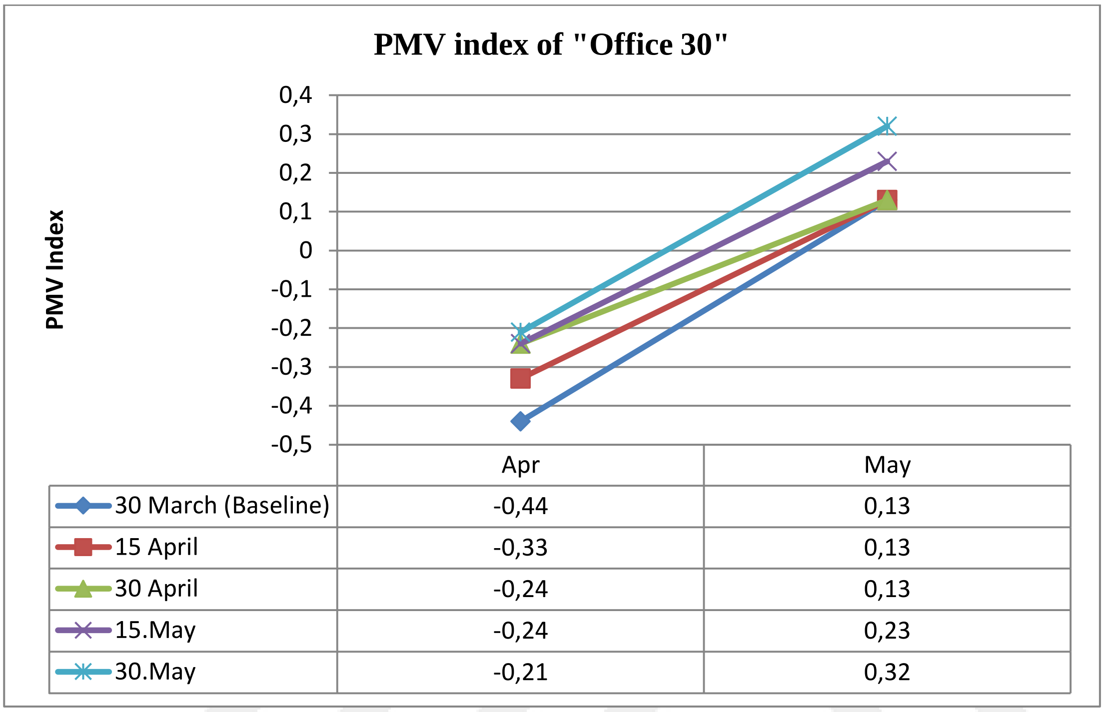


Figure 6.17 : PMV comparison of shading element optimisation in “Office 30”.

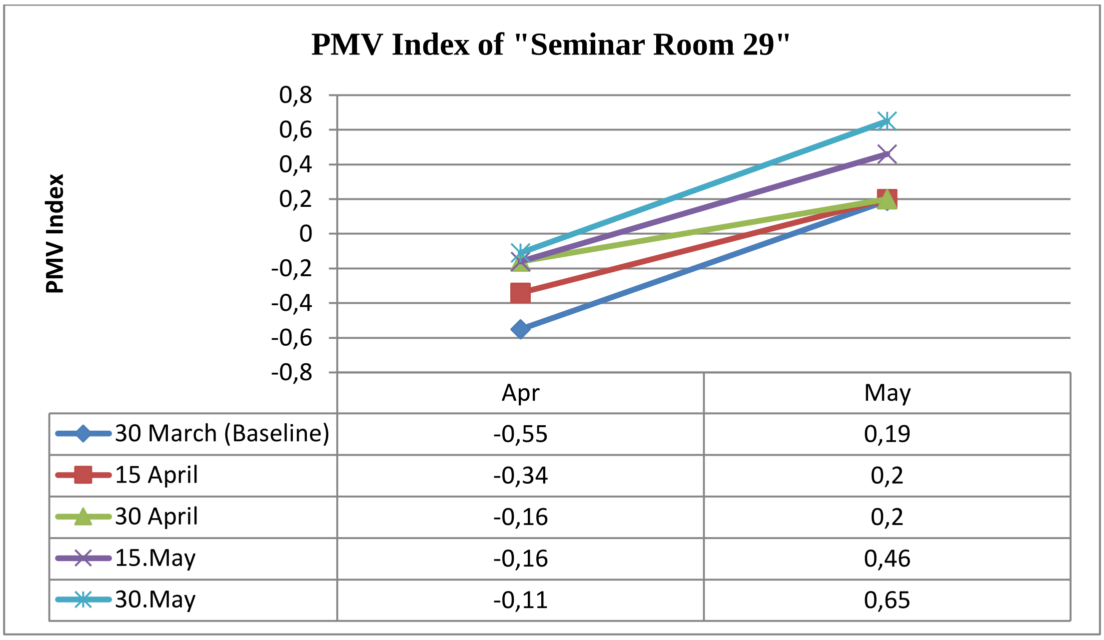


Figure 6.18 : PMV comparison of shading element optimisation in “Seminar 29”.

In the baseline case, shading element is off positioned since the end of the March. Yet, as evaluated in the baseline case, April is evaluated to be slightly cold in winters with negative PMV values. In this sense, keeping the shading element off positioned in April would be a precise solution to increase the thermal comfort in the mentioned zone groups according to Figure 6.16 and 6.17. However, since the May PMV rates are higher, it is not practical to keep the shading element off in month May. As a conclusion, 30th of April is evaluated to be the best period for shading elements to be closed in the south-oriented zone groups of the building.

6.2.5 Effect of the clothing

As investigated before, clothing insulation is another parameter affecting thermal comfort. In the baseline model, simulation programme assumes the clothing insulation values (clo) for each month as it is given in the following Table 6.4.

Table 6.4 : Clo and PMV index of baseline case.

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Clo	1	1	1	0,5	0,5	0,5	0,5	0,5	0,5	1	1	1
PMV Index	-0,57	-0,59	-0,37	-0,95	-0,36	0,10	0,30	0,31	0,01	0,34	-0,17	-0,53

As represented in Table 6.4, clo values are taken as fixed values in the simulation according to ASHRAE 55 and ISO 7730 standards; 0,5 for cooling season and 1 for heating season. However, it causes discomfort and peaks in months April and October as depicted in the Table 6.4. In order to overcome this problem and find the best solution, two different strategies are evaluated by simulating the baseline model with 0,5 and 1 clo values. Results are given in Figure 6.18 as it follows.

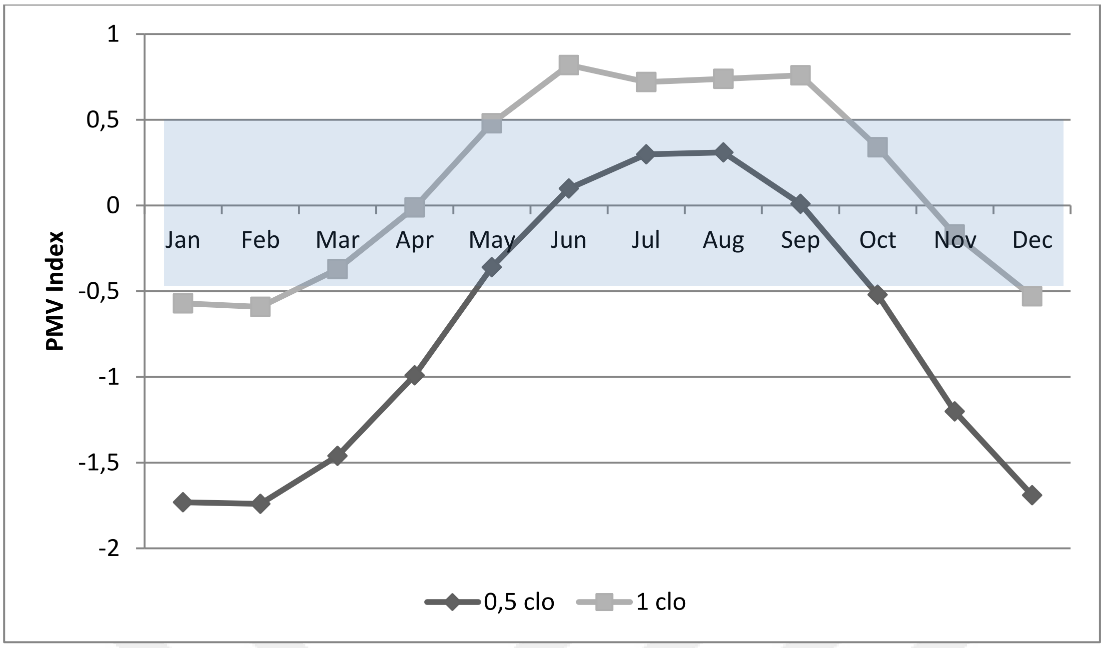


Figure 6.19 : Comparison of 0,5 and 1 clo strategy.

As depicted in Figure 6.14, to keep PMV index in desired ±0,5 interim, 0,5 clo between the months of May and October is convenient for occupants' thermal comfort, rather than April and September as it is assumed in the baseline case.

7. RESULTS AND DISCUSSION

In the previous chapter, study of the existing thermal comfort conditions is evaluated. Additionally, in order to optimize thermal comfort conditions and also overlook their effect, several design and control strategies are carried out.

Existing building has three different types of heated and cooled zone groups excluding the common areas;

- Lecture halls
- Seminar Classrooms
- Academican Offices

As above-mentioned, 12 lecture halls cover the two floor height and have the largest volume with its floor height of 8.1 meters. Additionally, they are located in the northern backside of the building with only small glazing. Consequently, in the first thermal comfort analysis, it is shown that lecture halls are the most problematic zones in the building. Monthly PMV index of the lecture halls are between 0,04 and -1,55; yet, it has the negative value for the most of the time. It indicates that thermal sensation in lecture halls is between "slightly cold and cold".

Another drawback of the lecture halls is the time span that the rooms are not occupied. Normally, classrooms are occupied between 07.30 and 17.00, accordingly mechanical systems of the refferred zones are only utilized in that time period. Namely, till 17.00 to 07.00, the zones are neihter heated nor cooled for 14 hours which causes thermal discomfort especially during morning hours.

On the other hand, seminar classrooms and academican offices are oriented in the southern part of the building and have floor-height glazing in the south-side. In that way, shading element plays a crucial role in thermal comfort during the summer months. Average PMV rate in those zone groups are much higher than the lecture halls. Yet, since offices' mechanical system is controlled by users and mentioned zones can be occupied after 17.00; time period that mechanical system is not working is relatively less than the classrooms. Subsequently, in the PMV scale, offices have the best values. PMV value of the offices differs between -0,67 and 0,47.

As above-summarized, there might be optimisation in each group of zones in order to enhance the thermal comfort for the occupants. With that purpose, five different strategy are evaluated.

- Optimisation of Air Handling Unit:

Optimisation of Air Handling Unit strategy is applied to all building. In the existing condition, building has no cooling coils present which causes undesirable heat in summer months when ventilation takes place. Normally, the building was designed to be used in only teaching periods. Yet, at the present time it has a usage in summer time especially in the office zones. Accordingly, although absence of cooling coil in AHU system has almost no effect on the unventilated classrooms, it leads temperatures rise in the office zones during summer. Therefore, with the application of AHU optimation, summer PMV's of office zones are falled back to 0,19 to 0,40 from 0,27 to 0,48.

- Pre-heating

It was already mentioned that lecture halls are not heated or cooled for a long time period after 17.00 which results with thermal discomfort in the morning hours. To prevent this, pre-heating strategy in lecture hall zones is evaluated. In the strategy, running the mechanical system 1-hour earlier (at 06.00) is investigated and results revealed that it is possible to optimize thermal comfort with 8,5% total heating and 3,9% total cooling consumption increase in the building. With pre-heating strategy it is possible to obtain better PMV values during winter months.

- Set-point variation

Along with pre-heating strategy, set-point variation is also conducted in lecture halls. Since lecture halls have the lowest PMV rates which are mainly negative, two different set-points with the value of 23°C and 24°C are investigated. According to results, winter PMV values are optimised from -0,90 range to -0,70. Also it is noted that, set-point increase in 12 lecture halls would increase the building's heating consumption by 5% on an annual basis.

- Optimisation of shading system

MED has nearly 30% window to wall ratio in the southern façade. In that way, optimisation of the shading system is very crucial in the southern zones. In order to block the disavantegeous solar gains, a movable shading element is constructed in the existing building. Although it is very positive for building, its scheduling should be considered, since solar gains are thought to be desirable in winter months, whereas, undesirable in hot summer months. Accodingly, positioning of the shading element is investigated using different time periods for off-positioning. Shading element is off positioned since the 30th of March in the baseline case. However, it was seen in the April is slightly cold in southern offices and seminar rooms. Consequently, keeping the shading element off positioned in April would be a precise solution to increase the thermal comfort in the offices and seminar rooms. However, since the May PMV index is high and thermal sensation is rather hot,it is better to position shading element “on” in May. After the analysis of optimum time period, 30th of April is evaluated to be the best time for shading elements to be closed in the south-oriented zone groups of the building.

- Effect of the clothing

In the analysis, effect of the clothing is investigated lastly. In DesignBuilder, clothing insulation values (clo) are taken as 1 and 0,5 with respect to the thermal comfort standards in summer and winter times respectively. Yet, it leads to peak values in April and October month PMV calculations. To overcome this problem and see the effect of the clothing in the building, model was simulated using 1 and 0,5 seperetaly for whole year. Results revealed that 0,5 clo between the months of May and October is convenient for occupants’ thermal comfort, rather than April and September as it is assumed in the baseline case.

In the study the effect of each strategy is presented above seperetaly. Design and operational strategies and their affect on the building’s thermal comfort is assessed. Yet, to get the optimum thermal comfort for the building, combination of each strategy is also evaluated. Combination of each strategy and baseline case comparison is given in the following Figure 7.1.

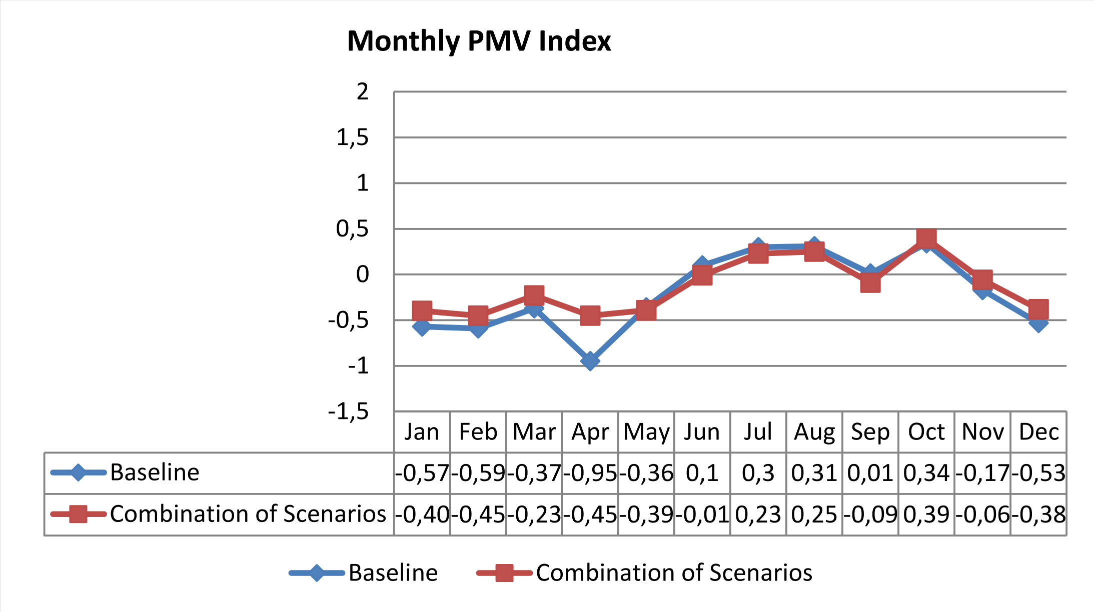


Figure 7.1: PMV comparison of baseline and combination of all strategies in overall building.

As given in the Figure 7.1, PMV index can be kept between +0,5 and -0,5 values as it is reccomended in the thermal comfort standards with the application of each evaluated strategy. In addition to building’s thermal comfort analysis, productivity will be also investigated. According to pre-defined calculation methods, thinking and typing productivity losses comparison of baseline case and combination of scenarios are given in Figure 7.2 and 7.3 for overall building.

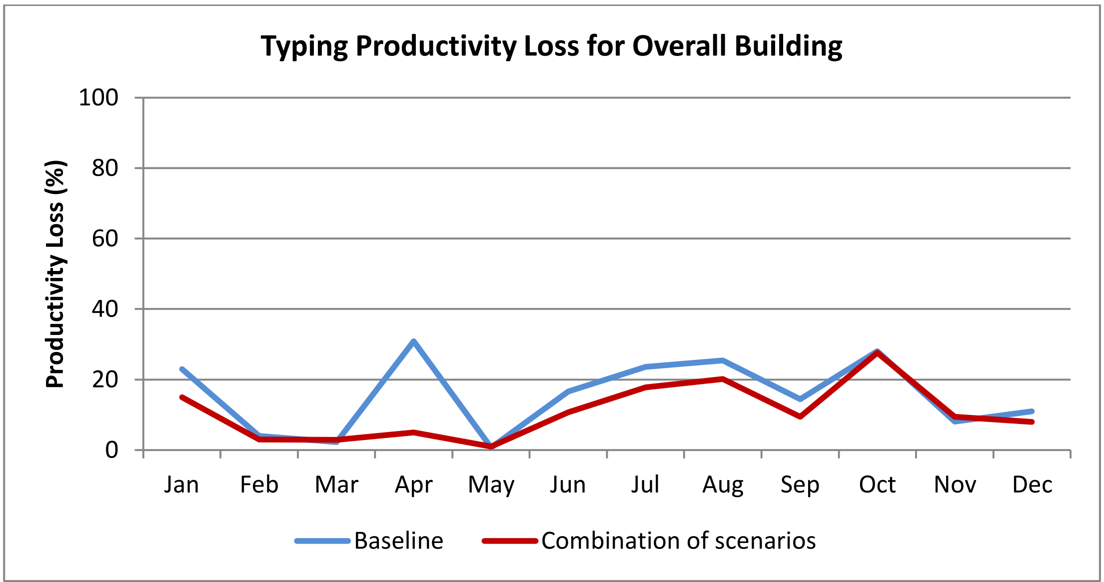


Figure 7.2: Typing productivity loss for overall building.

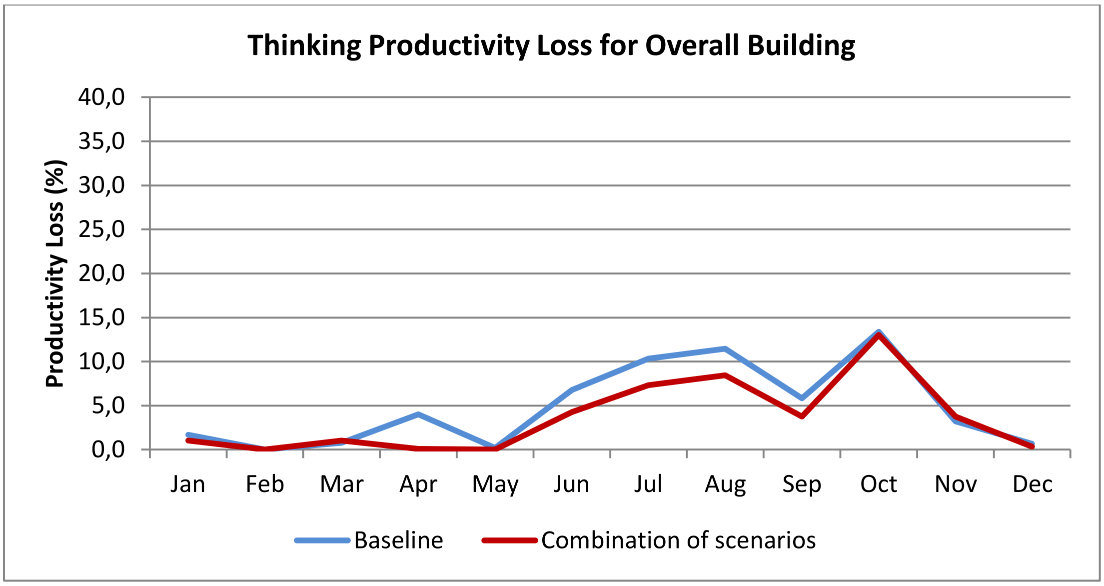
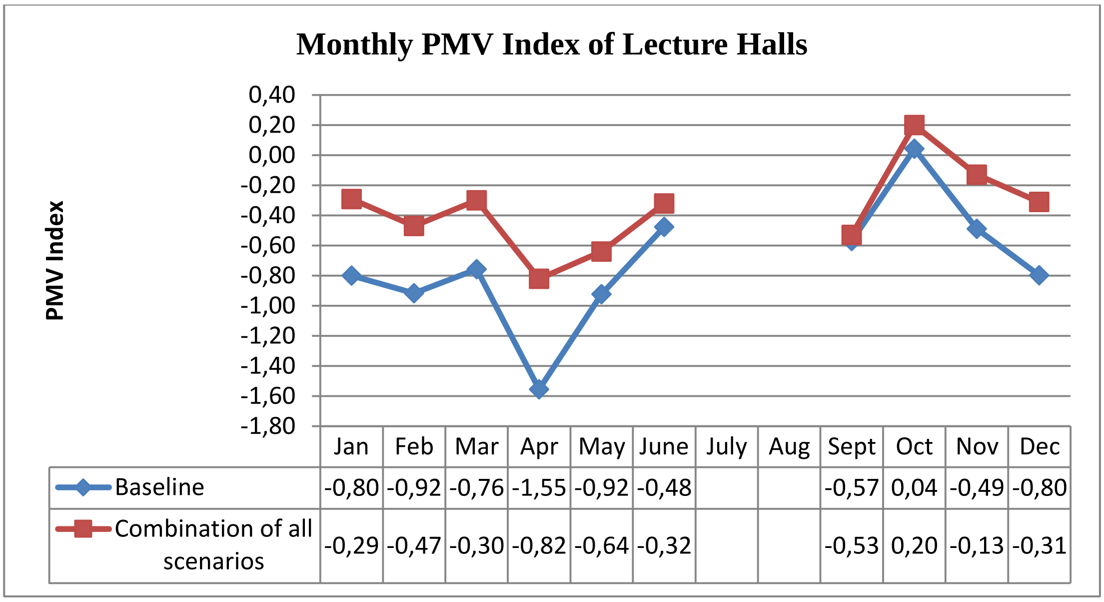


Figure 7.3: Thinking productivity loss for overall building.

As it can be seen from Figure 7.2 and 7.3, there is an improvement in the both thinking and typing productivity. According to results, typing productivity can be improved averagely 3% whereas, this value corresponds to 1,4% for thinking tasks. The most remarkable change in productivity loss improvement is in month April, since the PMV rate is adjusted to value of "-0,45" from "-0,95" which is considerably high.

As well as building's overall status lecture halls, seminar rooms, offices are also investigated seperetaly on the basis of both PMV index and productivity losses.

PMV and productivity comparison of lecture halls can be seen on Figure 7.4, 7.5 and 7.6 as it follows.



	Jan	Feb	Mar	Apr	May	June	July	Aug	Sept	Oct	Nov	Dec
Baseline	-0,80	-0,92	-0,76	-1,55	-0,92	-0,48			-0,57	0,04	-0,49	-0,80
Combination of all scenarios	-0,29	-0,47	-0,30	-0,82	-0,64	-0,32			-0,53	0,20	-0,13	-0,31

Figure 7.4: PMV comparison of baseline and combination of all strategies in lecture halls.

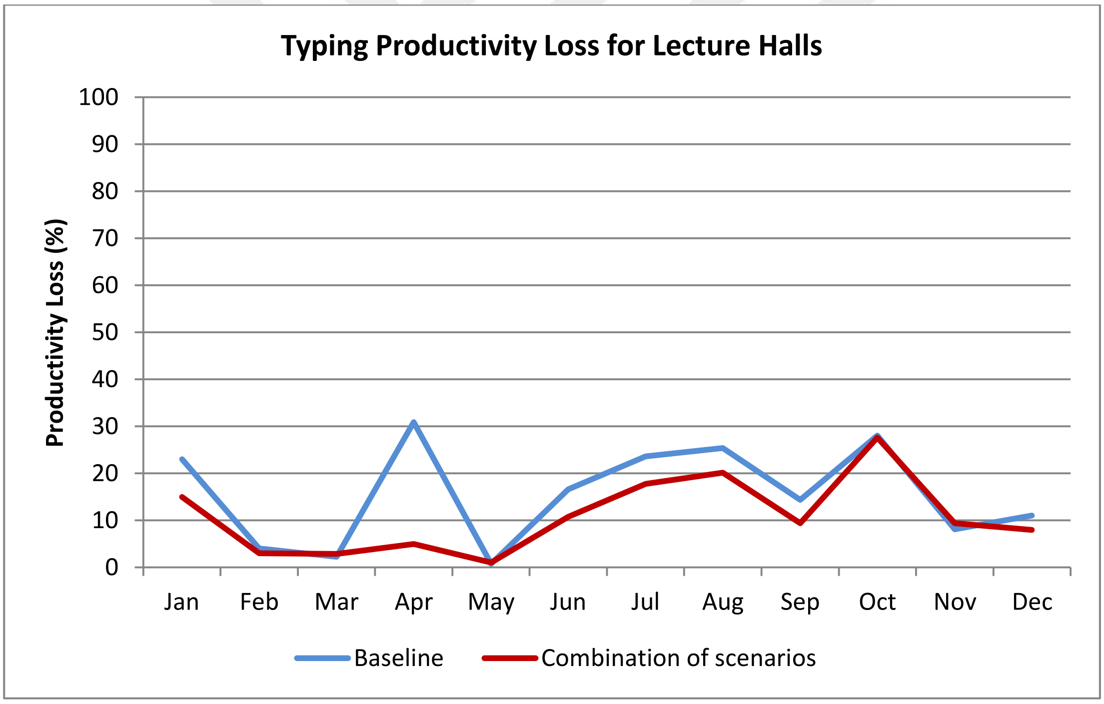


Figure 7.5: Typing productivity loss comparison of baseline and combination of all strategies in lecture halls.

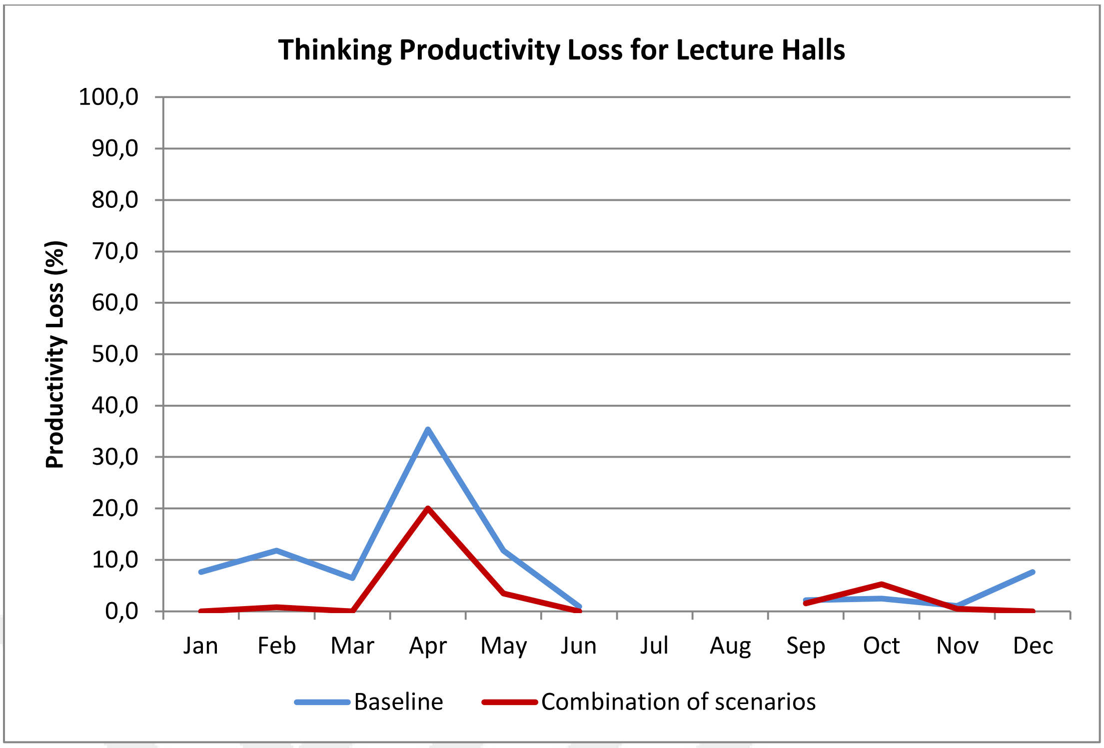
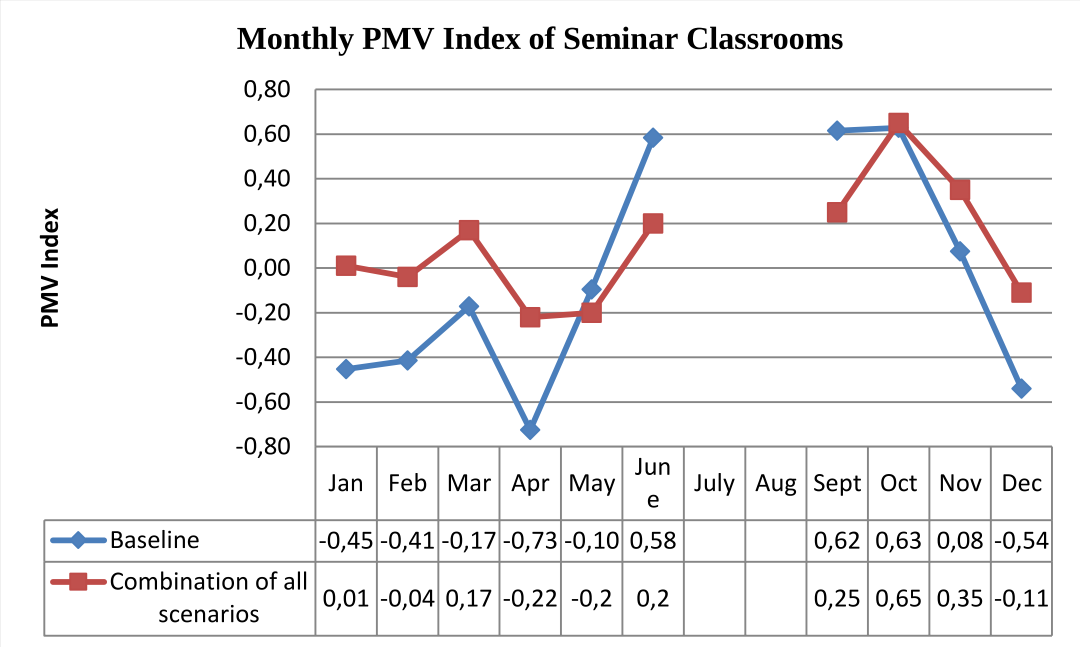


Figure 7.6: Thinking productivity loss comparison of baseline and combination of all strategies in lecture halls.

As depicted in Figure 7.4, lecture halls' PMV values have a significant improvement in each month. Apart from April, nearly all months stay in the reccomended interim of ±0,5 PMV value. It also affects the productivity losses in the lecture halls. Both typing and thinking productivity is progessed in lecture halls. In overall, combination of all scenarios has led 12% increase in typing productivity and 4,6% increase in thinking productivity.

Besides lecture halls, seminar classrooms are investigated following the same procedure. Monthly PMV index comparison of the reffered zones and productivity loss analysis can be found in Figure 7.7, 7.8 and 7.9.



	Jan	Feb	Mar	Apr	May	June	July	Aug	Sept	Oct	Nov	Dec
Baseline	-0,45	-0,41	-0,17	-0,73	-0,10	0,58			0,62	0,63	0,08	-0,54
Combination of all scenarios	0,01	-0,04	0,17	-0,22	-0,2	0,2			0,25	0,65	0,35	-0,11

Figure 7.7: PMV comparison of baseline and combination of all strategies in seminar classrooms.

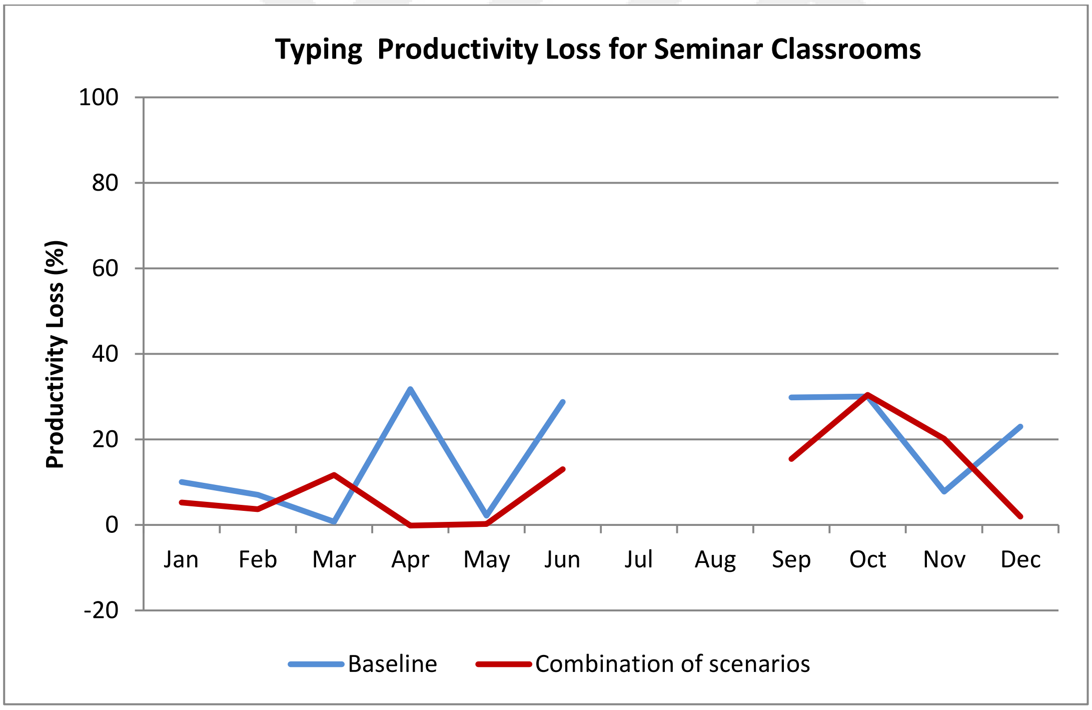


Figure 7.8: Typing productivity loss comparison of baseline and combination of all strategies in seminar classrooms.

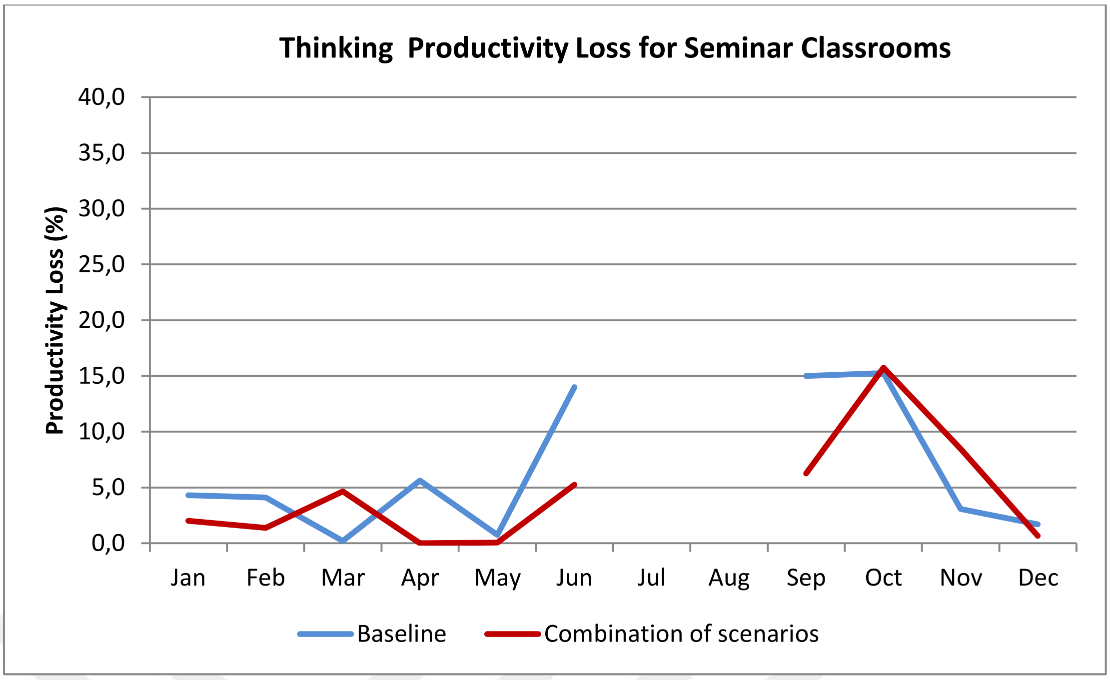
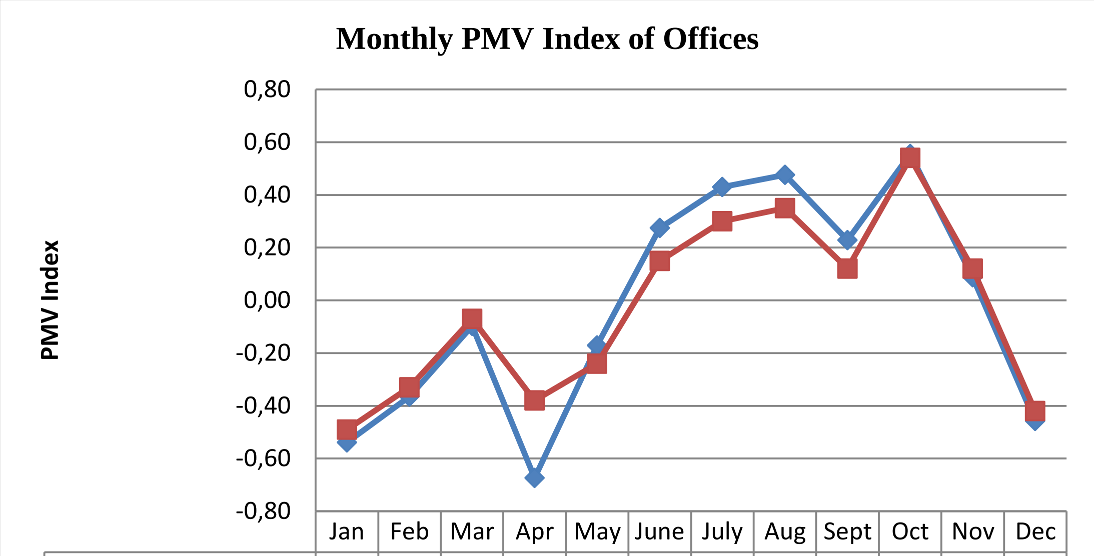


Figure 7.9: Thinking productivity loss comparison of baseline and combination of all strategies in seminar classrooms.

Given in Figure 7.7, combination of all scenarios has resulted with a considerable impact on PMV indices of seminar classrooms. Analysis revealed that, with the refferred design and operational strategies, it is possible to keep seminar rooms in reccomended PMV values in the thermal comfort standards. Following the comfort analysis, productivity comparison is represented in the Figure 7.8 and 79. According to calculations, especially improvement in productivity is observed in most of the months. However, especially in March and November productivity is decreased. In baseline case March and November PMV rates have negative values, whereas, in the last scenario the rates correspond to positive values. It leads loss in productivity, since the lower PMV indices (up to -0,5) is more preferable in thinking and typing productivity. Still in overall, 6% productivity increase in typing tasks and 1,6% productivity increase in thinking tasks is obtained by the application of the scenarios.

Lastly, south-oritented office zones are investigated. PMV index and productivity loss percentages are given in Figure 7.10, 7.11 and 7.12.



	Jan	Feb	Mar	Apr	May	June	July	Aug	Sept	Oct	Nov	Dec
Baseline	-0,54	-0,37	-0,10	-0,67	-0,17	0,27	0,43	0,48	0,23	0,56	0,09	-0,46
Combination of all scenarios	-0,49	-0,33	-0,07	-0,38	-0,24	0,15	0,30	0,35	0,12	0,54	0,12	-0,42

Figure 7.10: PMV comparison of baseline and combination of all strategies in offices.

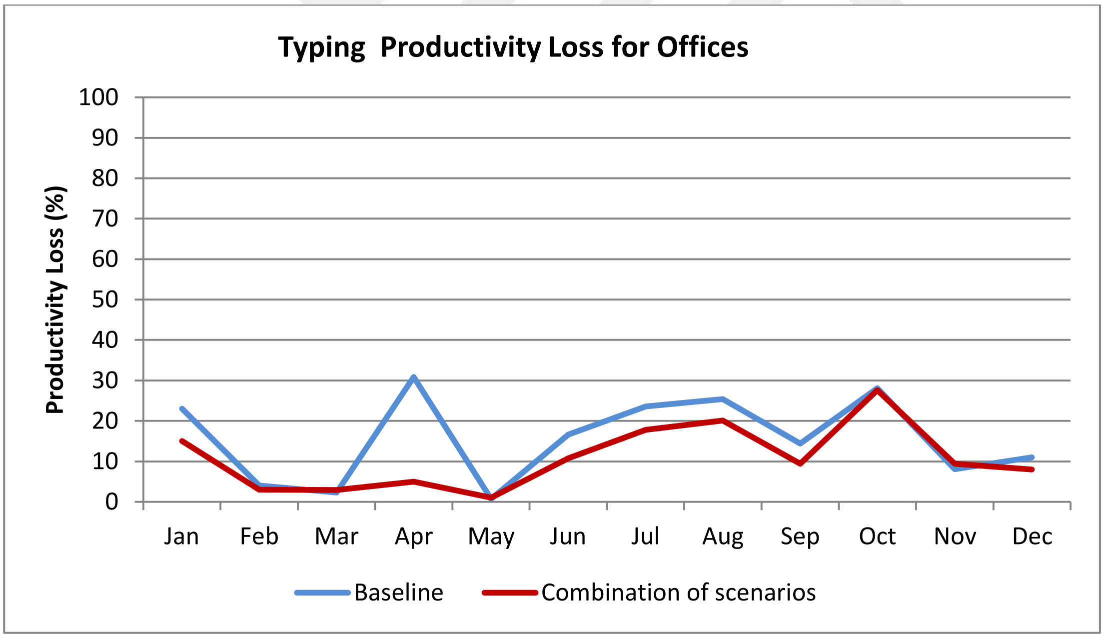


Figure 7.11: Typing productivity loss comparison of baseline and combination of all strategies in offices.

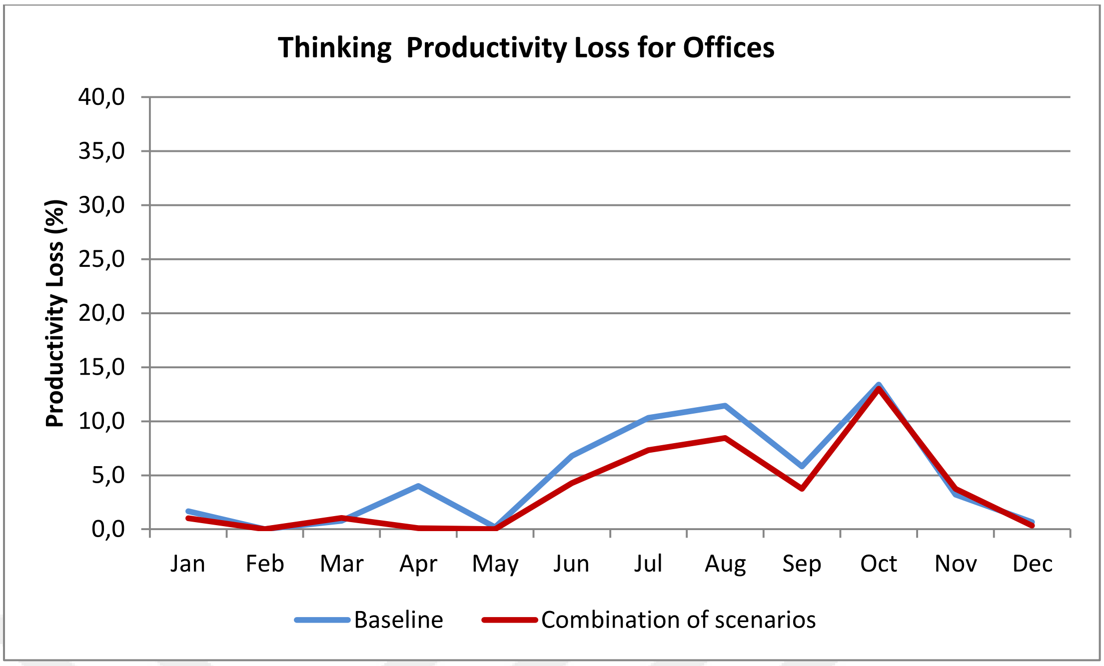


Figure 7.12: Thinking productivity loss comparison of baseline and combination of all strategies in offices.

PMV comparison of baseline case and combination of all scenarios as given in the Figure 7.10 depicts that evaluated scenarios have a positive effect in the zones. Similar to seminar rooms, offices' PMV index are also stays in the reccomended range based on the thermal comfort standards. Furthermore, typing and thinking productivity are increased by 5% and 1,2% respectively within the application of all scenarios.

In the present research, MED is investigated in the terms of thermal comfort, energy consumption and producivity. The building is examined both as a whole and in parts to see the effects in more detail. In conclusion, it is already presented that all evaluated control and operational scenarios had a positive impact in all fields. Combination of all scenarios resulted with better PMV values for both whole of the building and zones accordingly. Consequently, better thermal indoor environment has lead to increased productivity for the occupants. When it is thought that the building is being used as an classroom and office; productivity aspect becomes significantly crucial. Though, better IAQ corresponds to up to 10% productivity increase in both typing and thinking tasks. On the other hand, as well as energy consumption is concerned, 8,5% energy consumption increase in heating and 14,2% in cooling is observed by application of the scenarios.

Yet, it can be said that there is still room for improvement in the building to enhance both energy efficiency and thermal comfort. One of the main drawbacks of the building is the lack of renewable energy sources which causes higher energy consumptions. Still, with establishing design and operational strategies to the existing building (AHU optimisation, shading element optimisation, set-point variation, pre-heating and clothing strategy) it is possible to maintain thermal comfort within the desired and recommended values, including both lecture halls, seminar rooms and offices.

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APPENDICES

APPENDIX A: Thermal Insulation References.
APPENDIX B: Ventilation, Lighting, Occupancy Rates for Each Zone in MED.

APPENDIX A

Table A.1 : Metabolic Rates for Typical Tasks.[36]

Activity	Metabolic Rate		
	Met Units	W/m^2	Btu/h.ft^2
Resting			
Sleeping	0,7	40	13
Reclining	0,8	45	15
Seated, quiet	1,0	60	18
Seated, relaxed	1,2	70	22
Walking (on level surface)			
0.9 m/s, 3.2 km/h, 2.0 mph	2,0	115	37
1.2 m/s, 4.3 km/h, 2.7 mph	2,6	150	48
1.8 m/s, 6.8 km/h, 4.2 mph	3,8	220	70
Office activities			
Reading, seated	1,0	55	18
Writing	1,0	60	18
Typing	1,1	65	20
Filing, seated	1,2	70	22
Filing, standing	1,4	80	26
Walking about	1,7	100	31
Lifting/packing	2,1	120	39
Driving/flying			
Automobile	1,0 – 2,0	60 – 115	18 – 37
Aircraft, routine	1,2	70	22
Aircraft, instrument landing	1,8	105	33
Aircraft, combat	2,4	140	44
Heavy vehicle	3,2	185	59
Misc. occupational activities			
Cooking	1,6 – 2,0	95 – 115	29 – 37
House cleaning	2,0 – 3,4	115 – 200	37 – 63
Seated, heavy limb movement			
Machine work			
Sawing (table saw)	1,8	105	33
Light (electrical industry)	2,0 – 2,4	115 – 140	37 – 44
Heavy	4,0	235	74
Handling 50 kg (100 lb) bags	4,0	235	74
Pick and shovel work	4,0 – 4,8	235 – 280	74 – 88

Misc. leisure activities			
Dancing, social	2,4 – 4,4	140 – 255	44 – 81
Calisthenics/exercise	3,0 – 4,0	175 – 235	55 – 74
Tennis, single	3,6 – 4,0	210 – 270	66 – 74
Basketball	5,0 – 7,6	290 – 440	90 – 140
Wrestling, competitive	7,0 – 8,7	410 – 505	130 – 160

Table A.2 : Metabolic rate for specific activities (EN ISO 8996, 2004)

Activity		Metabolic Rate W/m^2
Sleeping		40
Reclining		45
At rest, sitting		55
At rest, standing		70
Walking on the level, even path, solid		
1. without load	at 2 km/h	110
	at 3 km/h	140
	at 4 km/h	165
	at 5 km/h	200
2. with load	10 kg, 4 km/h	185
	30 kg, 4 km/h	250
Walking uphill, even path, solid		
1. without load	5˚ inclination, 4 km/h	180
	15˚ inclinations, 3 km/h	210
	25˚ inclinations, 3 km/h	300
2. with load of 20 kg,	15˚ inclinations, 4 km/h	270
	25˚ inclinations, 4 km/h	410
Walking downhill at 5 km/h, without load	5˚ inclinations	135
	15˚ inclinations	140
	25˚ inclinations	180
Ladder at 70˚, climbing at a rate of 11,2 m/min		
without load		290
with a 20 kg load		360
Pushing or pulling a tip-wagon, 3,6 km/h, even path, solid		
pushing force: 12 kg		290
pulling force: 16 kg		375
Pushing a wheelbarrow, even path, 4,5 km/h, rubber tyres, 100 kg load		230
Filing iron	42 file strokes/min	100
	60 file strokes/min	190
Work with a hammer, 2 hands, mass of the hammer 4,4 kg, 15 strokes/min		290
Carpentry work	hand sawing	220
	machine sawing	100
	hand planing	300
Brick-laying, 5 brick/min		170
Screw driving		100
Digging a trench		290

Sedentary activity (office, dwelling, school, laboratory)	70
Standing, light activity (shopping, laboratory, light industry)	95
Standing, medium activity (shop assistant, domestic work, machine work)	115
Work on machine tool	
Light (adjusting, assembling)	100
Medium (loading)	140
Heavy	210
Work with a hand tool	
Light (light polishing)	100
Medium (polishing)	160
Heavy (heavy drilling)	230

APPENDIX B

Table A.3 : Ventilation, Lighting, Occupancy Rates for Each Zone in MED

Zone	Area (m^2)	Ventilation Rate ($l/s/m^2$)	Occupancy ($people/m^2$)	Lighting density (W/m^2)
ZK01 - GİRİŞ HOLÜ	26,88	n/a	0,1065	4,32
ZK02 - YANGIN MERDİVENİ	20,11	n/a	0,1065	5,77
ZK03 - GÖREVLİ ODASI	19,67	1,4	0,1122	5,90
ZK04 - ASANSÖR	n/a	n/a	n/a	n/a
ZK05 - ITU MAGAZA	175,58	1,1	0,15	9,25
ZK06 - MUTFAK	83,95	41,4	0,0956	10,36
ZK07 - MESCIT	37,73	3,2	0,3	9,22
ZK08 - KULUP ODASI	66,7	1,7	0,5	8,70
ZK09 - KULUP ODASI	66,68	1,7	0,2313	8,70
ZK10 - WC	24,46	n/a	0,1065	9,48
ZK11 - WC KORIDOR	11,17	n/a	0,1065	10,38
ZK12 - ENG. WC	2,9	n/a	0,1065	8,97
ZK13 - TEMIZLIK ODASI	6,42	n/a	0,1015	9,03
ZK14 - WC	22	n/a	0,1065	10,55
ZK15 - ELEKTRIK ODASI	30,75	n/a	0	7,54
ZK16 - KAZAN DAIRESI	47,99	23,7	0,11	7,25
ZK17 - KLIMA DAIRESI	130,54	12,8	0,11	8,89
ZK18 - SATIS BIRIMI	29,04	n/a	0,07	7,99
ZK19 - GIYINME	33,29	2,3	0	8,71
ZK20 - TOPLANTI ODASI	26,1	n/a	0,103	11,11
ZK21 - OFIS	66,53	1,1	0,103	8,72
ZK22 - ASANSOR		n/a		n/a
ZK23- ELEKTRIK ODASI	9,24	n/a	0	12,55
ZK24- YANGIN MERD.	21,3	n/a	0	5,45
ZK25 - HOL VE KORIDORLAR	3481,17	1,8	0,1065	7,29
1NK01- AMFI 1	195,35	2,4	0,2017	7,13
1NK02 - AMFI 2	143,2	2,4	0,2017	9,72

1NK03 - AMFI 3	143,18	2,4	0,2017	9,72
1NK04 - AMFI 4	145,15	2,4	0,2017	9,59
1NK05 - AMFI 5	146,87	2,4	0,2017	9,48
1NK06 - AMFI 6	145,91	2,4	0,2017	9,54
1NK07 - YANGIN MERD.	21,52	n/a	0	5,39
1NK08 - ELEKTRIK ODASI	11,51	n/a	0	5,04
1NK09 - YANGIN KORIDOR	9,68	n/a	0	11,98
1NK10 - ASANSOR		n/a		n/a
1NK11 - WC	20,25	n/a	0,1065	11,85
1NK12 - ENG. WC	2,91	n/a	0,1065	8,93
1NK13 - WC KORIDOR	3,6	n/a	0,1065	7,22
1NK14 - WC	23,63	n/a	0,1065	8,13
1NK15- TEKNIK MEKAN	24,55	n/a	0	11,73
1NK16- WC	25,18	n/a	0,1065	7,63
1NK17 - TEKNIK MEKAN	21,66	n/a	0	13,30
1NK18 - WC	26,42	n/a	0,1065	7,27
1NK19 - ASANSOR		n/a		n/a
1NK20 - ELEKTRIK ODASI	3,66	n/a	0	13,11
1NK21 - YANGIN KORIDOR	4,99	n/a	0	9,62
1NK22 - YANGIN MERD.	21,19	n/a	0	4,53
1AK07 - YANGIN MERD.	21,57	n/a	0	5,38
1AK08 - TEKNIK MEKAN	11,51	n/a	0	5,04
1AK09 - ASANSOR	n/a	n/a	0	n/a
1AK10 - YANGIN KORIDOR	9,68	n/a	0	11,98
1AK11 - TEKNIK MEKAN	77,94	n/a	0	7,44
1AK12 -WC	24,73	n/a	0,1065	7,76
1AK13- TEKNIK MEKAN	20,71	n/a	0	16,80
1AK14 -WC	25,94	n/a	0,1065	7,40
1AK15 - ASANSOR	n/a	n/a	0	n/a
1AK16 - ELEKTRIK OD	3,65	n/a	0	15,89
1AK17 - YANGIN KORIDOR	5,16	n/a	0	11,24
1AK18- YANGIN MERD.	21,13	n/a	0	5,49
1AK19 - OFIS	35,98	1,4	0,103	9,67
1AK20 - OFIS	35,82	1,4	0,103	9,72
1AK21 - OFIS	26,91	1,5	0,103	8,62
1AK22 - OFIS	27,46	1,5	0,103	8,45
1AK23 - OFIS	27,47	1,5	0,103	8,45
1AK24 - OFIS	26,91	1,5	0,103	8,62

1AK25- OFIS	26,55	1,6	0,103	8,74
1AK26- OFIS	27,1	1,5	0,103	8,56
1AK27-OFIS	27,1	1,5	0,103	8,56
1AK28 - OFIS	26,55	1,6	0,103	8,74
1AK29 - OFIS	26,71	1,6	0,103	8,69
1AK30 - OFIS	26,56	1,6	0,103	8,73
2NK01- AMFI 7	195,28	2,4	0,2017	7,13
2NK02 - AMFI 8	143,18	2,4	0,2017	9,72
2NK03 - AMFI 9	143,17	2,4	0,2017	9,72
2NK04 - AMFI 10	145,13	2,4	0,2017	9,59
2NK05 - AMFI 11	146,84	2,4	0,2017	9,48
2NK06 - AMFI 12	146,04	2,4	0,2017	9,53
2NK07 - YANGIN MERD.	22,54	n/a	0	5,15
2NK08 - ELEKTRIK ODASI	11,51	n/a	0	5,04
2NK09 - ASANSOR	n/a	n/a		n/a
2NK10 - YANGIN KORIDOR	9,68	n/a	0	11,98
2NK11 - WC	19,91	n/a	0,1065	11,65
2NK12 - ENG. WC	3,32	n/a	0,1065	7,83
2NK13 - WC KORIDOR	2,89	n/a	0,1065	9,00
2NK14 - WC	20,02	n/a	0,1065	11,59
2NK15- TEKNIK MEKAN	24,56	n/a	0	14,17
2NK16- WC	25,18	n/a	0,1065	9,21
2NK17 - TEKNIK MEKAN	21,66	n/a	0	16,07
2NK18 - WC	26,42	n/a	0,1065	8,78
2NK19 - ASANSOR	n/a	n/a		n/a
2NK20 - ELEKTRIK ODASI	3,65	n/a	0	15,89
2NK21 - YANGIN MERD	21,13	n/a	0	5,49
2NK22 - YANGIN KORIDOR	5,1	n/a	0	11,37
2NK23 - SEMINER 1	48,07	9,2	0,2017	7,24
2NK24 - SEMINER 2	47,85	5,9	0,2017	7,27
2NK25 - SEMINER 3	73,58	5,7	0,2017	7,88
2NK26 - SEMINER 4	73,59	5,7	0,2017	7,88
2NK27 - SEMINER 5	72,62	5,9	0,2017	7,99
2NK28 - SEMINER 6	72,62	5,9	0,2017	7,99
2NK19 - SEMINER 7	72,11	6,7	0,2017	8,04
2AK07 - YANGIN MERD.	21,52	n/a	0	5,39
2AK08 - TEKNIK MEKAN	11,51	n/a	0	5,04
2AK09 - ASANSOR	n/a	n/a		
2AK10 - YANGIN KORIDOR	9,62	n/a	0	12,06
2AK11 - TEKNIK MEKAN	77,94	n/a	0	7,44

2AK12 -WC	24,73	n/a	0,1065	9,38
2AK13- TEKNIK MEKAN	20,71	n/a	0	16,80
2AK14 -WC	25,94	n/a	0,1065	8,94
2AK15 - ASANSOR	n/a	n/a		
2AK16 - ELEKTRIK OD	3,65	n/a	0	15,89
2AK17 - YANGIN KORIDOR	5,1	n/a	0	11,37
2AK18- YANGIN MERD.	21,13	n/a	0	5,49
2AK19 - SEMINER	35,98	9,9	0,2017	9,67
2AK20 - SEMINER	35,82	6,4	0,2017	9,72
2AK21 - OFIS	26,91	1,5	0,103	8,62
2AK22 - OFIS	27,46	1,5	0,103	8,45
2AK23 - OFIS	27,47	1,5	0,103	8,45
2AK24 - OFIS	26,91	1,5	0,103	8,62
2AK25- OFIS	26,55	1,6	0,103	8,74
2AK26- OFIS	27,1	1,5	0,103	8,56
2AK27-OFIS	27,1	1,5	0,103	8,56
2AK28 - OFIS	26,55	1,6	0,103	8,74
2AK29 - SEMINER	53,97	8,1	0,2017	10,75

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