Experimental Analysis of Thermal Environment in an Air Conditioned Space to Evaluate the Measures for its Improvement

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Abstract-*With the improvement of standard of living, air-conditioning has widely been applied. However, health problems associated with air-conditioning systems and indoor air quality appear more frequently. In this paper gives an review on Effect of Supply Air Temperature on Indoor Thermal Comfort in a Room with Radiant Heating and Mechanical Ventilation. Air-conditioning systems have been used in many parts of the world. The purpose of most systems is to provide thermal comfort and an acceptable indoor air quality (IAQ) for occupants. With the improvement of standard of living, occupants require more and more comfortable and healthful indoor environment. People spend 80-90% of their time indoors, and indoor environment has important effects on human health and work efficiency.*

Keywords*: Radiant heating, Mechanical ventilation, Supply air temperature, Thermal comfort, Air conditioning, Indoor air quality.*

I. INTRODUCTION

Radiant heating systems, such as floor heating (FH) systems and ceiling heating (CH) systems are being considered as energy efficient and comfortable heating systems and have been extensively used in modern buildings [1-4]. Many modern buildings with radiant heating systems generally have increased thermal insulation and air tightness. Mixing and displacement air distribution are the main mechanical ventilation principles applied in both residential and non-residential buildings. Displacement ventilation (DV) is often a suitable choice for offices, where high air quality must be maintained and where internal heat gains can be very high due to the presence of employees and office equipment. Under these conditions it has the potential to provide better air quality [1] and energy savings of 20% to 30% [1-5]. Although the main aim of DV is the removal of airborne pollution, it can also provide cooling, although the cooling potential is limited to about 40 W/m^2 [3] due to the risk of draught discomfort caused by the layer of turbulent cold air created ast floor level [7]. In summer this cooling capacity may not be sufficient for office buildings with large window areas. To reduce the peak cooling load and to avoid draught, DV can be combined with a radiant system, such as a chilled ceiling or floor. An important advantage of such radiant systems is the low temperature difference between the room air and the cooling surface, making it possible to use renewable energy sources, such as heat pumps, solar collectors, or existing natural water temperatures (e.g., in boreholes, lakes, and rivers). In a typical displacement ventilated room the fresh air is supplied at a temperature lower than room air temperature, so that half of the temperature difference between the supply air and the extract air temperature takes place by mixing at floor level and a linear vertical gradient accounts for the rest of the difference [8]. The increase of air temperature at floor level occurs due to heat transfer from the warmer ceiling to the floor by radiation and the subsequent convective transfer of heat from the floor to the supply air at floor level [9].When combined with a radiant system, the air temperature and air distribution may be different. When a chilled ceiling is used to remove the major part of the cooling load, downward airflow from the chilled ceiling panels is strong and the vertical temperature distribution is more uniform compared to a DV system. The stratified displacement airflow pattern is destroyed and the room air becomes well mixed with a relatively uniform temperature and contaminant distribution [11], leading to more acceptable thermal conditions [12]. Although a number of studies of DV combined with chilled ceiling have been made, the combination of DV with floor cooling has seldom been studied. Floor cooling could provide a potentially viable solution when the same system is used for heating and when ceiling and wall cooling is not a technically viable solution. Floor cooling can work well and is being increasingly used in spaces with large fenestration, atriums, and entrance halls [13], e.g., in the new Bangkok airport terminal [14]. The question is, whether it can work as well in spaces that are smaller such as in offices with external solar shading. The use of such a system in an office (17 m^2) was studied by [15], and the measured vertical air temperature differences were generally higher than the limits recommended by the standards, although the ability to remove a contaminant from a simulated active pollution source at the breathing level was good at higher supply airflow rates (above 2 L per second per square meter floor area). The higher vertical air temperature differences that occurred were explained by the fact that the radiant heat transferred from the warm ceiling was directly absorbed by the cold floor, so little or no convective heat was transferred from the cold floor to the supply air by convection. Another set of experimental measurements with human subjects present confirmed that there was a risk of high vertical air temperature difference between ankles and head of a seated person (1.1–0.1 m above the floor) of up to 5 K, and of related problems with room air temperature control and possible local discomfort due to cold feet, at lower levels of thermal insulation [16].

The present study focused on an evaluation of the indoor environment in a simulated office with DV, floor cooling, and external blinds to reduce direct solar radiation. Thermal comfort and ventilation effectiveness were investigated at the same time over a range of nominal air change rates, from a typical value of 4.5 air changes per hour, down to as low as 1.5 air changes per hour. The purpose was to investigate the possibility of energy conservation by decreasing the ventilation rate to lower levels while still exploiting the advantages of DV, while adjusting floor temperatures so as to maintain a comfortable thermal environment in terms of the room air temperature (26◦C) at a reference point in the middle of the room. Vertical air and operative temperature profiles, air velocity profiles, and thermal manikin equivalent temperatures were used to assess the thermal environment. In ventilation effectiveness measurements, local air change index (ε^c P) was obtained by tracer gas mean age of air measurements at the breathing level of seated and standing persons in order to investigate the distribution of fresh air to the occupants, and contaminant removal effectiveness (ε^c) was obtained by steady-state tracer gas measurements in the occupied zone in order to investigate the ability of the system to remove contaminants emitted by a simulated occupant.

II. METHOD

A. Test room and measuring instruments

The experimental measurements were carried out in an air conditioned room or office with the dimensions of 8.0 L × 4.0 B × 4.0 H in m. The cooling load consisted of an 8 m2 shaded window, simulated by means of a radiant heating wall to simulate solar heat gains in summer conditions, together with the internal heat gains created by two desk lamps, two heated metal boxes simulating office equipment (computer, monitor, etc.), and two simulated occupants, represented by an electrically heated dummy and a thermal manikin. The thermal manikin, with 17 individually controlled bodies sections [12], was also used to record and analyze the heat transfer from different body parts to allow for a subsequent calculation of equivalent temperatures. A semi-circular perforated displacement diffuser, suitable for the supply of large volumes of moderately cooled air, was placed on the back wall opposite the window as the air supply outlet, and a circular extract air terminal (diameter 0.18 m) was installed in the middle of the ceiling. The cooling load was removed by the cooled air supply and by a radiant floor cooling system. Surface temperatures, air temperature, operative temperature, relative humidity, and air velocity were measured and logged with the same instrumentation as described in [13]. The measurement uncertainty of the surface, air, and operative temperatures was ±0.3◦C; for the air velocity, it was in the range of ±0.02 m/s from . This source was attached to the heated metal dummy placed opposite the manikin, so that the tracer gas supplied could follow an air distribution pattern similar to that of an occupant's thermal plume. The CO2 tracer gas released represented the exhaled and body odor bioeffluents that are emitted from an occupant. R134a was used as the tracer gas in measurements of the local air change index. Concentration measurements and gas dosing in the room were carried out using two Photo acoustic multigas monitors and two multipoint samplers and dosers. The measurement uncertainty stated by the manufacturer of the multi-gas monitor was 1% of the measured value.

B. Thermal comfort model

Thermal comfort model is the base of thermal comfort evaluation methods and thermal environment standards. The purpose of this section is to describe the selection of thermal comfort model used in the study. According to the simplified method of human thermal system, there are mainly three types of thermal comfort models including single-node model, two-node model, and multi-node model. In addition, there is another model called adaptive model mainly for non-air conditioned buildings. The Predicted Mean Vote/Predicted Percentage Dissatisfied (PMV/ PPD) model is the most widely used single node model at present. Based on human thermal equilibrium theory, it is mainly used to predict human thermal responses and evaluate the overall thermal sensation in steady-state thermal environment controlled by air-conditioning systems. The two-node and multi-node models all consider the physiological parameters and thermoregulatory mechanisms of human body, and they can be used to predict human thermal comfort in dynamic and non-uniform environments. The so-called adaptive model is mainly used for naturally ventilated buildings. It is not actually used to predict people's comfort responses, but is used to predict thermal environment parameters which can make people feel comfortable. It reflects the indoor comfort temperature variation trends along with the mean monthly outdoor air temperature. As for the determination of indoor design thermal parameters in air-conditioning system design, it is based on such a steady-state thermal environment well controlled by air-conditioning system. A lot of standards such as ISO 7730, [38]ASHRAE Standard 55-2010,[23] Chinese Standard GB

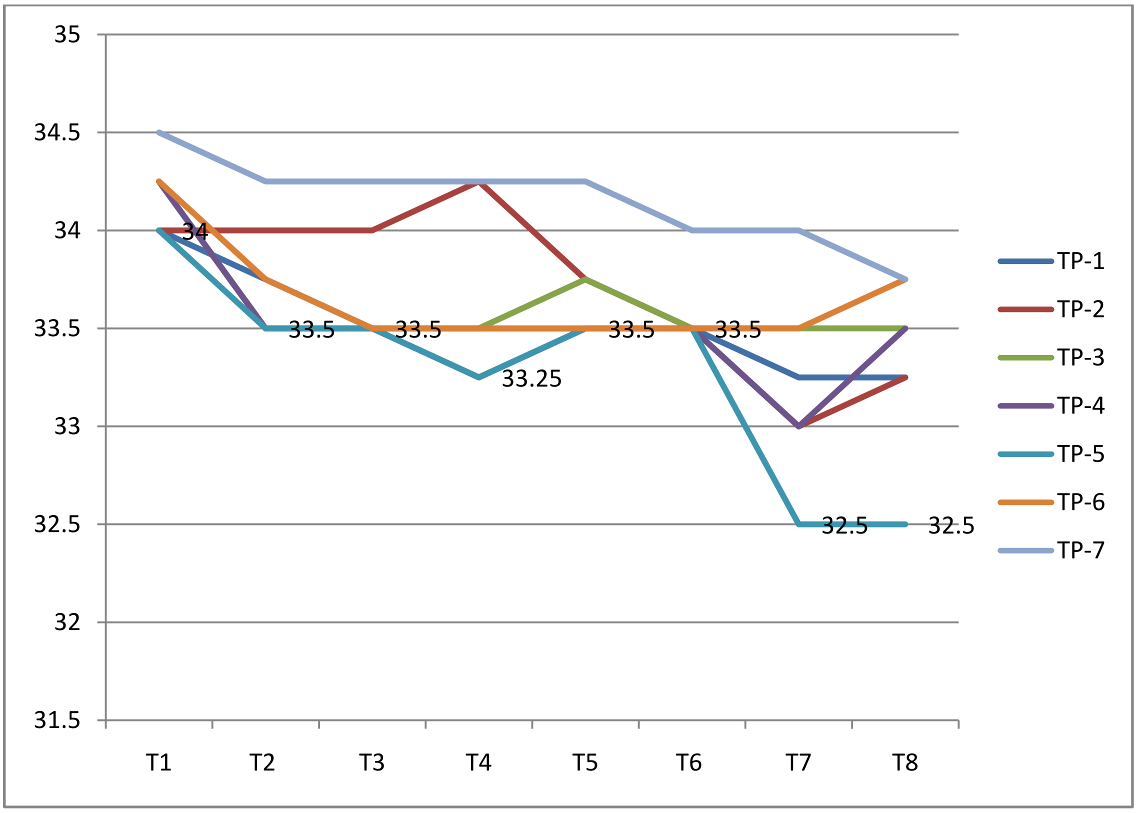
50019-2003,[24] JGJ134-2010,[25] and GB 50736-2012,[30] all use the PMV/PPD indices to describe and evaluate thermal environment, and also give the suggested design parameters of indoor thermal environment based on these indices. However, those provisions are aimed at conventional convectional air conditioning systems. The purpose of this study is to determine the design thermal environment parameters of residential buildings which employ radiant cooling systems. The thermal environment is well controlled and belongs to steady state thermal environment, so the PMV/PPD model was also be used to derive the reasonable indoor thermal microclimate parameters combinations in the paper. The PMV/PPD model has been validated by many laboratory and field studies around the world since many years ago. Besides, early researchers conducted a large number of experimental studies on thermal comfort in radiant cooled environment and proposed that the PMV/PPD model was also applicable to radiant cooled environment.44 Factors that affect the PMV are metabolic rate, clothing insulation, air temperature, mean radiant temperature, air speed, and relative humidity. PPD is an index expressing the thermal comfort level as a percentage of thermally dissatisfied people, and is directly determined from PMV. Much more details including calculation methods of PMV and PPD are described in ISO 7730. It recommends the values of PMV/ PPD indices to be adopted: 0.5 ≤ PMV≤ 0.5; PPD≤10%. The main methodology of the paper is using MATLAB programming for determining combinations of microclimate parameters, based on the PMV/PPD model.

III. RESULT

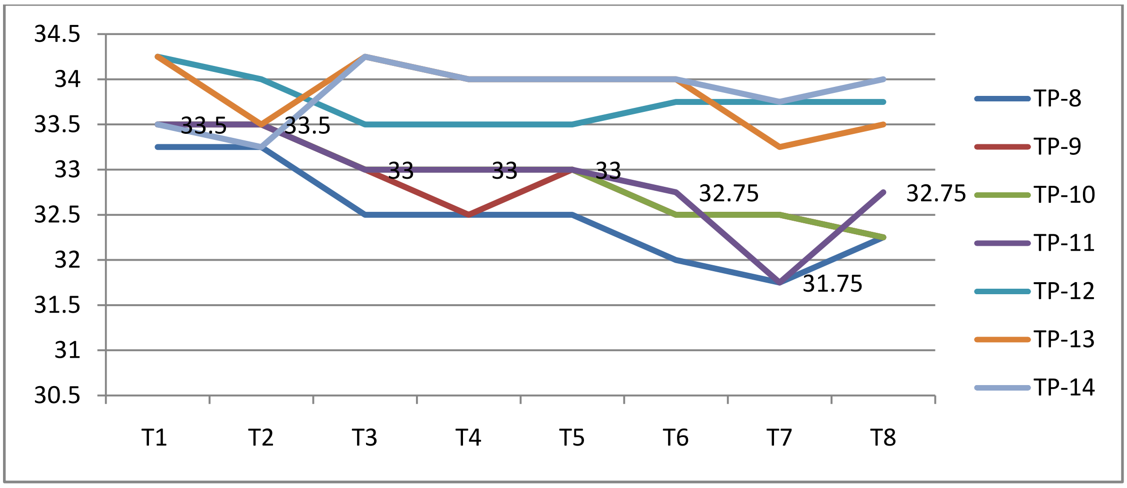
In study all studies at different temperature and time is taken with AC/Fan off and on all the conditions

Condition- AC on FAN off TP-testing points t-temperature total testing points-21

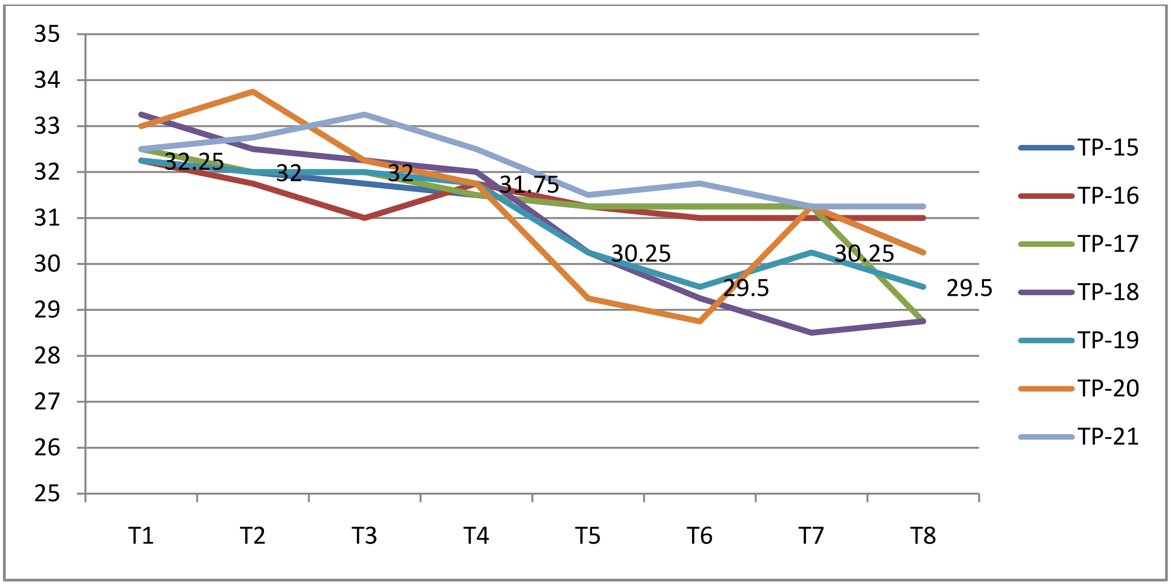
	T1	T2	T3	T4	T5	T6	T7	T8
TP-1	34	33.75	33.5	33.25	33.5	33.5	33.25	33.25
TP-2	34	34	34	34.25	33.75	33.5	33	33.25
TP-3	34.25	33.5	33.5	33.5	33.75	33.5	33.5	33.5
TP-4	34.25	33.5	33.5	33.5	33.5	33.5	33	33.5
TP-5	34	33.5	33.5	33.25	33.5	33.5	32.5	32.5
TP-6	34.25	33.75	33.5	33.5	33.5	33.5	33.5	33.75
TP-7	34.5	34.25	34.25	34.25	34.25	34	34	33.75



	T1	T2	T3	T4	T5	T6	T7	T8
TP-8	33.25	33.25	32.5	32.5	32.5	32	31.75	32.25
TP-9	33.5	33.5	33	32.5	33	32.5	32.5	32.25
TP-10	33.5	33.5	33	33	33	32.5	32.5	32.25
TP-11	33.5	33.5	33	33	33	32.75	31.75	32.75
TP-12	34.25	34	33.5	33.5	33.5	33.75	33.75	33.75
TP-13	34.25	33.5	34.25	34	34	34	33.25	33.5
TP-14	33.5	33.25	34.25	34	34	34	33.75	34

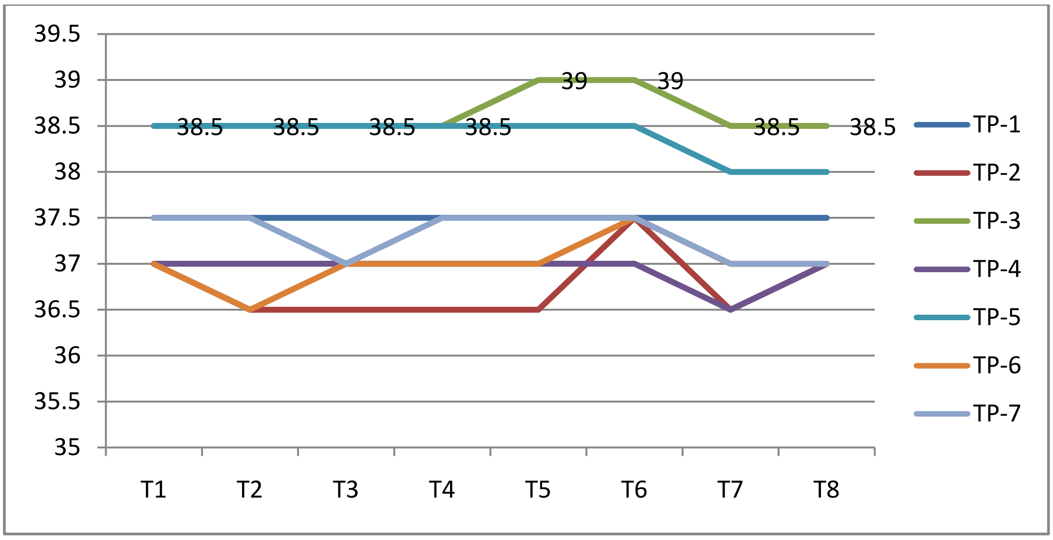


	T1	T2	T3	T4	T5	T6	T7	T8
TP-15	32.25	32	31.75	31.5	31.25	31.25	31.25	30.25
TP-16	32.25	31.75	31	31.75	31.25	31	31	31
TP-17	32.5	32	32	31.5	31.25	31.25	31.25	28.75
TP-18	33.25	32.5	32.25	32	30.25	29.25	28.5	28.75
TP-19	32.25	32	32	31.75	30.25	29.5	30.25	29.5
TP-20	33	33.75	32.25	31.75	29.25	28.75	31.25	30.25
TP-21	32.5	32.75	33.25	32.5	31.5	31.75	31.25	31.25

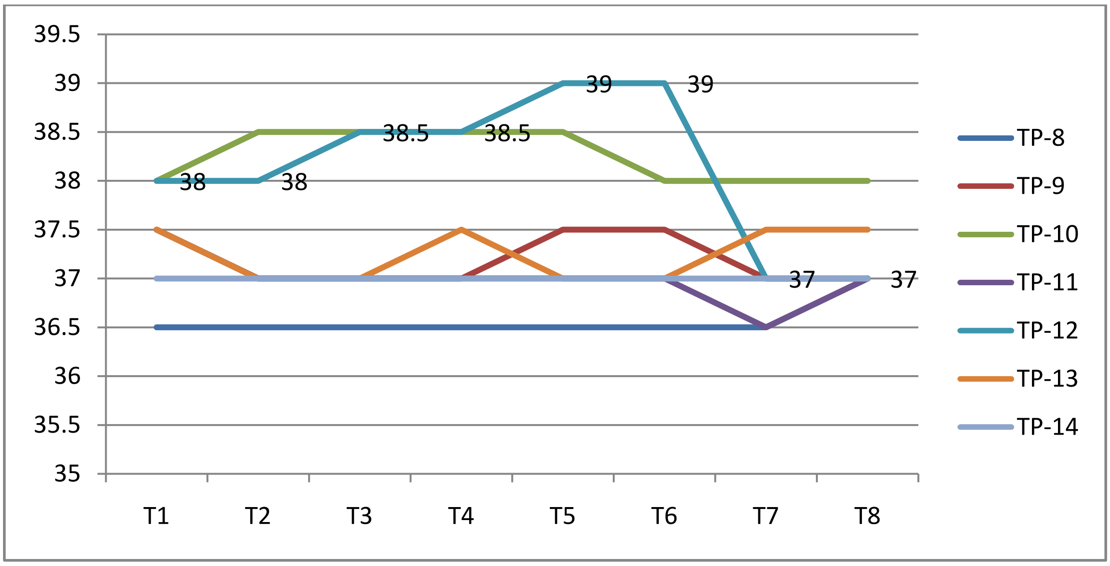


Condition-AC off fan off

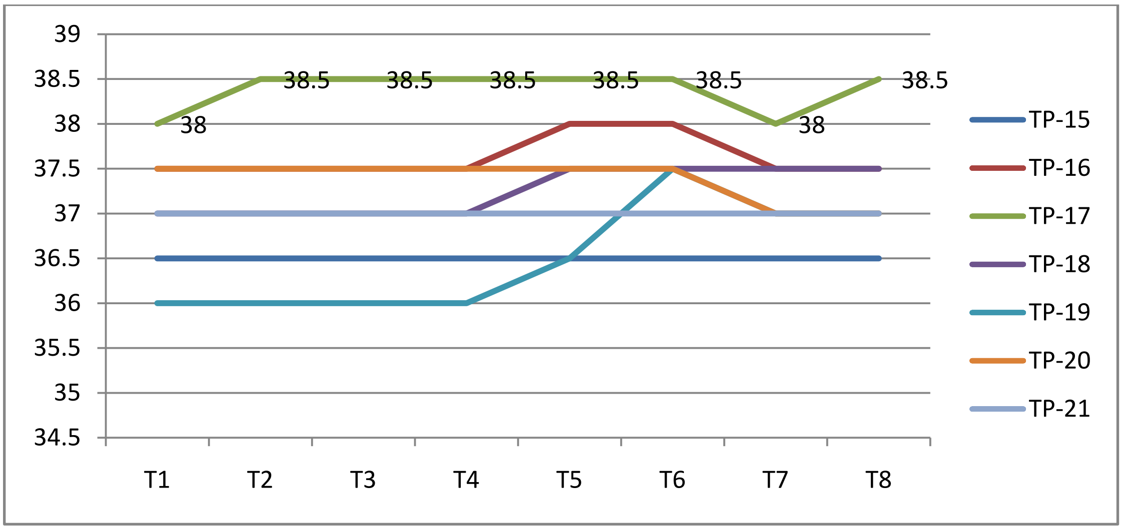
	T1	T2	T3	T4	T5	T6	T7	T8
TP-1	37.5	37.5	37.5	37.5	37.5	37.5	37.5	37.5
TP-2	37	36.5	36.5	36.5	36.5	37.5	36.5	37
TP-3	38.5	38.5	38.5	38.5	39	39	38.5	38.5
TP-4	37	37	37	37	37	37	36.5	37
TP-5	38.5	38.5	38.5	38.5	38.5	38.5	38	38
TP-6	37	36.5	37	37	37	37.5	37	37
TP-7	37.5	37.5	37	37.5	37.5	37.5	37	37



	T1	T2	T3	T4	T5	T6	T7	T8
TP-8	36.5	36.5	36.5	36.5	36.5	36.5	36.5	37
TP-9	37.5	37	37	37	37.5	37.5	37	37
TP-10	38	38.5	38.5	38.5	38.5	38	38	38
TP-11	37.5	37	37	37	37	37	36.5	37
TP-12	38	38	38.5	38.5	39	39	37	37
TP-13	37.5	37	37	37.5	37	37	37.5	37.5
TP-14	37	37	37	37	37	37	37	37

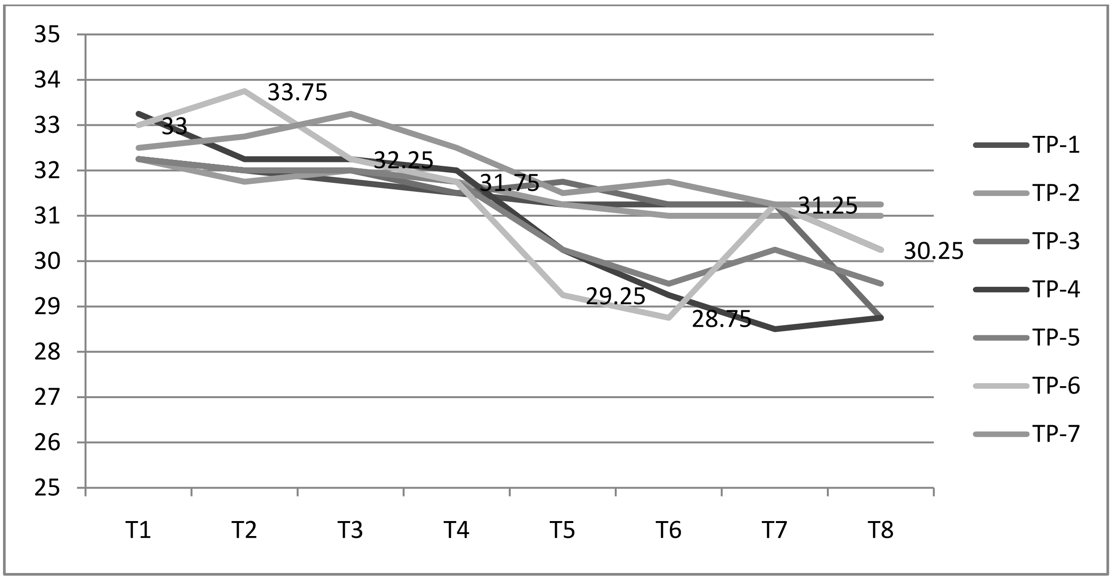


	T1	T2	T3	T4	T5	T6	T7	T8
TP-15	36.5	36.5	36.5	36.5	36.5	36.5	36.5	36.5
TP-16	37.5	37.5	37.5	37.5	38	38	37.5	37.5
TP-17	38	38.5	38.5	38.5	38.5	38.5	38	38.5
TP-18	37	37	37	37	37.5	37.5	37.5	37.5
TP-19	36	36	36	36	36.5	37.5	37	37
TP-20	37.5	37.5	37.5	37.5	37.5	37.5	37	37
TP-21	37	37	37	37	37	37	37	37



Condition-AC on FAN off

	T1	T2	T3	T4	T5	T6	T7	T8
TP-1	32.25	32	31.75	31.5	31.25	31.25	31.25	30.25
TP-2	32.25	31.75	32	31.75	31.25	31	31	31
TP-3	32.25	32	32	31.5	31.75	31.25	31.25	28.75
TP-4	33.25	32.25	32.25	32	30.25	29.25	28.5	28.75
TP-5	32.25	32	32	31.75	30.25	29.5	30.25	29.5
TP-6	33	33.75	32.25	31.75	29.25	28.75	31.25	30.25
TP-7	32.5	32.75	33.25	32.5	31.5	31.75	31.25	31.25



IV. CONCLUSION

In this paper experimentally determine thermal resistances of different types of rolling linear guide ways and to investigate the influence of preload, type of rolling elements and size of guide way on its thermal resistance.

Theoretical part of this work firstly describes basic types of linear guide ways used in machine tool design. Then it concentrates on measuring of temperature, types of thermometers and their principles. It also states basic types of heat transfer and defines quantities that describe it. Deduced to the empirical models of thermal contact conductance and then an example of heat transfer across bolted joints as a typical machine part is described. Last chapters of theoretical part states some experiments for determining thermal resistance and other experiments that deal with thermo-mechanical properties of linear guide ways.

For the determination of thermal resistances of linear guide ways an experiment was proposed and several measurements with different types of guide ways were executed. Results of measurements are stated.

REFERENCES

[1] Xiaozhou Wu*, Yujia Liu, Genglin Liu, Fenghao Wang, Zhihua Wang,” Effect of Supply Air Temperature on Indoor Thermal Comfort in a Room with Radiant Heating and Mechanical Ventilation” International Conference on Improving Residential Energy Efficiency, IREE 2017.

[2] Max H. Sherman and Iain S. Walker “Air Distribution Effectiveness for Different Mechanical Ventilation Systems.” Environmental Energy Technologies Division August 2007

[3] ASTM, “Standard Test Method for Determining Air Change in a Single Zone by Tracer Dilution,” Americn Society for Testing and Materials Standard E-741, 2000.

[4] P. OLE FANGER, " Indoor Air Quality in the 21st Century: Search forExcellence," Indoor Air 2000; 10: 68-73

[5] Lagercrantz, L., Wistrand, M., Willen, U., Wargocki, P., Wittersel1, T. and Sundell, J. (2000) "Negative impact of air pollution on produ?tivity: previo':1s Danish findings :•epeated in new Swe?lSh test room", Paper for Healthy BulldiHgs 2000, Espoo, F inland, August 2000.

[6] Toftum, J., Jmgensen, AS., and Fanger, P.O. (1998) "Upper limits for air humidity to prevent warm respiratory discomfort", Energy and Buildings, 28, 15-23.

[7] Watson Richard and Kirby Chapman. Radiant heating and cooling handbook. McGraw Hill Professional, 2002.

[8] Rongling Lia *, Togo Yoshidomib , Ryozo Ookac , Bjarne W. Olesend, “Case-study of thermo active building systems in Japanese climate”, 6th International Building Physics Conference, IBPC 2015.

[9] Jens Ü. Pfafferott, Sebastian Herkel, Doreen E. Kalz and Andreas Zeuschner, Comparison of low-energy office buildings in summer using different thermal comfort criteria, Energy and Buildings, 39 (2007), p. 750-757.

[10] Wim Zeiler and Gert Boxem, Effects of thermal activated building systems in schools on thermal comfort in winter, Building and Environment, 44 (2009), p. 2308-2317

[11] Gregor P. Henze, Jens Pfafferott, Sebastian Herkel and Clemens Felsmann, Impact of adaptive comfort criteria and heat waves on optimal building thermal mass control, Energy and Buildings, 39 (2007), p. 221-235.

[12] J. Pfafferott. Enhancing the design and operation of passive cooling concepts. Ph.D. dissertation, Fraunhofer IRB Verlag, Stuttgart, Germany, ISBN 3-8167-6626-9, 2004.

[13] prEN 15251:2005. Criteria for the indoor environment. Beuth Verlag, Berlin, 2005.

[14] prEN ISO 7730:2005. Ergonomics of the thermal environment. Beuth Verlag, Berlin, 2005.

[15]] J.E. Braun, T.M. Lawrence, C.J. Klaassen, J.M. House, Demonstration of load shifting and peak load reduction with control of building thermal mass, in: Proceedings of the 2002 ACEEE Conference, 2002.

[16] J. Pfafferott, S. Herkel, M. Ja¨schke, Design of passive cooling by night ventilation: evaluation of a parametric model and building simulation with measurements, Energy and Buildings 35 (11) (2003) 1129–1143.

[17] G.P. Henze, D. Kalz, S. Liu, C. Felsmann, Experimental Analysis of Model-Based Predictive Optimal Control for Active and Passive Building Thermal Storage Inventory, International Journal of HVAC&R Research 11 (2) (2005) 189–214.

[18] E. Halawaa,∗, J. van Hoof b,c, “The adaptive approach to thermal comfort: A critical overview” Energy and Buildings Volume 51, August, 2012, Pages 101-110.