**International Aerospace CFD Conference**
***(Paris, June 18 – 19 2007)***

# Verification of CFD-based Computation of Thermal Comfort Indices

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***ABSTRACT:*** This study investigates the applicability of User-Defined-Functions (UDFs) for determination of thermal comfort indices for non-uniform environments. The demand for such an investigation derived from the need to evaluate thermal environments in aircraft cabins in a more comprehensive way.

It is normal practice by Airbus to determine the initial layout of ventilation systems also on the basis of computational fluid dynamics (CFD). The resulting set ups are subsequently tested in scale 1:1 cabin mock-ups that are basically life size cabin models. When airflow simulations for the A400M cargo hold were performed with CFD, thermal comfort was assessed in accordance with ISO 7730. The assessment of such environments was approached by computing the Predicted Mean Vote (PMV) and the Predicted Percentage of Dissatisfied (PPD) (Fanger, 1970). CFD-measurements in strongly non-uniform environments shall be conducted at several locations, at or around the subject to form average quantities (ISO 7726).

The study compares indices based on CFD computed parameter quantities averaged over an area or a volume of cells near a human model to indices based on human subject votes. The CFD model offered reality equivalent airflow behavior confirmed by comparison with practical experiment data. The human subject votes derived from laboratory experiments conducted by P.O. Fanger.

Surfaces that partly covered the human models were created for measuring the thermal comfort indices as well as single parameter values.

Results with accuracy within +20/-10 percentage points PPD were achieved when no special precautions were taken. Measurements performed at surfaces placed 50 cm from the human models resulted in accuracies within ± 10 percentage points PPD.

A proposed alternative solution, which involves a user-defined function that disregards human model induced radiant heat, yielded results with accuracy within ±5 percentage points PPD off target values for the optimal measuring distance to the human models.

*Keywords: Thermal Comfort, Fanger, Cabin, PPD, PMV*

## CABIN FLOW COMPUTATION AT AIRBUS

The motivation for this study about the applicability of thermal comfort computation for non-uniform environments derived from the need to evaluate thermal environments in aircraft cabins. Environmental control engineers at Airbus employ CFD to determine the initial layout of ventilation systems of new aircraft and also to improve thermal comfort in existing planes. A broad variety of investigations are made in order to find optima for location and airflow velocities of air outlets, insulation properties, temperature distributions, airflow-, cooling- and heating-requirements, control parameters and other properties of interest. In this context, CFD-calculations range from locally limited phenomena up to the modeling of complete full size cabins. Whereas CFD is very helpful for design, it is still indispensable to verify final results by tests.

For thermal comfort considerations, it is until recently common practice to evaluate temperature and velocity of the flow and also wall temperatures in separate plots, values and diagrams. Nevertheless, when airflow simulations for the A400M cargo hold were performed, thermal comfort was also assessed in accordance with ISO 7730. This means that Predicted Mean Vote (PMV) and Predicted Percentage Dissatisfied (PPD) (Fanger, 1970) were also evaluated.

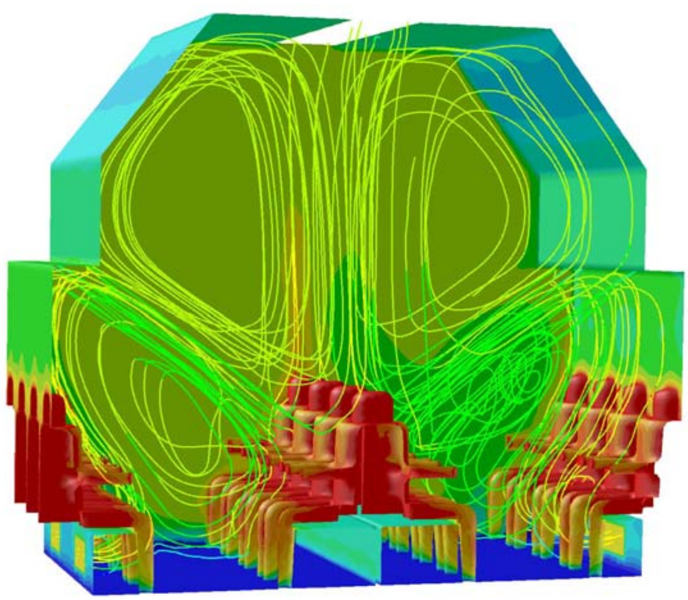

*Fig. 1*: Path lines within the cargo hold of the A400M occupied by passengers.

## PREFACE

The standard ISO 7730 describes calculation of thermal comfort indices based on physical parameters. The quantities need typically only to be measured at a small number of locations that represent the entire occupant's zone.

However, when dealing with non-uniform environments, like in aircraft cabins, there are no single locations, that are representative for the entire cabin. Under such circumstances a number of quantities is needed in order to form average values that represent the zone in question. A human's perceived thermal comfort is impacted by the airflow properties of its surrounding flow field, this region is however, also affected by the human's own presence.

The optimal method for calculation of thermal comfort indices in non-uniform environments needs to be based on the properties of the near-human airflow. Nevertheless, the quantities should be obtained outside the region where the human's own presence induces faulty results.

The overall goal of this study was to establish a recommendation for the use of surface-averaged comfort indices in CFD computational set ups by answering the question: How well do surface- or volume-averaged PMV- and PPD-values correlate to the values determined from the uniform environmental conditions at the boundary of the computational domain?

## OUTLINE OF STUDY

The above-mentioned question was answered by the following procedure:

(1) Compute the flow properties and temperatures of the original laboratory experiments conducted by P.O. Fanger by means of CFD. In order to ensure correct modeling of the flow, an analog benchmark configuration was computed first in order to determine the best CFD parameters.
(2) Determine average values of PMV and PPD over several differently located surfaces within the computational domain. For this purpose, a User-Defined Function (UDF) was employed.
(3) Compare the CFD-computed PMV- and PPD-values averaged over the various surfaces with the human subject votes reported from the laboratory experiments conducted by P.O. Fanger.

## THERMAL COMFORT

Wherever humans occupy artificial climates, as it is very much the case in aircraft cabins, the aim is to ensure not only a healthy but also a pleasant environment. Thermal comfort is in ISO 7730 defined as “The condition of mind which expresses satisfaction with the thermal environment”.

Thermal discomfort will be felt as either an excessively cold or excessively warm sensation of the body. If the body is getting too warm, its internal heat sensors notice it. They tell the thermoregulatory system via the hypothalamus to start the cooling process. Firstly the blood vessels vasodilate (expand) which increases the blood flow through the human’s skin, secondly sweat production on the skin is initiated. When the sweat evaporates it consumes energy, the energy comes from the hot blood running through the skin and subsequently the core of the body is cooled down when the blood returns. Only a few tenths of a degree more than the permitted 37°C core temperature stimulates sweat production, which very effectively provides the necessary heat loss.

If the body is getting too cold i.e. when the skin temperature falls below 34°C, it is registered by the skin-sensors that, also via the hypothalamus, tell the thermoregulatory system to start the body’s defence against cooling down. Firstly the blood vessels vasoconstrict (contract) and thereby reduce the blood flow through the skin to reduce heat loss. Secondly the internal heat production is increased by stimulation of the muscles that start shivering. The defence against cooling down is very effective, even under extreme circumstances where the blood stream to arms and legs can be shut off and shivering becomes very severe.

To be able to determine how far from perfect the environment actually is, the PMV and PPD indices have been introduced. According ISO 7730, the PMV value can be calculated on the basis of the following 6 main parameters: metabolic rate, air velocity, air temperature, humidity, mean radiant temperature and clothing insulation.

The PMV index is the predicted mean rating from a large number of people on a psycho-physiological scale. The scale has 7 different ratings ranging from Cold (-3) to Hot (+3). The PMV index tells if occupants feel too warm or too cold under given circumstances. The information about which percentage of the occupants would be dissatisfied is provided by the PPD, which can be derived from the PMV according ISO 7730. Its dependency is shown in the following *Fig. 2.*

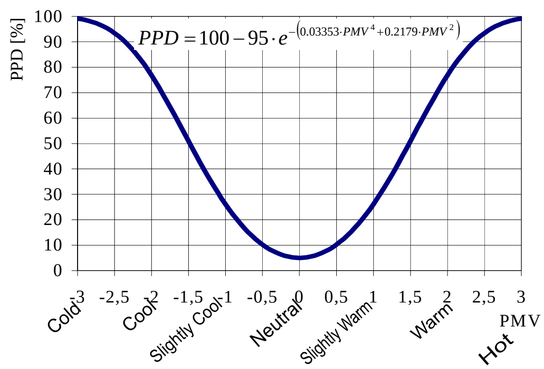


*Fig. 2*: PPD vs. PMV according ISO 7730.

## THE ORIGINAL FANGER EXPERIMENTS

The theory behind the PMV and the PPD was derived on the basis of practical experiments with hundreds of human subjects performing validation of thermal comfort under various thermal conditions, see (Fanger P.O. 1982). For each of the tests, 8 college-age females, 8 college-age males, 8 elderly females and 8 elderly males were subjected to different test conditions for 3 hours in a climate chamber. The chamber dimensions were 5.6m long x 2.8m wide x 2.8m high. During the tests, the air temperature was maintained at 4 different levels: 21.1°C, 23.3°C, 25.6°C and 27.8°C. The mean radiant temperature was kept equal to the air temperature and all tests were conducted at RH = 30% and RH = 70%. The chamber had furthermore been fitted with heated wall panels for controlling the radiant heat load as well as air temperature and humidity control devices.

The subjects were all seated performing light sedentary work at a metabolic rate of 1.0 met and wearing clothes representing a clo-value of 0.6. The climate chamber was ventilated at an air exchange rate of 40 $h^{-1}$ by an equally distributed air stream from the ceiling exiting through slots in the floor periphery.

## CFD-RECALCULATION OF THE FANGER EXPERIMENTS

### *DETERMINATION OF APPROPRIATE CFD-PARAMETERS*

Fluent offers a wide range of options for setting up the boundary conditions and the numerical solver. A benchmark experiment involving air velocity and temperature measurements above a heated cylinder was chosen as reference for the CFD calibration. Just as the Fanger experiments, this is a low-Reynolds-number, natural convection case with identical temperature gradients. The test conducted at the Technical University of Berlin (Streblow, 2006) involved measurement of air velocities and heat distribution above a heated cylindrical body as shown in *Fig. 3*.

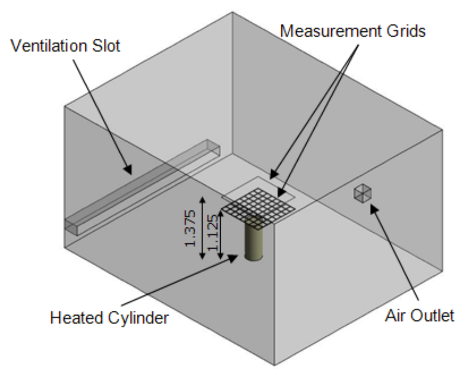


*Fig. 3*: Benchmark test experiment (Streblow, 2006).

Airflow was introduced through the ventilation slot in the lower left wall. The upwards air velocity (natural convection) induced by the cylinder's 100W heat dissipation was measured by means of laser Doppler anemometry at two 880 x 880 mm planes above the cylinder. The heat distribution was recorded with a thermal camera on a surface above and on the cylinder itself. The size of the room as well as the environmental conditions in the test was very similar to the conditions in the climate chamber used by Fanger. Therefore the empirically obtained data from the benchmark test provided an excellent template for setting up the CFD-model to be used in this project. An exact replication of the test facility was created in Gambit and discretised by a circular grid around the heated cylinder, a tetrahedral transitional grid with rectangular outer geometry and a hexahedral grid modeling basically the remaining volume of the test room.

The optimal set up of turbulence models and solver settings was determined by comparison between the theoretically simulated data and the empirically obtained data from the test. As criteria for good agreement between CFD and experiments, the average velocity at the measuring grid was used.

The near-wall region around the heated cylinder was discretised by two different approaches: The “Standard Wall Function approach”, which is suitable for coarse grids and the “Near Wall Model approach”, which requires very fine meshing. For turbulence modeling, Fluent’s RNG k-ε model was used including Enhanced Wall Treatment (EWT) for the “Near Wall Model approach” cases.

Whereas the “Standard Wall Function approach” without boundary resolution showed unsatisfactory results, the “Near Wall Model approach” with resolution of the boundary layer performed very well. For the latter case, a structured resolution of the boundary layer with a growth function and also an unstructured Tgrid-approach were successful. For the unstructured boundary layer resolution, the cylinder face was initially meshed with triangles of much smaller size than the Tetrahedrals within the surrounding volume. This constellation ensured high grid resolution in the immediate vicinity of the cylinder, while the total number of elements was kept at a reasonable level due to the rapid cell growth. All residuals were stable and below $10^{-3}$ and the mass flow imbalance was <1%. The mean velocities through the measuring planes were found stable and within 10% of the empirically measured values.

### *CONCLUSION OF BENCHMARK TEST REPLICATION*

Both meshing methods with boundary resolution performed basically very well. The main problem with the pave scheme grid is however, that it is unsuitable for meshing complex geometries like e.g. a human model. Therefore, the unstructured grid is preferable due to the versatility of the unstructured grid type, which is suitable for meshing complex geometries. It was later used for the replication of the original Fanger experiment together with the successful solver settings and the gained information concerning optimum grid size.

## DETERMINATION OF PMV- AND PPD-VALUES FROM THE CFD-CALCULATIONS

### *GENERAL*

A UDF for instant computation of thermal comfort indices has previously been developed at Airbus. It is based on the UDF ‘fanger.c’ obtained from Fluent. The script extracts the four environmental quantities air velocity, air temperature, radiation temperature and humidity from the computational set up, whereas metabolic rate, external work and insulation of clothing must be manually changed in the script file. It is based on the BASIC-program provided in ISO 7730.

The preconditions for calculation of PMV and PPD are that energy and species transport for water vapor as well as radiation calculations have taken place. For simulation of the radiant temperature the Discrete Ordinates (DO) radiation model was used.

The Virtual Climate Chamber (VCC) was created according to the dimensional properties of the original test facilities provided in the work of Fanger. Human models were placed in a volume within the VCC, which was meshed with a tetrahedral grid. This volume acted as transition between the models' small size surface triangles and the main volume's large tetrahedral grid. As in the original work of Fanger, the human models were placed in a seated position, 2x4 human models, face to face in the middle of the chamber; see *Fig. 4* (ref. Nevins, 1966).

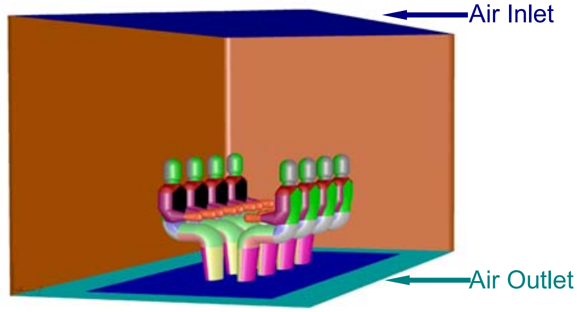


*Fig. 4*: Computational set up of Fanger experiments.

### *VIRTUAL CLIMATE CHAMBER BOUNDARY CONDITIONS*

The VCC walls were defined as solid walls whereas the entire ceiling was defined as a velocity inlet. The floor area was defined as part wall and part pressure outlet. The wall temperatures were set to the desired temperature of each experiment in order to attain a mean radiant temperature equal to the air temperature. The emissivity for all walls was set at a value of 0.97, as in the cylinder experiments, with a diffusion factor, $f_d$ of 1.

According to the work of Fanger the average climate chamber temperature was kept at 21.1°C, 23.3°C, 25.6°C and 27.8°C for the respective experiments. Thus the air inlet temperatures needed to be slightly lower in order to compensate for the human heat emission. The desired temperature for each case was originally achieved by means of a temperature sensor and an inlet air temperature control unit. Due to the unknown amount of heat transferred from the humans to the air before having the CFD-results, the inlet air temperature had to be adjusted manually in an iterative way. It had to be modified until the correct volume average temperature of the VCC was reached.

The inlet air velocity of 0,031 m/s resulted from an air exchange rate of $40h^{-1}$ as in the original experiments. For all cases, a turbulence intensity of 5 % was used. Each set of boundary conditions listed in *Table 1* represents one of the tests conducted by Fanger.

*Table 1:* Air Inlet Boundary Conditions.

| VCC-temperature [°C] | 21,1 | 21,1 | 23,3 | 23,3 | 25,6 | 25,6 | 27,8 | 27,8 |
|---|---|---|---|---|---|---|---|---|
| Relative humidity [%] | 30 | 70 | 30 | 70 | 30 | 70 | 30 | 70 |
| Inlet air temp. [°C] | 20,7 | 20,7 | 22,9 | 22,9 | 25,2 | 25,2 | 27,5 | 27,5 |
| MassFrac. $H_2O$ [g/kg] | 4,9 | 11,4 | 5,6 | 13,1 | 6,5 | 15,1 | 7,4 | 17,2 |

## *HUMAN MODEL CONSTRUCTION*

The creation of the human models was constrained by two important requirements: The models needed to be created with dimensional properties equivalent to real humans in order to ensure realistic flow conditions in the near-occupant regions. Also the surface area and body part proportions needed to be equivalent to those of real human subjects. The dimensional requirement was met by the use of DIN 33402-2, which deals with man's ergonomic dimensions. The surface was split up in segments in order to make definition of the heat dissipation for individual areas possible. The area requirement was met by correlating each model segment to equivalent segments of a thermal manikin for which the surface area has been measured by laser scanning (Sørensen, 2003).

## *HUMAN MODEL BOUNDARY CONDITIONS*

The heat dissipation of the 8 human models was set as a heat flux boundary condition with variable boundary surface temperature.

Wet heat loss components that cause water evaporation were neglected in this study, because they have only minor effect on the surrounding airflow and temperature as they only increase the humidity. Dry heat loss components in the form of dry respiration heat loss, radiation and convection have significant impact on the surroundings causing radiant heat transfer and buoyancy driven airflows. The human models' total heat dissipation was set as function of the surrounding air temperature in accordance with the dry heat dissipation given in (Danvak, 1997 p. 501).

The heat dissipation was split up in a clothed skin part and a bare skin part. 70% of the total heat loss was set to be dissipated over the clothed area of the human model whereas the remaining 30% were dissipated over hands and heads.

*FLOW SIMULATIONS*

As can be seen in *Figure 5*, the heat dissipated from the humans caused considerable natural convection, leading to updrafts in their vicinity. The shape of the toroidal vortex can easily be deducted from the figure.

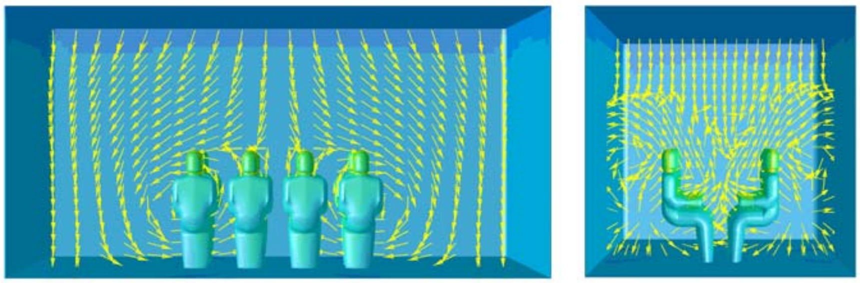

*Fig. 5*: Airflow velocity vectors in midplanes of VCC.

*CFD MEASUREMENTS*

For each row of human models 6 surfaces, A to F, were created. Each surface consisted of three planes named Back, Front and Top. Thus the term "surface", followed by a capitol letter (A-F), refers to the combination of three planes (Back, Top and Front), whereas the term "plane" refers in the following to single planes.

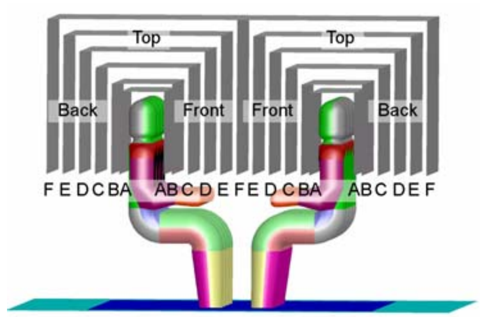


*Fig. 6*: CFD Measuring surfaces.

Surface A, was placed at a distance of 50mm from the human models, surface B at 100mm and the remaining surfaces in increments of 100mm away from the human models. It was decided not to let the surfaces cover the entire body in order to avoid interference between front planes and human models on the opposite side.

### *COMPARISON BETWEEN CFD-DETERMINED AND HUMAN-VOTED PMV- AND PPD-VALUES*

The experiments conducted at identical temperatures all showed the same trends regardless of humidity level. In the following figures, the value for each plane is placed behind the bars labelled "Average", which represent the surface values. A *Target Value*, which indicates the original result or parameter value from Fanger's experiments, has been included in all charts.

*Figures 7* till *10* show results for the 21,3°C / 30%RH case. For all other cases, similar charts were prepared and interpreted during the study.

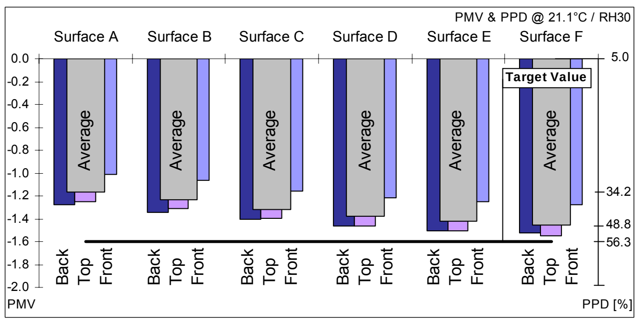


*Fig. 7*: PMV- and PPD-values for the case 21.1°C / 30%RH.

From *Figure 7*, one can read:

(1) PMV is too high for all planes and surfaces.
(2) Planes Front are significantly high.
(3) Average PPD is too low for all planes and surfaces, however surfaces D, E & F are within ±10 of target PPD.
(4) All planes and surfaces approaching target values with increasing distance to human model.

(1)

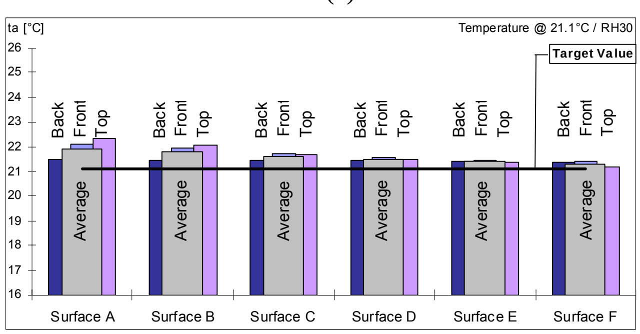


*Fig. 8*: Air temperature for the case 21.1°C / 30%RH.

From *Figure 8*, one can read:

(1) Air temperature is too high for all planes and surfaces.
(2) Planes Top and Front are significantly high.
(3) All plane and surface values approaching target value with increasing distance to human model.

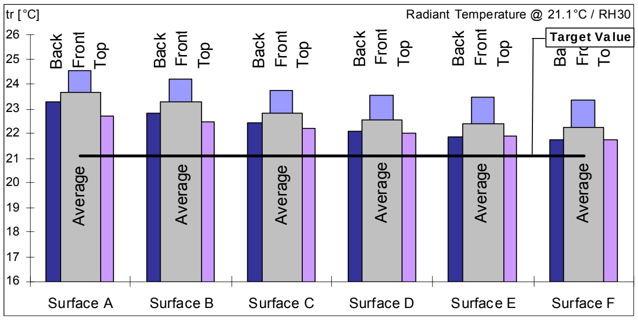


*Fig. 9*: Mean radiant temperature for the case 21.1°C / 30%RH.

From *Figure 9*, one can read:

(1) Mean radiant temperature is too high for all planes and surfaces.
(2) Planes Front are significantly high.
(3) Mean radiant temperature is decreasing with increasing distance to human model.

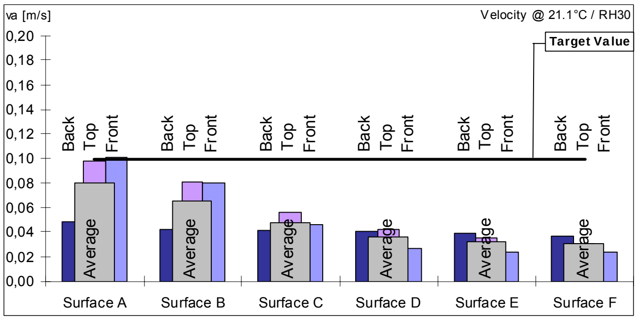


*Fig. 10*: Air velocity for the case 21.1°C / 30%RH.

From *Figure 10*, one can read:

(1) Air velocity is too low for all surfaces.
(2) Planes Top and Front reach target value for surface A
(3) Air velocity decreasing with increasing distance to human model.

## SUMMARY OF RESULT DATA

*Figure 11* shows that the PMV index was interpreted too high for all surfaces. As can be seen in *Figure 12,* this causes too low PPD indices, at temperatures below thermal comfort level and too high indices above.

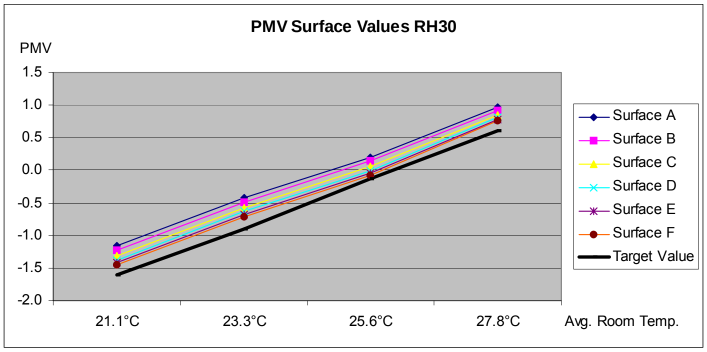


*Fig. 11*: Summary of PMV Surface Results at 30% RH.

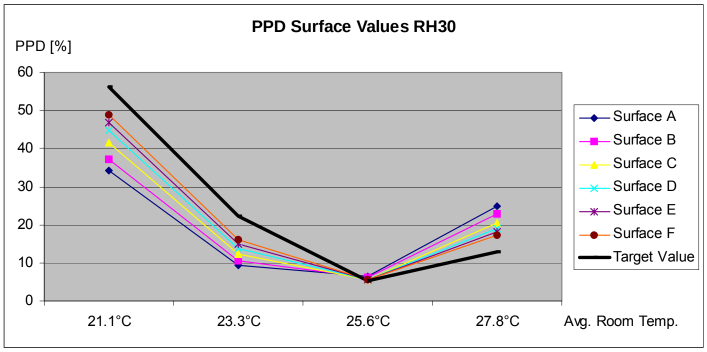


*Fig. 12*: Summary of PPD Surface Results at 30% RH.

## DISCUSSION

For all experiments the computed PMV indices were too high compared to the target values. This resulted in faulty PPD indices that were found up to 20 percentage points off target for the most severe cases. The excessive PMV read-outs were caused by too low air velocities as well as too highly interpreted air- and radiant temperatures. As expected, the air velocity as well as the air and radiant temperatures were found to be decreasing with growing distance to the human models.

The velocity read-outs were found below the target values for all surfaces although the Front and Top planes provided reasonable accuracy in close vicinity of the human models. Air velocity does however, play a minor role in the range between 0.0 m/s and 0.1 m/s where natural convection dominates. The air velocity is furthermore given as an approximate value in the description of the original experiments, which indicates some degree of uncertainty about the accuracy of this parameter.

The deviance in air temperature was primarily caused by excessive temperature values at the Top and Front planes. The excessive heat on those planes was due to natural convection and the higher heat dissipation rates in the head and hands regions. The impact from this source of error could have been minimised by performing measurements around the entire body as this would somewhat compensate for the extra heat dissipation in the head and hands region.

The mean radiant temperature deviated significantly from the target value for all experiments. The deviance amounted to several degrees Celsius above the target values at surfaces near the human models. This was due to excessive mean radiant temperature read-outs at all planes, particularly high values were observed on the Front planes that were placed between the two rows of human models. The planes in this location were subjected to human model induced radiant heat from several directions and only to a small extent subjected to wall radiation. In order to achieve correct mean radiant temperatures, only the wall radiation should be taken into consideration. Allowing only the surrounding boundary surfaces (walls, floor and ceiling) and not the human models to take part in the radiation measurements would reduce the main source of error considerably when calculating surface averaged thermal comfort indices.

Inserting the results at the CFD measuring surfaces for temperature, air velocity and humidity together with a mean radiant temperature equal to the wall temperature into the equations for PMV and PPD from ISO 7730 improved the accuracy of the PPD indices significantly. This procedure, which is almost equivalent to using the previously mentioned user-defined function, yielded maximum errors within 10 percentage points off target PPD.

Humidity was found close to the target values at all planes/surfaces for all experiments and thus causes only minor error.

## CONCLUSION

Thermal comfort indices can be successfully computed as surface averaged values by means of CFD. The accuracy was found to be within +10/-20 percentage points PPD for all measurement surfaces without special precautions. The 3 main sources of error were:

(1) Low velocities
(2) High temperatures
(3) Excessively high radiant temperatures

The velocity read-outs only reached the target value when measured very close to the human models and decreased with growing distance. This parameter does on the other hand, only have minor impact on the PMV at air velocities as low as 0.1 m/s.

All experiments show that the human model heat dissipation causes elevated temperatures on the measuring surfaces. The heat impact decreases with growing distance however, this also results in changed airflow conditions. No conclusive optimal surface-to-human model distance can be determined as low distances result in excessive heat read-outs, whereas large distance measurements may not represent the near-human flow conditions.

The measuring distance will inevitably pose a trade-off between elevated heat measurements and altered flow conditions and should therefore be individually considered for the flow field in question. For flow fields similar to those considered in this project, a surface distance of 50 cm can be recommended.

A proposed alternative method, which disregards the human model induced radiant heat, indicated considerable accuracy improvements. The human model induced radiation heat should be disregarded on the measuring surfaces.

Estimation calculations have shown that accuracies in the region of ±10 percentage points are achievable by using this method. All results for Surface F were furthermore less than 5 percentage points off the target values when disregarding human model induced radiation heat. This order of accuracy is considered acceptable when it is taken into consideration that the surface averaged thermal comfort indices are only to be used for initial ventilation layouts and comparison between various ventilation configurations.

## REFERENCES

Fanger P. O. (1982) *Thermal Comfort – Analysis and Applications in Environmental Engineering*, Malabar, Florida: Robert E. Krieger Publishing Company.

Danvak ApS (1997) *Varme og Klimateknik, Grundbog 2. udgave 1. oplag*, Bonde's og Dyva Grafisk A/S, København.

Nevins R. G. et al (1966) *Temperature-Humidity Chart for Thermal Comfort of Seated Persons*. In: Ashrae Trans. 72. I: 283 – 291.

Sørensen D. N. and Voigt L. K. (2003) *Modeling flow and heat transfer around a seated human body by computational fluid dynamics*. In: Building and Environment 38 pp. 753-762.

Streblow R. (2006) *Route 06 V&V CFD for Special Cabins,* Technical University of Berlin, Institute for Energy Engineering.