Bypass flow influence on energy performance of data center cooling

J. Novotný, J. Nožička, J. Matěcha

***Abstract*—**The study aims to describe the influence of bypass on cooling of the data center. In this study, both numerical and experimental methods were combined in order to describe an undesirable effect of the bypass flow; a simplified analytical procedure was also used. The paper first describes basic metrics used when defining main parameters of data center cooling. The following part describes experimental data center where individual experiments were performed. For the needs of CFD simulations, suitable boundary conditions were defined in order to get as close as possible to the real flow inside the data center, based on measurements taken. Finally, a simplified analytical solution was used. On the basis of the analytical solution, detailed information on various components of the simulated data center are applied, the information being obtained during measurement. All three approaches are compared at the end and the influence of missing blanking plugs on the overall energy performance of data center cooling is described with the help of the performed analysis

***Keywords*—** Bypass data center, cooling, thermal management, PIV, CFD.

I. Introduction

With increasing demands of users of data-processing programs, the requirements for performance of used computing stations are growing as well. With increasing performance of particular IT equipment installed in data centers, the requirement for cooling performance of data centers is growing. Since every year, there is a significant increase in computing power installed in the data centers, the improving the energy performance of the cooling process of the data centers is one of the few ways the data centers manufacturer and provider may reduce the total power consumption. Currently, there are a number of ways to minimize the energy performance of the data center. [1],[2],[3],[4],[5] and others present an overview of comprehensive studies on the subject. The aim of these works is to create conclusive studies describing the impact of particular layout of the data centers on required installed cooling performance. For the purpose of comparing different variants, different metrics are introduced, enabling simple comparison of the efficiency of the cooling system in different data centers. The total flow in the data center can be simply illustrated by the diagram in figure1. Based on the diagram above, it is possible do define several basic metrics [6],[7],[8].

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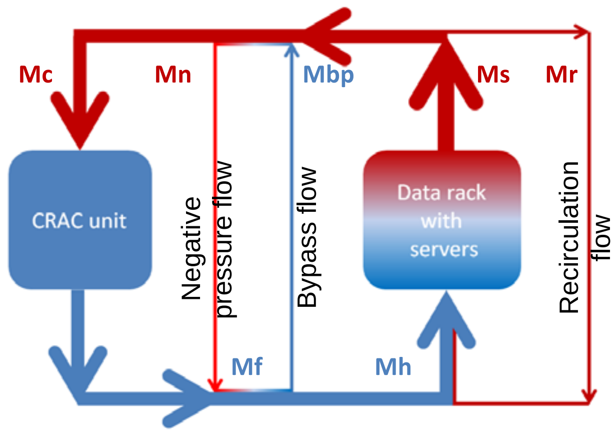

Figure 1. Data center air distribution. CRAC unit-computer room air conditioning unit.

CRAC flow (Mc): is total mass flow in chiller.
Server flow (Ms): is cool air used for cooling the servers.
Bypass air flow (Mbp): is cooled air passing through or outside the data rack without being used for cooling the servers, subsequently it is mixed with ”heated" air and enters the CRAC unit.
Recirculation air flow (Mr): is air used for cooling the servers, it is mixed with cooled air and repeatedly used for cooling the servers.
Negative pressure flow (Mn): is air that is sucked repeatedly due to unfavorable pressure conditions in data center (Venturi effect).
As shown in figure 1, three equations for mass flows in three "nodes" of circulation can be written down:

$$Mf = Mn + Mc \qquad (1)$$

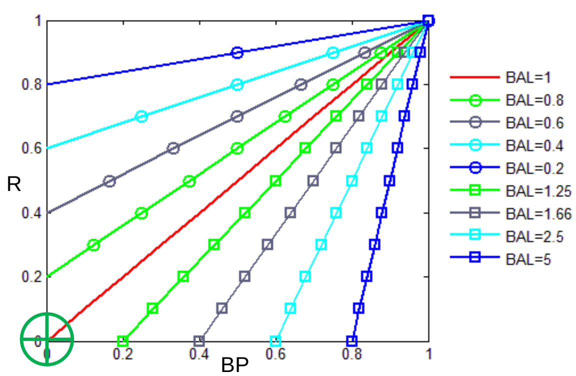

Figure 2. Data center air distribution in air management metrics

$$Mh = Mf\text{-}Mb \tag{2}$$

$$Ms = Mh+Mr \tag{3}$$

Based on these several simple parameters describing inside the data center, the following formulas can be determined:

ByPass ratio **Bp**:

$$BP = Mbp/Mc \tag{4}$$

Recirculation ratio**R**:

$$R = Mr/Mc \tag{5}$$

Balance ratio**BAL**:

$$BAL = Mc/Ms \tag{6}$$

With the use of the above formulas, a final Balance ratio including also the effect of negative pressure gradient according to [3] can be introduced:

$$BAL(1+Np)=Mf/Ms \tag{7}$$

$$BAL(1+Np)=(1\text{-}R)/(1\text{-}BP) \tag{8}$$

The resultant dependence is shown in figure. 2. From the equations above, it is evident that if the data center is without undesired recirculations and Bypass flow, the resulting state of the flow field at undesired zero gradient can be identified in the diagram in figure. 2 in the lower left corner. It is evident that if the amount of the cooled air flowing outside the "main loop" increases, the costs of data center cooling grow disproportionately. In diagram in figure. 2, we are moving from the lower left corner to the upper right one. In case that all the cooled air is flowing through the "main loop", this critical condition of the data center can be identified in figure 2 in the other three corners depending on the values of parameters Np, R and BP. Besides the above mentioned metrics, there are still a number of other parameters enabling to compare particular solutions of data centers [7],[8]. One of the most commonly used metrics is the Power Usage Effectiveness (PUE) or its reciprocal, the Data center infrastructure Efficiency (DCiE).

$$\mathrm{PUE} = \frac{\text{Annual averange delivered todata center}}{\text{Annual averange IT equipment power use}} \tag{9}$$

Another highly used metrics is the Return Temperature Index (RTI), which can be defined as:

$$RTI = \left[1 - \frac{(\text{Treturn} - \text{Tsupply})}{\text{dTequpiment}}\right]{*}100 \tag{10}$$

With the recirculation, the RTI value is higher than 100%, while the bypass flow decreases the value. The presented study is based on CFD flow simulation inside the data rack with a pair of chillers located on the data rack. The presented work is a partial study of the impact of bypass flow on the overall process of cooling of the data center, based on experiments and measurements that have been carried out during the development of cooling of data centers with extended possibilities of using the free-cooling. The usage of model with two subsequent chillers is advisable in order to extend the possibilities of using the free-cooling system even in cases where no sufficient cooling performance for dissipation of all heat with the help of free-cooling is available. In such a case, the other subsequent chiller for additional heat dissipation to the cooled air with the help of classic compressor cooling[5] is used.

II. EXPERIMENT SET UP

The aim of this study was to assess the adverse effect of the bypass flow on cooling of the data center in configuration that enables extended use of free-cooling. The proposed and tested solution involves the use of a pair of subsequent chillers located on data racks along the entire length of the aisle. In order to overcome the chillers resistance, fans are placed behind the chillers. Since this arrangement significantly increases the chillers' surface, it is possible to use axial fans with low pressure drop in order to overcome the required pressure drop. A complete separation of cold and hot aisle is part of the proposed solution. Experimental location was fitted with 10 pieces of heat simulators figure 3, which were designed on the basis of ideological similarity with real server Supermicro 1026TT-IBQF figure 4. The measurement of the mass flow rates and distribution of velocities at the outlet of the server were implemented with the help of the stereo PIV method. For stereo PIV measurement the following standard system with two hi-sensitive cameras and pulsed laser has been used.

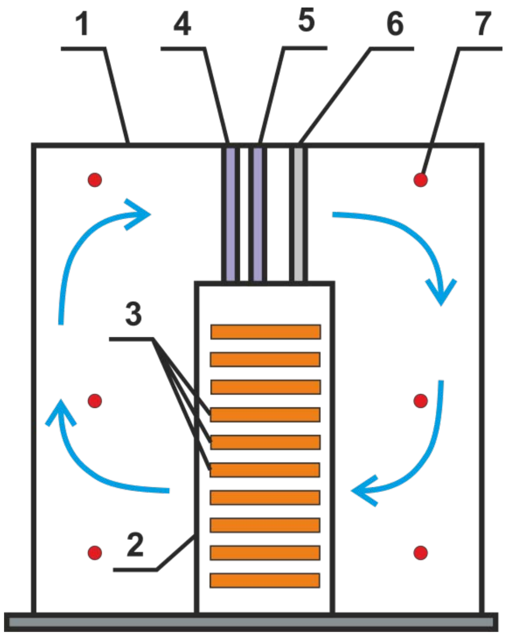

Figure 3. Real experiment simulating data center management, which was subsequently used as a model for numerical simulations. 1-Cold aisle (separated),2-27U data rack, 3-Server models,4- High-temperature chiller, 5-Low-temperature chiller, 6 -Group of six axial fans, 7-Hot aisle (separated)

Used Stereo PIV system:

- 4M flow sense camera
- 220 mJ Nd:YAG Laser, double cavity
- Data analysis:Dynamics studio software
- Matlab - Final data processing

The temperature field at the outlet of the servers was measured by a set of thermocouples. Temperatures of individual simulated components were controlled by thermocouples again. Our model is very narrow since it is a model of 1U server, thus the outlet velocity profile is rectangular in shape. When taking measurements using the stereo PIV method, we proceeded from the requirement of mapping the velocity field at the outlet of the server. For these reasons, the whole flow field had to be divided into five parts in those subregions and it was necessary to measure the mean velocity field and to compute the total velocity field by integrating these results together. The mutual shift between different areas was 90mm and the area of the overlap was a minimum of 10%. The division of the outlet flow field into particular subregions is shown in figure 5.The measurement of the velocity field was carried out for simulator individual measured 3D velocity fields are shown in figure 6 and temperature field in figure 7. Final results of experiment for fully isolated hot and cool isle are in table 1 and 2. The simulated data rack was 27U and the arrangement of models of servers was carried out in agreement with the experiment. The study comprises a comparison of the effect of missing 2U blending plate and also the effect of switched-off server. The results of the numerical study were subsequently used to verify the suitability of the used numerical model for later simulations of larger units.

Figure 4. Heat simulator - model of server manufactured on the basis of ideological similarity with real server Supermicro 1026TT-IBQF.

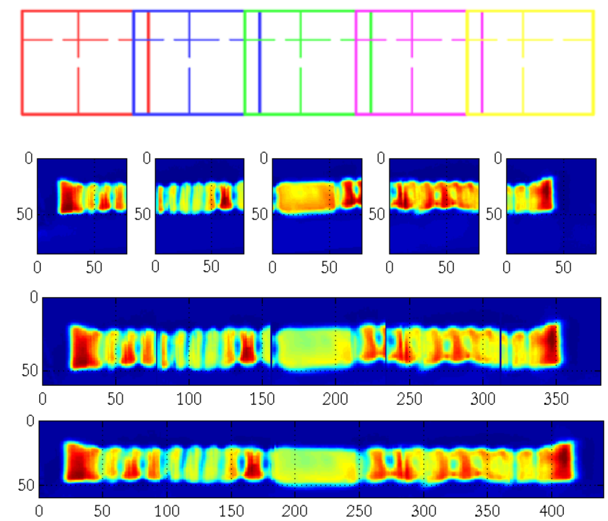

Figure 5. Division of velocity field at the outlet into five subregions, in which measurement using the stereo PIV method was subsequently carried out- first line. Raw result of five stereo PIV measurement-second line. Complete three dimensional velocity profile-third line. Final three dimensional profile.

III. CFD SIMULATION

When designing the numerical model, a maximum emphasis was put on the greatest similarity with the experimental setup. Within the simulation, each of ten heat simulators including a pair of exchangers and six fans providing sufficient pressure

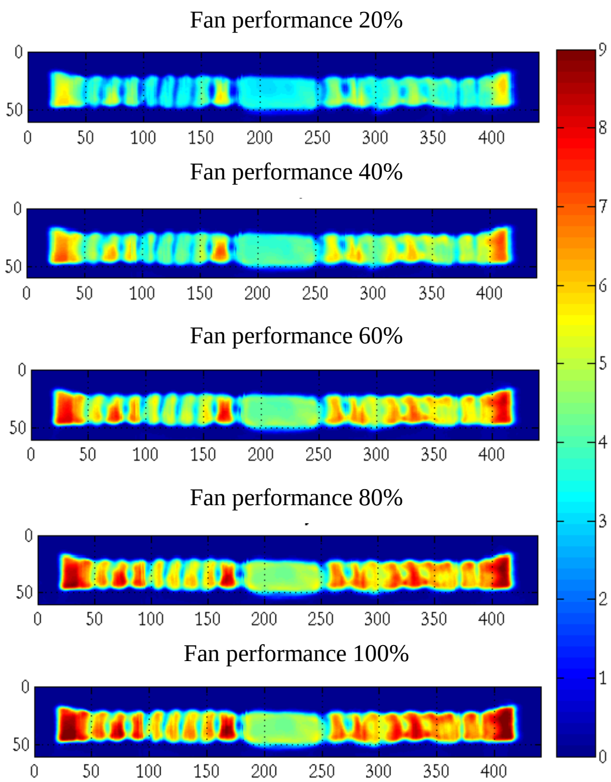

Figure 6. Outflow velocity from heat simulator

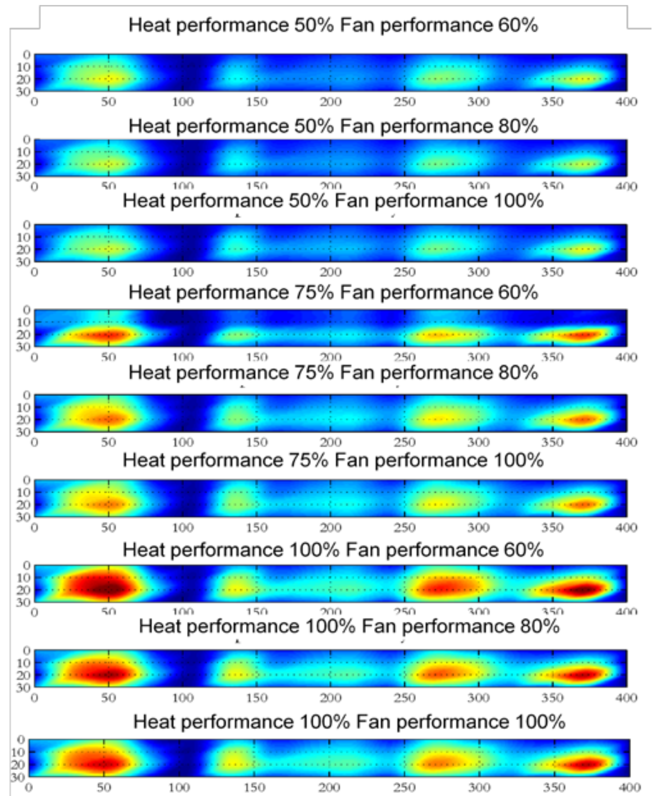

Figure 7 Temperature field at the outlet of heat simulator.

High-temperature chiller		
T_{inlet}	T_{outlet}	Q
25,6°C	28,2 °C	2,7 kW
Low-temperature chiller		
T_{inlet}	T_{outlet}	Q
15,1°C	17,9°C	3,03 kW
Air temperature		
Cool isle	Hot isle	
20,4 °C	38,3°C	

Tab 1 Result of experimental simulation. Total heats flux 5.7 kW, 47% of heat transfer on first high temperature chiller, no Bypass flow.

High-temperature chiller		
T_{inlet}	T_{outlet}	Q
25,6°C	28,2 °C	2,7 kW
Low-temperature chiller		
T_{inlet}	T_{outlet}	Q
15,1°C	17,9°C	3,03 kW
Air temperature		
Cool isle	Hot isle	
20,4 °C	38,3°C	

Tab 2 Result of experimental simulation. Total heats flux 5.7 kW, 47% of heat transfer on first high temperature chiller, Bypass flow missing blending plate 2U.

drop was modeled. An overall view of the simulated volume with indicated boundary conditions is shown in figure 9.

Detail of numerical mesh with missing blending plate. is described in figure 10. A detailed view of the model of heat simulator, which was considered at first approximation, is shown in figure 11. This model contained, compared to the real heat simulators, only a pair of fans (compared to six). The simplification then had a negative impact on the resultant temperature and velocity distribution at the outlet of the data rack. For these reasons, it was necessary to supplement the model with a total number of fans with respect to their location inside the server model. A modified numerical model of server is shown in figure 10. All boundary conditions were defined based on previous measurements with respect to the best agreement with the experiment. All numerical models were implemented with the boundary conditions according the result experiment.

- Chiller: Radiator boundary condition, Temperature 291 K and 298, K loss factor 150 and heat transfer coefficient 800$Wm^{-2}K^{-1}$.

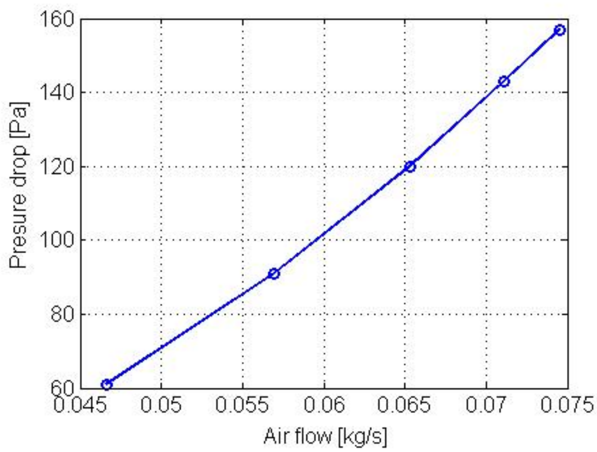

Figure 8. Relationship between flow Rate and pressure drop of heat simulator

- Fans: Fan boundary condition, Pressure drop 180 Pa, maximum velocity reduced to 20 ms^{-1}
- Heat source: Radiator boundary condition, Temperature 335 K, loss factor 5 and heat transfer coefficient 1440 $Wm^{-2}K^{-1}$.
- Fans inside server: Fan boundary condition -Pressure drop 100 Pa, maximum velocity reduced to 20 ms^{-1}.

The calculation was performed with the help of the Fluent program. Within the project, two-equation model K-epsilon Realizable was used. Ideal-gas equation was used for density of the flowing air. K epsilon model has been selected because is appropriate for large, open space [2],[8],[9]. Within the project, gravitational acceleration was considered. All meshes were after creation converted with the help of the Fluent program into polyhedrons. The calculation was terminated when the value of the mass flows in the data rack was constant. The shape of the heat simulators was subsequently used in other simulations, in which both the influence of the switched-off server (first from bottom) and the influence of absence of 2U gasket located symmetrically in the middle were being observed.

IV. Results of numerical simulations. Inappropriate simplification of model of heat simulator

As mentioned at the beginning of this chapter, the first numerical simulation was carried out for a simplified shape of the heat simulator. Two fans, instead of six, were modeled figure 11. This simplification, however, resulted in an inappropriate shape of both velocity and temperature field. Given the above, it was necessary to create a new figure 12, more accurate model of heat simulator with six fans; this model was additionally supplemented by a narrowing at the inlet, which is in most servers caused by the presence of discs and DVD drives. This new, more accurate model was subsequently used for all resultant numerical simulations without change. The new, more accurate model does not indicate undesirable whirling in the area between fan boundary condition and radiator boundary condition, which causes undesirable overheating of the outlet air not observed in the real simulator. On the contrary, the new more accurate model does not indicate such behavior and the flow through this area is rather calmed. Outlet temperatures from heat simulators do not match the measured data. On the other hand, the use of the new modified model enables a more uniform temperature distribution and matches better the measured data. The temperature difference at the outlet of servers for the first

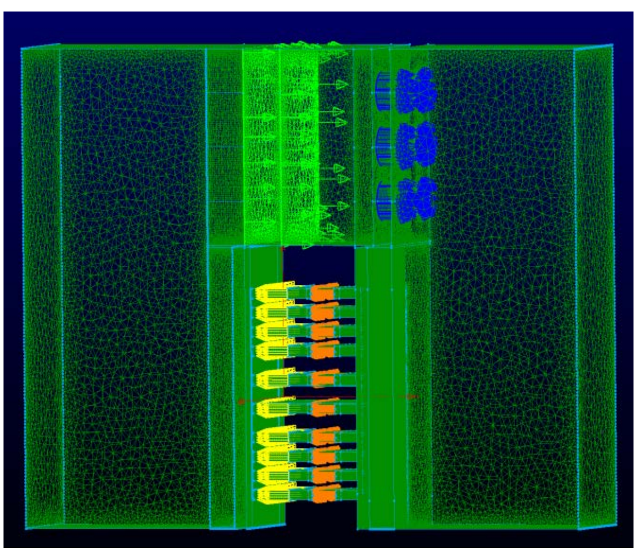

Figure 9. View of simulated volume. Numerical network with boundary conditions. "Chillers" are marked green, "fans" blue, "fans inside the heat simulators" orange, surfaces simulating heat of processors are marked yellow

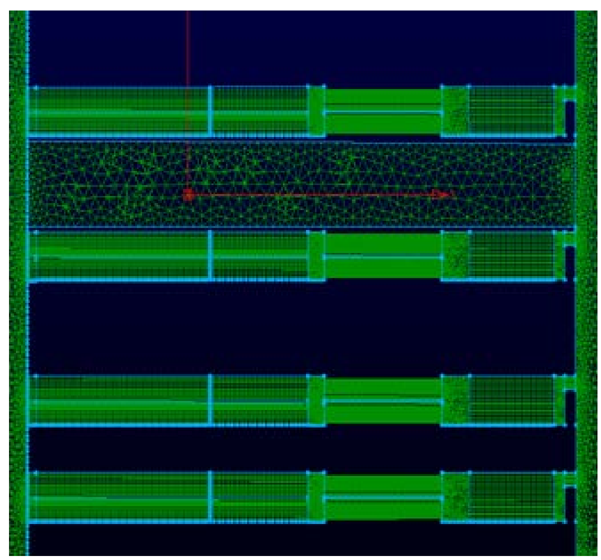

Figure 10. Detail of numerical mesh with missing blending plate.

solution of heat simulators with a pair of fans is more than six degrees. With new solution with six fans, the variance of

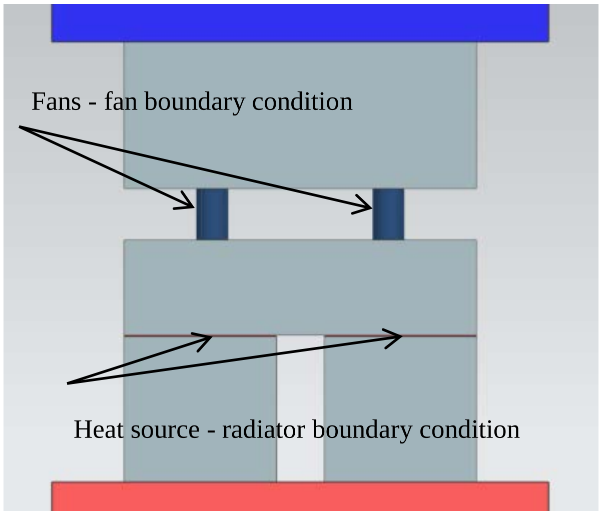

Figure 11. View of heat simulator. The simplified model with a pair of fans.

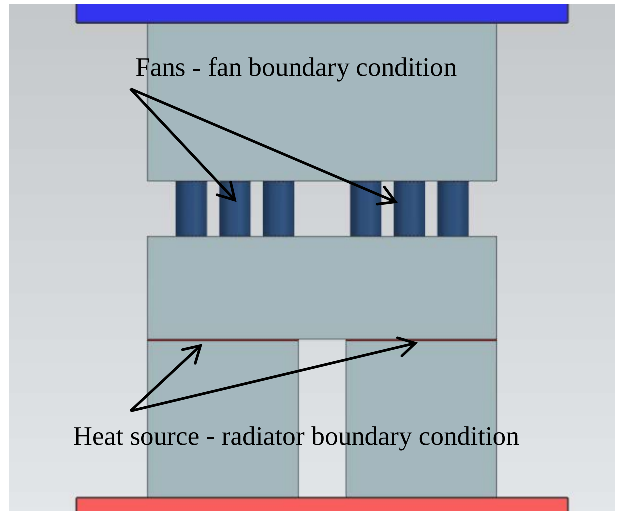

Figure 12. View of heat simulator. The modified model with six fans.

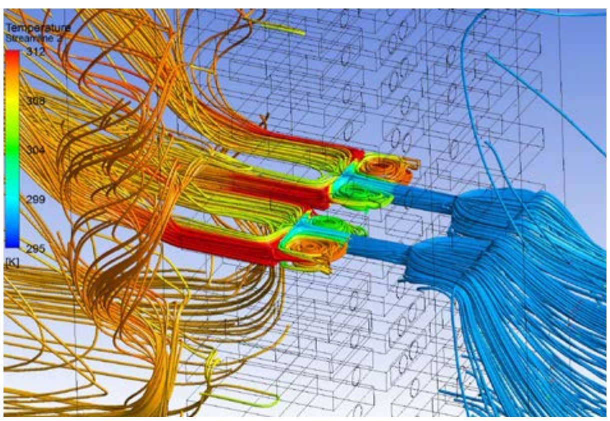

Figure 13. Detail of flow field in s simplified model of heat simulator with just two fans.

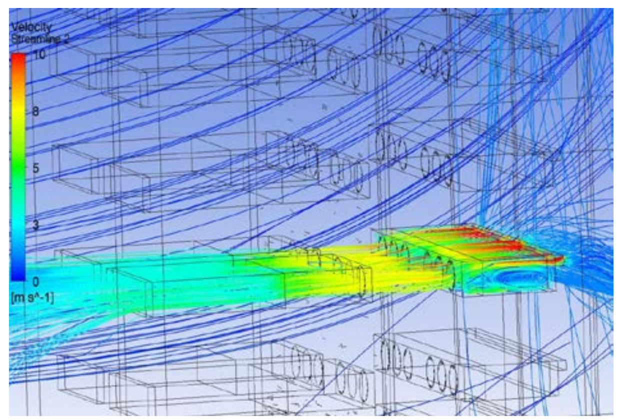

Figure 14. Detail of flow field in modified model of heat simulator with six fans.

temperature at the outlet of the "server" is less than one degree. The temperature and velocity field at the outlet of servers for inappropriately and new modeled heat simulator is shown in figure 13 and 14. The increased temperature at the walls is caused by the viral structure arising during modeling of only one fan in each half of the server. On the edges, an intensive overheating can be observed and the area of the air outlet of servers indicates very different velocities. Temperature field at the outlet of servers for new model of the heat simulator is shown in figure 15.

V. Analytical Solution

When determining the undesirable effect of bypass it is possible to apply the analogy with the piping systems on condition that hot and cold aisles are completely isolated. The missing plugs then create another "channel" connecting the hot and cold aisle figure. 16). The mass flow rate in this "channel" is proportional to the pressure difference between the cold and hot aisle. In order to avoid negative pressure flow it is necessary to maintain a slight overpressure in the hot aisle area usually in the range of 4-20 Pa. Our study assumes that servers work in a design mode (overpressure at the inlet is 5 Pa) and the pressure drop in the hot aisle zone caused by missing plugs can be compensated only with fans fitted in the CRAC unit. The mass flow rate in the missing plug can then be determined according to the following equation:

$$\Delta P = \left(1 + \xi_{bp}\right)\rho \frac{1}{2} v_{BP}^{2} \qquad (11)$$

where:
ΔP is the pressure difference between cold and hot aisle
ξ_{BP} is the coefficient of friction losses for additional "channel" resulting from the missing plug
v_{BP} is the mean flow velocity
ρ is air density

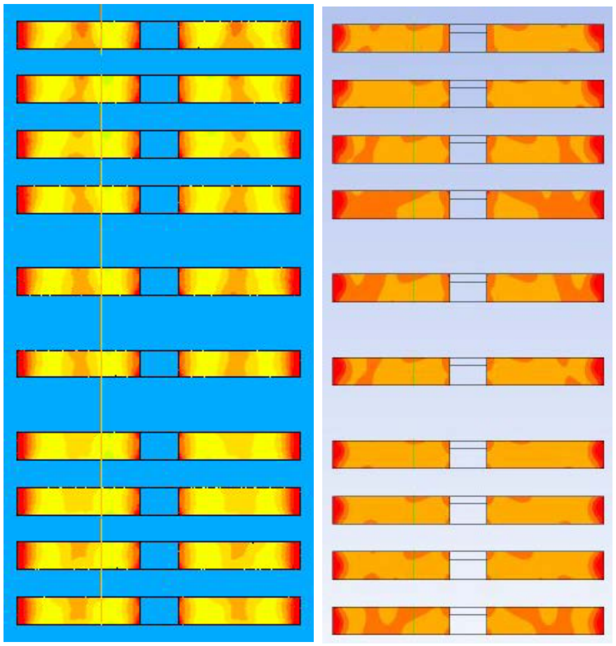

Figure .15 Temperature field in plane of outlet of servers, first model of heat simulator in the left, new model in the right.

If we assume maintaining the total mass flow rate in individual servers, it is also necessary to maintain the value of overpressure in the cold aisle and thus increase the performance of fans in the CRAC unit. Total flow in the CRAC unit as a result of the above mentioned would be:

$$\dot{m}_{CRAC} = \dot{m}_{server} + \dot{m}_{BP} \qquad (12)$$

where:

$\dot{m}_{CRAC}$ is the mass flow rate in CRAC unit
$\dot{m}_{server}$ is the mass flow rate in all servers
$\dot{m}_{BP}$ is the mass flow rate in bypass

In order to achieve such increase of mass air flow rate in the CRAC unit, it is necessary to increase the performance of fans and thus the value of overpressure in front of the CRAC unit coolers. The required value of overpressure in front of the coolers can be determined based on the following equation:

$$\Delta P = \Delta P_{fan} + (1 + \xi_{CRAC})\rho\frac{1}{2}v_{CRAC}^2 \qquad (13)$$

where:

ΔP_{fan} is the pressure difference generated by fans of CRAC unit
ξ_{CRAC} is the coefficient of local losses for CRAC unit coolers
v_{CRAC} is the mean flow velocity in CRAC unit

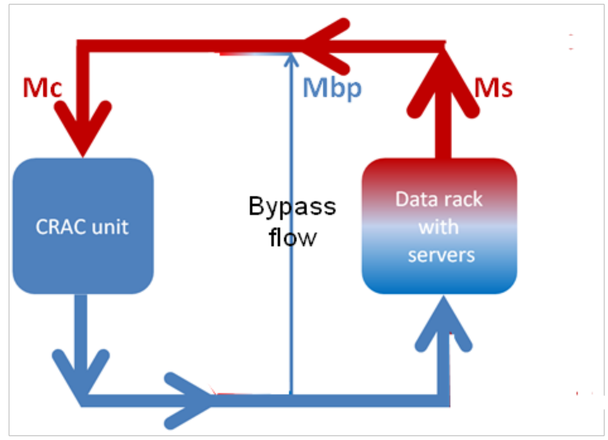

Figure 16 Simplified flow in the data center

Assuming that we consider the compression of fans to be the ideal adiabatic process, it is possible to determine technical work needed for the required increase of the mass flow rate:

$$w_t = P_{CRAC} * \frac{1}{\rho} \qquad (14)$$

Coefficient of local losses for the channel formed as a result of the missing plug can be determined depending on the ratio of channel height and width [11]. Since we measured the dependence of the mass flow rate on the pressure drop with the help of model server, we are capable of determining the coefficient of local losses of that thermal simulator. The manufacturer indicates similar dependence for individual heat exchangers and thereby such simplified analytical solution can be used. The results of such a simplified approach are shown in the following diagrams. figure 17 shows the dependence of ratio of mass flow rate in bypass and servers on size of the missing plug and value of overpressure in the cold aisle. The diagram clearly shows the increase of the mass flow rate in bypass, which can, in extreme cases, reach even more than 70% depending on the size of the missing plug and value of overpressure in the cold aisle. figure 18 shows the corresponding increase of technical work consumed by fans. Since the missing plugs cause a significant increase of the mass flow rate in the CRAC unit, the increase of technical work, which is proportional to the cube of velocity, is substantial and can reach more than four times the original value.

VI. Comparison of Results of CFD Simulation

The following images display velocity fields inside the data rack during simulation of two-stage cooling with extended possibility of using the free-cooling with no bypass flow figure 19 and with bypass flow figure 20. The use of free-cooling is simulated here with the help of two exchangers, where the first, down-stream exchanger, has a coolant temperature of 25°C and the other 18°C. The results of numerical simulations help us to determine the following

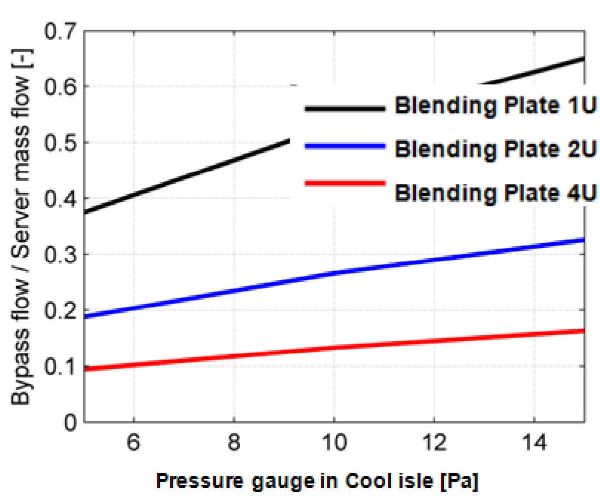

Figure 17 Dependence of increasing mass flow rate in baypass on size of missing blending plate

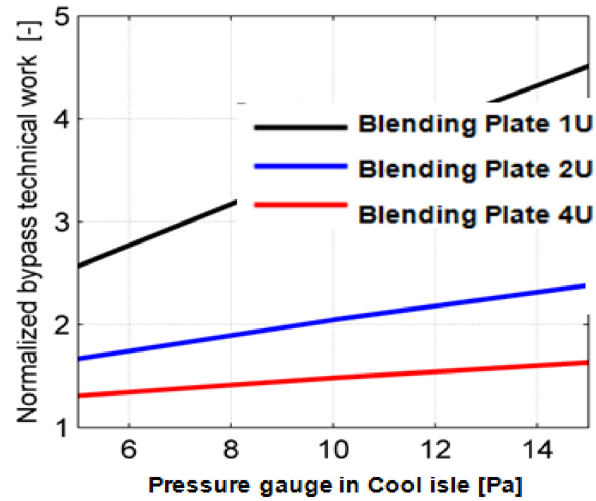

Figure 18 Dependence of increasing technical work on size of missing blending plate

values: Mass flow in heat simulators is 0.594 kgs^{-1}, temperature at the inlet to the first chiller is 311 K and temperature at the inlet to the servers is 298 K. Total heat flux is 6.7 kW. In this compilation, the first exchanger dissipates 48% of heat. The temperature field at the inlet to the exchangers, as well as at the inlet to the heat simulators, indicates uniform distribution with the variance of temperature less than one degree Celsius. The resultant temperature and velocity fields indicate that heat simulators in the upper half of the data rack are in this compilation worse supplied due to a big wake. Considering the above, it is necessary to supplement the proposed solution with a solution, which properly directs the air flow and minimizes large area of recirculation in the upper right corner and the resultant wake in upper half of the data rack. In the simulation of influence of the switched-off server, a zero drop with "fans" in the first server from bottom was defined. This mode was simulated because of the need to identify undesirable effect of the switched-off IT equipment located inside the data rack. The data center provider inno way influences whether the IT equipment installed inside the data racks are on or switched-off (the space in data rack is rented), but he must bear the costs relating to the undesirable effects. The results of numerical simulations help us determine the following values: Mass flow in heat simulators is 0.55 kgs^{-1}, mass flow in switched-off heat simulator is 0.029 kgs^{-1}, total mass flow in chiller is 0.58 kgs^{-1}, temperature at the inlet to first chiller is 310 K and temperature at the inlet to servers is 298 K. Total heat flux is 5.6 kW. In this compilation, the first exchanger dissipates 41% of heat. The temperature field at the inlet to the exchangers indicates uniform distribution with the variance of temperature less than one degree Celsius. The temperature field at the outlet to the heat simulators indicates, due to the switched-off heat simulator, large nonuniformity leading to significant heat transfer in the area behind the outlet of the servers, thus reducing the inlet temperature to the chiller by about one degree. The resultant temperature and velocity fields indicate that heat simulators in the upper half of the data rack are in this compilation worse supplied due to the wake and thus less overpressure at the inlet. The results clearly show an undesirable effect of the switched-off server, even only one, in the data rack. The mass flow in the switched-off server is, in our case, 5% of total mass flow in the data rack; accordingly, the temperature gradient in the chiller will be reduced. The undesirable flow in the switched-off server results in an increase of the mass flow in the whole system, which leads to an increase of the pressure loss (by 10%). This pressure increase causes, together with an undesirable increase of the mass flow, an increase of consumed electricity expended on air distribution by a minimum of 10.5%. This increase of electricity consumption respects only the increase of the mass flow in the system; if we try to compensate for the reduction of temperature at the inlet to the chiller and the decrease of the chiller cooling performance resulting therefrom, we have two options: To reduce the temperature of the coolant, thus preserving the same temperature gradient in the chiller. This solution is immediately reflected in the increased consumption of the chiller. The other option, which is used in the data center management as the first, is to increase the mass flow in the system.

The increase of the mass flow in the system however results in another increase of pressure loss of the flowing air in the IT equipment. If the temperature drops at the inlet to the IT equipment due to the missing blending plates by 5% (as in our simulation), it is necessary to increase the mass flow in the system by about 5%. The resulting increase of the mass flow in the system is then 10%. With this increase of the mass flow, the resultant pressure loss shall increase by about 21% and the consumption of electricity required for the performance of fans increases by the resultant 33% compared to the original consumption.

In the simulation of the use of two-stage cooling, the influence of the missing 2U blending plate inside the data rack was also simulated. The data center providers primarily seek to provide a sufficient amount of cooled air necessary for cooling the IT

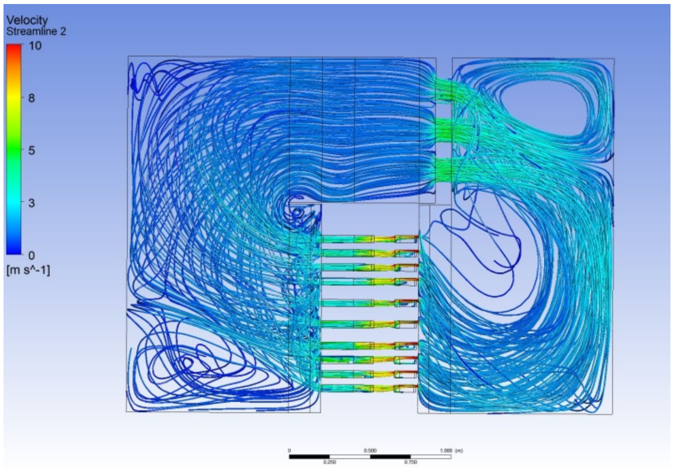

Figure 19. Completely close hot and cool isle. Streamline in whole volume of model case with two chillers placed above data rack, color of streamline indicates air velocity.

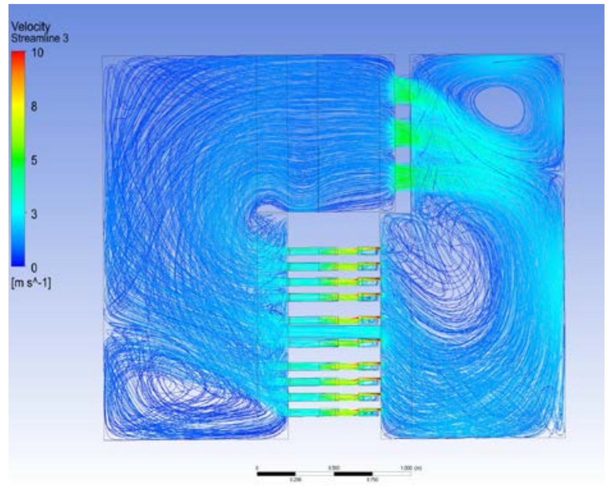

Figure 20. Missing 2U blending plate. Streamline in whole volume of model case with two chillers placed above data rack, color of streamline indicates air velocity.

units. However, the prevention of undesirable effect of bypasses and recirculation is equally important. The results of numerical simulations help us determine the following values: Total mass flow in chiller is 0.632 kgs^{-1}, mass flow in heat simulators is 0.544 kgs$^{-1,}$ mass flow in 2U blending plate is 0.089 kgs-1, temperature at the inlet to first chiller ranges from 307 K to 310 K and temperature at the inlet to servers is 298 K. Total heat flux is 6.4 kW. In this compilation, the first exchanger dissipates 45% of heat. The temperature field at the inlet to the exchangers indicates uniform distribution with the variance of temperature less than one degree Celsius. The temperature field at the outlet of servers indicates uneven distribution with the variance of temperature more than ten degrees Celsius. The results clearly show an undesirable effect of the missing blending plate. The mass flow in the 2U blending plate in the data rack connecting the cold and hot aisle amounts to, in our case, 16% of the total mass flow in the servers. Proportionally to the mass flow passing through the data rack without being used, the temperature gradient in the chiller will be reduced, thus increasing the need to either reduce the temperature of the coolant, or to increase the air flow in the chiller. Undesirable flow in the 2U blending plate results in an increase of the mass flow in the whole system, only increasing the pressure loss (in our case by 34 %). This pressure increase causes, together with undesirable increase of

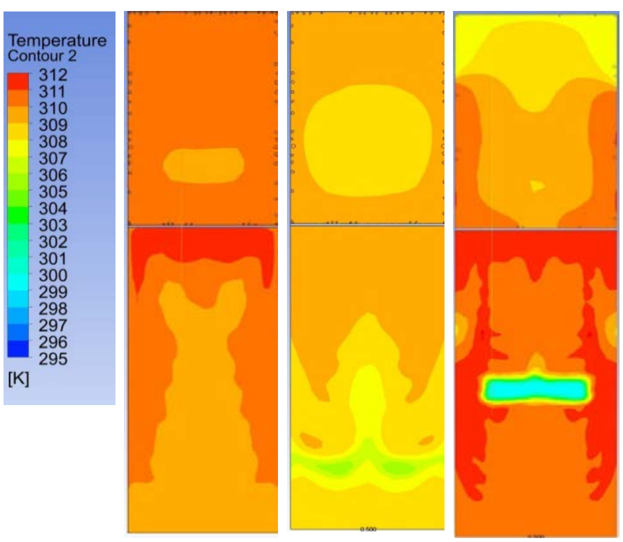

Figure. 21. Temperature field at the outlet of servers (lower part) and temperature field at the inlet to the chiller (upper part) for each tested variants. From the left: completely separated cold and hot aisle, switched-off bottom server, missing blending plate inside the data rack.

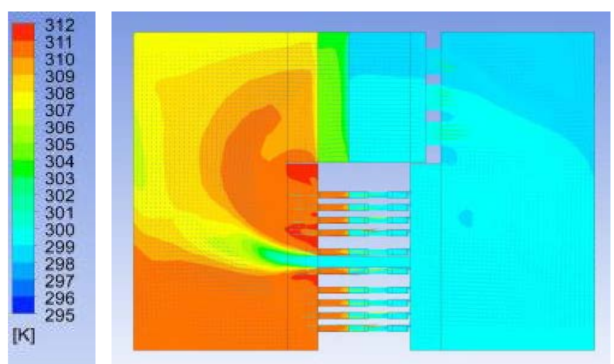

Figure 22. Field of temperature in cool and hot isle. Effect of missing 2U blending plate

the mass flow, an increase of consumed electricity expended on air distribution by a minimum of 56%. This increase of electricity consumption respects only the increase of the mass flow in the system; if we try to compensate for the reduction of temperature at the inlet to the chiller and the decrease of the chiller cooling performance resulting there from, we have two options: To reduce the temperature of the coolant, thus preserving the same temperature gradient in the chiller. This solution is immediately reflected in the increased consumption of the chiller. The other option, which is used in the data center management as the first, is to increase the mass flow in the system. The increase of the mass flow in the system however results in another increase of pressure loss of the flowing air in the IT equipment. If the temperature drops at the inlet to the IT equipment due to the missing blending plates by 16% (as in our simulation), it is necessary to increase the mass flow in the system by a minimum of 16%. The resulting increase of the mass flow in the system is then 34%. With this increase of the mass flow, the resultant pressure loss shall increase by about 79% and the consumption of electricity required for the performance of fans increases by the resultant 140 % compared to the original consumption. figure 21 shows the temperature field at the outlet of the servers (lower part) and temperature field at the inlet to the chiller (upper part) for each tested variants. This picture clearly shows an evident temperature drop at the inlet to the chiller, caused by mixing the hot air exiting from servers and cold air passing through the data rack without being used for cooling the servers (bypass flow) figure 22. Differences between completely isolated cool and hot isle and missing blending plate are described in figure 23 and 24.

VII. Conclusion

Within the conducted studies, an adverse effect of the switched-off IT equipment and the missing blending plate was proved. This adverse effect is reflected in the data center consumption in two ways. The mass flow passing in the switched-off IT equipment (in our case in the server model) or through the 2U blending plate, passes through the data rack completely unused and it mixes in the hot aisle with hot air flowing through the IT equipment. The result is the reduction in temperature at the inlet to the chiller, which is directly proportional to the ratio of given mass flows. The resultant temperature reduction at the inlet to the chiller must be subsequently compensated. There are two ways of compensation: One of them is the temperature reduction of the coolant inside the chiller, thus preserving the original temperature gradient. This approach increases the costs of the compressor cooling in the ratio proportionally to how much the temperature difference in the chiller drops. The other option is to increase the cooled air flow rate in the data rack. This solution again requires the increase of the flow in the data rack proportionally to the temperature drop at the inlet to the chiller. However, this approach increases the pressure drop generated by the flowing air and subsequently imposes higher requirements on the performance of fans. This way of solving the problem brings an increase in operating costs of fans, proportionally to the cube of velocity (if another bypass flow increase is neglected). If 10% of the cooled air (not used for cooling the IT equipment) flows to the hot aisle due to the missing blending plate or due to the switched-off IT equipment, it is necessary to increase the flow rate in the data rack by a minimum of 10%. The resulting flow rate will therefore be by more than 20% bigger than it would have been necessary in case of the properly fitted data racks. The total increase in electricity consumption is then proportionate to the cube of the ratio of velocity with respect to the original, thus leading to 77% of the original fans consumption. The simulations above indicate that the increase in energy consumption of the fans in case the server is switched off is about 25% and in case of the missing 2U blending plate is about 100%. It follows that fitting the data rack with blending plates is extremely necessary for proper operation of the data center and significantly reduces the operating costs; similarly, it seems dangerous to operate the data rack with a switched-off IT equipment, through which only air passes.

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