INFLUENCES OF FRESH AIR SUPPLY LOCATIONS ON TEMPERATURE DISTRIBUTION OF A FAN FILTER UNIT (FFU) TYPE UNIDIRECTIONAL CLEANROOM

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ABSTRACT

Some semiconductor manufacturing processes such as photolithography is very sensitive to the temperature distribution in the cleanroom. Ideally, a uniform temperature distribution is required. This study investigates the effect of three different fresh air supply locations – beneath the ceiling of the supply air plenum (hereafter referred to as case A), at the return air shaft (case B, which uses a plate under the fresh air inlet to boost the mixing of fresh air and recirculation air), and under the raised floor (case C), on field velocity and temperature distribution. The research was conducted numerically using an airflow program based on a standard k-ε turbulence model. The fan curve was added to the fan boundary condition to determine the system’s operational conditions. The results show that the fresh air most greatly affects cleanroom temperature distribution in case A; it is not a good option from the point of view of space management in case C, and case B is a compromise option.

1. INTRODUCTION

To match the essential extremely stringent environmental conditions of semiconductor manufacturing processes, unidirectional cleanrooms are often used to provide acceptable room cleanliness as well as closely controlled room temperature and relative humidity. The FFU type of unidirectional cleanroom has been adopted of late due to its simplicity, ease of fabrication and installation, and its flexibility for upgrading or expanding it’s cleanliness level and size. However, this type of clean room needs special care in handling the supply of fresh air to the room. This fresh air may have a low temperature of only slightly higher than the room dew point temperature (DP), for example a cleanroom with dry bulb temperature (DB) of 23°C and relative humidity (RH) of 43%, where the corresponding dew point temperature (DP) is 9.5°C. In such a case, if fresh air is not thoroughly mixed with recirculation air, then the cleanroom temperature will not be uniform. The make-up air unit (MAU) system takes air directly from the outside, and has sufficient capacity to purify (with filters, absorbers, washers) and control humidity to provide basic latent heat control throughout the fabrication area. Fig. 1a schematically depicts a typical MAU and a clean room. The dry coils in the cleanroom are responsible for the control of the sensible heat produced from the facility and the manufacturing tools.

In semiconductor manufacturing processes such as the photolithography, both the airborne particle cleanliness level and the air temperature distribution are critical. Therefore, a uniform cleanroom temperature distribution is required. The variation of supply air temperature (Tsa) is directly linked to the cleanroom temperature distribution. However, controlling of the supply air temperature is a complex process that involves several factors including temperature variation of the fresh air (Tfa), volumes of supply air (SA) and fresh air (FA), cooling capacity of the dry coils, and of course the spatial distribution of room heat sources produced by the process tools. The temperature variation of the fresh air is a function of the design parameters of the make-up air unit (MAU) and its cooling water circulation system. Both the required cleanliness level and the cooling load determine the supply air volume. The volume of the fresh air is generally based on the volume of exhaust air, which is a function of the extent of room pressurization. Effective and even delivery of the fresh air to the clean room system is also related to the cleanroom’s architectural design parameters such as the dimensions of the cleanroom system and the location of fresh air intake. In general, three locations are adopted for the fresh air supply - under the ceiling of supply air plenum (SAP), at the return air shaft (RAF), and under the raised floor, as shown in Figs. 1b to 1d, respectively. These options are compared here using a standard cleanroom module with a uniform, equitable base. The standard cleanroom module is 4.8 m wide by 17.15 m long by 7.2 m high. This
standard cleanroom module is intended to represent one segment of a large ballroom space that consists of similar modules set side by side and mirrored about a central aisle.

The airflow distribution in unidirectional cleanrooms has received considerable interest, such as in the studies of Maeda et al. [1], Takahashi and Okada [2], Sajadi and Liu [3], Nishioka and Xie [4], Hu et al. [5], Cheng etc. [6], and recently Hu et al. [7]. Concerning very tight cleanroom temperature control, Takahashi et al. [8] reported a unique cooling water circulation system that can control cleanroom temperature within a very tight temperature variation. No data have been published to compare the airflow and temperature distribution of the three different fresh air intake designs. Therefore, this study aims to clarify the advantages and shortcomings of the three designs and to examine how the architectural design of the cleanroom affects the distribution of fresh air.

2. NUMERICAL METHODS

2.1 Airflow Model

The flow is assumed to be turbulent. The turbulent model used in this study is the standard k-ε turbulence model [9]. With this eddy-viscosity turbulence model, the airflow transport can be described by the following time-averaged Navier-Stokes equations (equation 1):

$$\text{div} \left( \rho \mathbf{V} \Phi - \Gamma_{\phi,\text{eff}} \text{grad} \Phi \right) = S_{\phi}$$  \hspace{1cm} (1)

Where $\rho$ is the air density (kg m$^{-3}$); $\Gamma_{\phi,\text{eff}}$ is the effective diffusion coefficient (N m$^{-2}$ s$^{-1}$); $\mathbf{V}$ is the air velocity vector (m s$^{-1}$); $S_{\phi}$ is the source term of the general fluid property, and $\phi$ can be any one of $u$, $v$, $w$, $T$, $k$, and $\varepsilon$, where $u$, $v$, $w$ are velocity components in three directions (m s$^{-1}$), $T$ is the temperature ($^\circ$C), $k$ is the turbulence kinetic energy (m$^2$ s$^{-2}$), $\varepsilon$ is the dissipation rate of turbulence kinetic energy (m$^3$ s$^{-3}$). When $\phi = 1$, the general equation becomes the continuity equation. The governing equation (equation 1) was solved by using the finite volume method SIMPLE (Semi-Implicit Method for Pressure Linked Equation) algorithm that was described by Patankar [10]. The space discretization methods for the various equations are depicted in Table 1. Staggered grids were used so that velocity components were located at the faces of each control volume.

The three dimensional grid system (X, Y, Z) of 44 × 24 × 26 grids for cases A and B, and 52 × 24 × 30 grids for C, which accommodated both memory size limitations and tolerable numerical diffusion were chosen. The line-by-line method was used to obtain converged solutions iteratively. Relaxation factors and false time steps were employed to promote the stability of the process (see Table 1). The iterations were terminated when all the absolute residuals were less than 5 × 10$^{-3}$. The velocity field needed about 6,000 iterations to satisfy the convergence criterion. No slip boundary conditions at the walls were used. The standard log-law wall function of Launder and Spalding [9] was used for the next-to-surface grid points. The CFD package (PHOENICS) developed by Rosten was employed for air distribution calculations.

2.2 Boundary Conditions

The fan performance curve (Fig. 2, with probable range of operation: 10 ~ 20 m$^3$ min$^{-1}$ and 280 ~ 220 Pa) was incorporated in the simulation model. Each analysis run was connected with the fan curve to determine the system’s operating condition, including the system static pressure and flow rate. The porous media model was used to describe the pressure drops across dry coils, perforated plates, and ULPA (Ultra Low Penetration Air) filters. The porous media model was implemented with the CFD model by adding a momentum source term to the standard fluid flow equations. The volume flow rate was assumed to be 1 (m$^3$ min$^{-1}$)/unit m$^2$ of cleanroom floor area, which is the value used in practical design. The fresh air inlet areas were 1.44 m$^2$, 0.25 m$^2$, and 1.44 m$^2$ for cases A, B and C, respectively. Consequently, the fresh air inlet velocities were 0.8 m s$^{-1}$, 4.6 m s$^{-1}$, and 0.8 m s$^{-1}$ for cases A, B, and C, respectively (see Fig. 1). The exhaust air volume flow rate was determined by the mass balance of the system with an exhaust air area of 1.44 m$^2$ located just on the floor of return air plenum (RAP). The heat flux at the roof of SAP and cleanroom ceiling were assumed to have the same value of 30 W m$^{-2}$. Heat fluxes at the cleanroom ceiling include the heat flux from lighting and from fan heat load. Heat flux generated by process tools were assumed to be uniformly distributed on the floor with heat flux of 250 W m$^{-2}$. Heat fluxes removed by the dry coils were assumed to be 300 W m$^{-2}$ of cleanroom area, that is 20,736 W (= 300 W m$^{-2}$ × 4.8 m × 14.4 m). The temperature of the fresh air is assumed to be 14 $^\circ$C. Notably the latent heat, heat flux through the walls, and thermal radiation effect were neglected in this study, because they were much smaller than the above-mentioned.
Fig. 1: Systems of fresh air intakes for the standard cleanroom module

(a) schematic diagram of a MAU and a cleanroom

(b) fresh air introduced under the ceiling of supply air plenum (case A)

(c) fresh air introduced at the return air shaft (case B)

(d) fresh air introduced under the raised floor (case C)
Table 1: The discretization methods

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<td>Turbulence dissipation rate</td>
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![Graph of Q(m³/min) vs Ps(mmAq) in Fig. 2: Fan curves for FFU](image)

(Note: 1mmAq = 9.8 Pa)

3. RESULTS AND DISCUSSION

The standard cleanroom module studied has a dimension of 4.8 m wide (Z direction), 17.15 m long (X direction) and 7.2 m high (Y direction). This standard cleanroom module represents one segment of a large ballroom space that consists of similar modules set side by side and mirrored about a central aisle.

3.1 Velocity Vector and Temperature Distribution in the Middle Section

Figs. 3 to 5 show the velocity vectors and temperature distributions in the middle section of cases A to C, respectively. This section was selected because it shows the prevalent variability of each case. In general, the velocity vectors in Figs. 3a and 5a show parallel streamlines throughout the entire flow field (weak mixing between adjacent streamlines). This important characteristic of a FFU type unidirectional cleanroom makes fresh air difficult to mix with recirculation air, resulting in a non-uniform temperature distribution. The differences of velocity vectors in cases A to C affect the temperature distribution in cleanroom greatly. Case B (Fig. 4a) presents a more thorough mixing of fresh air with recirculation air at the RAF than those of cases A and C (Figs. 3a and 5a). Specifically, the velocity vectors in the SAP of case A are vertical and those of cases B and C are horizontal. At the RAF, the velocity vectors of case B have larger values than those of cases A and C.

In case A, the temperature decreases downstream of the supply air plenum, with poor mixing of the fresh air and the recirculation air at the locations far away from the RAF. In contrast, in cases B and C, the mixing occurs before the air entering the SAP resulting in increasing temperatures downstream of the SAP.

Temperature of case A is generally lower than those of cases B and C (Figs. 4b and 5b). However, cases A and C exhibit a vertical temperature variation, while case B presents a horizontal variation. The horizontal variation is preferred to the vertical one with respect to the temperature uniformity. Taking into account all the temperature points in the section, the average temperature of the cases A, B, and C were 22.3°C, 22.4°C, and 23.1°C, respectively. The standard deviation of all the temperature points were 0.95°C, 0.64°C, and 0.78°C, respectively. Cases B and C present the lower standard deviation temperature value than that of case A, demonstrating that the temperature distribution in cases B and C is more uniform than that in case A. Therefore, cases B and C involve safer methods for introducing the fresh air without the risk of producing a non-uniform temperature distribution. However, in practice the RAP is normally located with various utilities such as exhaust air duct, gas piping, local scrubber, VESAD (very early smoke detect active), and others. Thus, locating the fresh air intake duct under the raised floor (case C) is not a good option from the point of view of space management. Case A involves the lowest average temperature, implying that the influence of fresh air on cleanroom temperature distribution is greatest in...
This case. Therefore, the required cooling capacity of the cooling coils can be reduced in this case with an associated cost saving. Case B is a compromise option (with the second low average temperature and no inferior influence on space utilization), which utilizes a plate under the fresh air inlet to boost the mixing of the fresh air and the recirculation air.

3.2 Temperature Distribution in the Horizontal Section

The temperature distribution in the horizontal section helps one to get more in-depth understanding of the three-dimensional behavior of the study. The horizontal temperature distributions (5 cm under the filter) of the three cases are depicted in Figs. 6a to 6c. This temperature can be regarded as the supply air temperature to the cleanroom. In case B, temperature is generally lower than those of cases A and C, especially in the upstream of the SAP. In case A, a low temperature area appears in the far downstream of the SAP, where the recirculation air is not mixed with fresh air.

The diffusion radius of temperature of fresh air in case B is apparently larger than those of cases A and C, mainly because the strong mixing at the RAF. Taking into account all the temperature points in this horizontal section, the average temperature of the three cases were 23.9°C, 22.4°C, and 24.1°C, respectively. The standard deviation of all the temperature points in this horizontal section of cases A to C are 1.14°C, 0.68°C, and 0.87°C, respectively. Case B reveals the lowest standard deviation temperature value, demonstrating that the temperature distribution in case B is most uniform in the cases studied.

Fig. 3: Velocity vectors and temperature distribution in the middle section for case A
Fig. 4: Velocity vectors and temperature distribution in the middle section for case B

Fig. 5: Velocity vectors and temperature distribution in the middle section for case C
Fig. 6: Temperature distribution (°C) in the horizontal section (5 cm under the filter)
4. CONCLUSIONS

This study compares the airflow and temperature distribution of a unidirectional FFU type cleanroom with three different fresh air intake designs. The results and discussion of the study yield the following conclusions:

- From the point of view of temperature control, introducing the fresh air either at the return air shaft (RAF) or under the raised floor produces horizontal temperature variations which are better than vertical temperature variations produced by introducing the fresh air under the ceiling of the SAP.
- Locating the fresh air intake duct under the raised floor (case C) is not a good option with respect to space utilization. Introducing the fresh air under the ceiling of the SAP (case A), however, produces a lowest average room temperature, implying that the cooling capacity of the cooling coils can be reduced. Introducing the fresh air at the return air shaft (case B) is a compromise option.

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NOMENCLATURE

$S_\phi$ source term of the general fluid property
$T_{fa}$ fresh air temperature, °C
$T_{sa}$ supply air temperature, °C
$V$ air velocity vector, m$^{-1}$
$\Gamma_{\phi,eff}$ effective diffusion coefficient, N.sm$^{-2}$
$P_s$ static pressure of the FFU
$P_e$ electricity consumption of the FFU
$X$ longitudinal direction, m
$Y$ upward direction, m
$Z$ transverse direction, m
$k$ turbulence kinetic energy, m$^{2}$.s$^{-2}$
$u$ velocity components in X directions, m$^{-1}$
$v$ velocity components in Y directions, m$^{-1}$
$w$ velocity components in Z directions, m$^{-1}$
$\varepsilon$ dissipation rate of turbulence kinetic energy, m$^{2}$.s$^{-3}$
$\phi$ can be any one of $V,u,v,w,T,k,$ and $\varepsilon$
$\rho$ air density, kgm$^{-3}$

REFERENCES