Effect of the tubular-fan drum shapes on the performance of cleaning head module

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Abstract. The geometrical effects of a tubular-fan drum on the performance improvement of the cleaning head module of a vacuum cleaner were investigated. In this study, the number of blades and the width of the blade were selected as the design parameters. Static pressure, eccentric vortex, turbulence kinetic energy (TKE) and suction efficiency were analysed and tabulated. Three-dimensional computational fluid dynamics method was used with an SST (Shear Stress Transfer) turbulence model to simulate the flow field at the suction of the cleaning head module using the commercial code ANSYS-CFX. Suction pressure distributions were graphically depicted for different values of the design parameters.

1. Introduction

The average size of dust particles in a common vacuum cleaner is about 6–60 μm. Recently, the improvement in the quality of life from the increasing interest in health has made the performance of vacuum cleaners in removing mites and fine dust a priority. Numerous mites and fine dust particles that can cause allergies and respiratory diseases lie deep within carpets and on bedding. Thus, cleaning has expanded not only to surfaces but also to the depths of textiles. The inlet flow pattern of a vacuum cleaner used on a carpet is complicated by the complex form of the drum and brush module. Park et al. [1] analysed the flow characteristics and suction efficiency of a cleaner head installed a centrifugal fan. Cudina and Prezilj [2] analysed the suction performance and noise characteristics of a cleaner head with a vane by numerical and experimental methods. Lim et al. [3] experimentally investigated the dust removal efficiency of a vacuum cleaner with a new suction nozzle.

Reduction of mite allergen levels in an indoor environment has been attempted in many studies, utilising methods such as intensive carpet cleaning, laundering of bedding and the use of an occlusive mattress cover [4].

The two major purposes of developing a new vacuum cleaner are to enhance the suction efficiency and to remove fine dust particles. In this study, a new cleaning head module that incorporates a tubular fan was designed, and the suction efficiency of the module was derived.
A tubular fan is a kind of cross flow fans (CFFs), which delivers evenly distributed air velocity along the width of the sieves, but they are generally criticised for their inability to generate high static pressures [5]. The designs of both components of the casing, i.e., the rear wall and vortex wall, have an important influence on the performance of CFFs [6]. The CFF performance primarily depends on the position of the vortex centre, which is affected by the geometry and position of the vortex wall [7].

In this study, the prediction methods based on steady-flow CFD methods are described, and the numerical results are also used to illustrate the detailed flow fields in the vicinity of the suction head installed with a tubular fan, which is proposed as the new suction head module for the carpet domain.

2. Numerical methods

2.1 Governing equations

In order to improve the performance of the cleaning head module, three-dimensional, steady state, incompressible, viscous and turbulent flow fields are investigated using the following governing equations:

1) Continuity:

\[
\frac{\partial \rho u_i}{\partial x_i} = 0
\]  

(1)

2) Momentum:

\[
\frac{\partial \rho u_i u_i}{\partial x_i} = \frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_i}
\]  

(2)

where

\[
\tau_{ij} = \eta \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \eta \frac{\partial u_k}{\partial x_k} \delta_{ij}
\]  

(3)

Nomenclature

| Symbol | Description |
|--------|-------------|
| A      | The cross-sectional area of the pipe; the suction surface of the cleaning head module [m²] |
| b      | Blade width, [mm] |
| CFF    | Cross flow fan |
| k      | Turbulent kinetic energy, [m²/s²] |
| N      | Fan speed [rpm] |
| p      | Pressure, [Pa] |
| Δp     | Pressure difference, [Pa] |
| Q      | Flow rate, [m³/s] |
| R_c    | Radius of curvature, [m] |
| U      | Averaged velocity magnitude, [m/s] |
| \( \bar{u} \) | Velocity vector, [m/s] |
| \( u_i \) | i-th component of velocity vector, [m/s] |
| \( V \) | Air flow velocity, [m/s] |
| \( \varepsilon \) | Dissipation rate of energy, [m²/s²] |
| \( \eta \) | Dynamic viscosity, [kg/m.s] |
| \( \rho \) | Density, [kg/m³] |
| \( \tau_{ij} \) | Viscous stress tensor, [Pa] |
| \( \omega \) | Specific dissipation rate of turbulent kinetic energy |
In the above equations (1)-(3), $\rho$, $u$, $p$, $\tau$, $\eta$ and $\delta_e$ denote the density of fluid, the velocity of flow, the pressure, the shear stress, the dynamic viscosity of fluid and the Kronecker delta function, respectively. In order to assess the internal airflow configuration of the tubular fan and the effect of the position of the vortex wall on the fan’s performance, the commercial CFD software CFX was used. The immersed solids method was used to simulate the tubular fan. This method involves rigid, solid objects that can move through fluid domains. The model uses an immersed solid domain placed inside a fluid domain [8]. The steady state, three-dimensional, viscous, incompressible, Reynolds-averaged Navier-Stokes equations were solved. The standard $k-\varepsilon$ turbulence model was used to describe the effects of turbulence.

### 2.2 Fluid volume modeling

The schematic components of the modeled cleaning head module are shown in Figs. 1-3. A tubular fan was applied for nine different types of blades (blade number, $n=3, 4, 5$; blade width, $b=5, 10, 15$ mm).

#### Table 1. Cases with different blade configurations.

| Cases | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|-------|---|---|---|---|---|---|---|---|---|
| Blade width, $b$ (mm) | 5 | 5 | 5 | 10 | 10 | 10 | 15 | 15 | 15 |
| Blade number, $n$ | 3 | 4 | 5 | 3 | 4 | 5 | 3 | 4 | 5 |

*Figure 1.* 3-D modelling of the cleaning head module for a vacuum cleaner.

*Figure 2.* Geometry of the suction head.

*Figure 3.* A 3-blade tubular cross flow fan.
Table 2. Boundary conditions.

| Variables                        | Value                        |
|----------------------------------|------------------------------|
| Inlet condition                  | 1 atm                        |
| Working fluid                    | Air (298.15 K)               |
| Exit velocity                    | 40 m/s                       |
| Wall condition                   | No-slip wall                 |
| Interface mass and momentum      | Conservative interface flux  |
| Fan angular velocity             | 12,200 rev/min               |
| Porosity, $\varepsilon$          | 0.5                          |
| Heat transfer                    | Adiabatic                    |

The model cases with the different blade configurations are presented in Table 1. The suction region was assumed to be composed of porous materials to simulate a standard carpet region [4]. The porous domain was adopted a glass wool material under continuous solid and isotropic loss conditions. The volume porosity and the interfacial area density were $\varepsilon =0.5$ and $1[\text{m}^{-1}]$, respectively. The boundary conditions are presented in Table 2.

The geometrical model was generated by Inventor and the grid was generated by ICEM [9] based on the finite volume method on a collocated grid. An unstructured grid system was used for the cleaning head module. The number of total grid elements was about 1,170,000. The grid was considered to be sufficiently reliable to make the numerical modelling results independent of the grid size.

3. Results and discussion

3.1 Suction pressure
The suction pressure is an important parameter of vacuum cleaner performance. Figure 4 shows the characteristics of the mean suction pressure according to blade number ($n$) and blade width ($b$). A low mean surface pressure represents the excellent performance of the vacuum cleaner. When the blade width was $b=5$ mm, the mean suction pressure linearly increased as the blade number increased. When the blade width was $b=10$ mm and $n=4$ or higher, the mean suction pressure was the smallest. In particular, the mean suction pressure for case 6 ($n=5$, $b=10$ mm) was the lowest.

![Figure 4. Suction pressure distribution at the surface suction area.](image-url)
### Table 3. Comparison results of the airflow rate and aerodynamic power with different blade types.

| Blade width, b (mm) | Cases | 5 | 10 | 15 | No tubular fan |
|---------------------|-------|---|----|----|---------------|
|                     | Case 1 | Case 2 | Case 3 | Case 4 | Case 5 | Case 6 | Case 7 | Case 8 | Case 9 |
| Suction nozzle      |       |       |       |       |       |       |       |       |       |
| Q (m³/s)            | 0.05646 | 0.04550 | 0.04567 | 0.04548 | 0.04548 | 0.04545 | 0.04545 | 0.04549 | 0.04549 |
| AP (W)              | 75.4542 | 75.1759 | 75.4487 | 75.1180 | 75.1455 | 75.7028 | 75.0829 | 75.1734 | 75.1451 |
| Suction surface     |       |       |       |       |       |       |       |       |       |
| Q (m³/s)            | 0.05566 | 0.0551 | 0.05401 | 0.05561 | 0.0551 | 0.05618 | 0.05546 | 0.05587 | 0.05573 |
| AP (W)              | 93.4335 | 92.4880 | 90.6731 | 93.3515 | 92.4829 | 94.2984 | 93.0806 | 93.7797 | 93.5355 |

#### 3.2 Aerodynamic power

To evaluate the vacuum cleaner performance, the aerodynamic power (AP) of the vacuum cleaner was calculated as follows [10]:

\[
AP = Q \times p
\]  

(4)

where \( AP \) is the aerodynamic power in Watts, \( Q \) is the flow rate, and \( p \) is the mean suction pressure represented in absolute vacuum pressure.

The airflow rate (\( Q \)) can be expressed as follows:

\[
Q = A \times V
\]  

(5)

where \( A \) is the cross-sectional area of the pipe or the suction surface of the cleaning head module, and \( V \) is the airflow velocity.

Table 3 shows the \( AP \) and airflow rate at the suction nozzle and suction surface for different blade types. The results at the suction nozzle showed less than 1.01% difference, whereas the \( AP \) at the suction surface showed up to 4.01% difference. In order to evaluate the effects of a tubular fan on the performance of a vacuum cleaner, the aerodynamic power without a tubular fan was calculated; it was 84.9618 W, which indicated a suction power increase of 6.72%~10.99% by the installation of a tubular fan. In addition, in order to compare the performance of the vacuum cleaner for various blade types, the \( AP \) ratio (\( AP_{sf} / AP_{0} \)) was depicted in Fig. 5.

![Figure 5. The AP ratio distribution for each case.](image)

Here, \( AP_{sf} \) is the aerodynamic power at the surface area, and \( AP_{0} \) is the aerodynamic power at the suction nozzle. Case 6 has the highest value of the \( AP \) ratio, so it is selected as the optimal model among the considered models.
4. Conclusions
In this study, the flow characteristics in the vicinity of the cleaning head module of a vacuum cleaner were numerically investigated. Using a CFD method, the optimal design of the cleaning head module installed with a tubular fan was carried out for various blade types. In particular, $AP$ was calculated as the performance evaluation indicator. The following conclusions were obtained:
(a) The aerodynamic power analyses were carried out by verifying the pressure distribution, which showed less than 1% and 4% difference at the suction nozzle and suction surface, respectively.
(b) The case 6 showed the largest $AP$ ratio, making it the optimal model, with five blades of 10 mm blade width.
In order to commercialize this optimal cleaner head module, its dust removal performance on the carpet and blade durability should be experimentally verified by actual application.

References
[1] Park I S, Sohn C H, Lee S, Song H and Oh J 2010 J. Applied Acoustics 71(5) 460-69
[2] Cudina M and Prezelj J 2007 J. Applied Acoustics 68(5) 503-20
[3] Lim K S, Lee S H, Yoon J H, Kim Y J and Hong W S 2001 The Korean Society of Mechanical Engineering 2001 19-22
[4] Sercombe J K, Liu-Brennan D, Causer S M and Tovey E R 2007 Int. J. Hygiene and Environmental Health 210(1) 43-50
[5] Gebrehiwot M G, Baerdemaeker J D and Baelmans M 2010 J. Bio Systems Engineering 106(4) 448-57
[6] Eck B 1973 Fans (Oxford: Pergamon Press)
[7] Lazzaretto A, Toffolo A and Martegani A D 2003 J. Fluid Engineering 125(4) 684-93
[8] ANSYS CFX User guide ver. 14, 2012.
[9] ANSYS ICEM User guide, 2012
[10] Park C W, Lee S I and Lee S J 2004 J. KSME International 18(9) 1648-60