Experimental and Numerical Studies of a Wind Turbine with a Self-Adaptive Flanged Diffuser

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Experimental and Numerical Studies of a Wind Turbine

with a Self-Adaptive Flanged Diffuser

by

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ABSTRACT

The objective of this dissertation is to propose a new type of flanged diffuser that can not only accelerate the approaching wind when velocities are below the rated wind velocity, but also gradually reduce the wind loads acting on the wind turbine structure at high wind velocities.

Theoretical model is proposed first to predict the relationship between the normal drag coefficient of a rigid curved-plate and the configuration of the plate, through a series of numerical analyses of structure and fluid dynamics. Subsequently, based on the theoretical model, an approximate method is constructed for the evaluation of the normal force acting on the plate and the deformation of the plate using iteration of structural mechanics analysis rather than conventional complex iterations of fluid-structure coupling analysis. Simulation tests for three-dimensional (3D) flexible plates with different lengths and different material moduli are conducted. In addition, a comparative simulation test of a 3D flexible plate used in a previous experiment is performed to further confirm the validity and accuracy of the approximate method. Numerical results obtained from the approximate method agree well with those obtained from the fluid dynamics analysis as well as the results of the previous wind tunnel experiment.

A diffuser with a self-adaptive flange based on a flexible flange idea of two cantilevered plates is developed. The self-adaptive flange works without the aid of extra electrical or mechanical devices. Moreover, the flange can maintain the advantages of the flanged diffuser at lower wind velocities while reduce the wind loads acting on the diffuser and blades at higher wind velocities. Numerical analyses of fluid-structure interactions are performed to investigate the flow field around the diffuser with a self-adaptive flange as
well as the variation of wind load acting on the diffuser because of the reconfiguration of the self-adaptive flange at various wind velocities. The numerical results show that wind load acting on the total flanged diffuser at 60 m/s can be reduced by approximately 34.5% by the reconfiguration of the self-adaptive flange.

A small diffuser with a self-adaptive flange is manufactured, which contains a cylindrical part of CFRP structure and a flange of coupled plastic sheets. Wind tunnel experiments and numerical analysis are performed to investigate the flow field around the diffuser with the self-adaptive flange as well as the variation of wind load acting on the diffuser at various wind velocities. Experimental results show that the drag reduction for the diffuser with the self-adaptive is 17.9% at a velocity of 20 m/s. The wind velocity and pressure coefficient distributions in the radial direction for diffusers with either a rigid and flat flange, or a self-adaptive flange were obtained from experiments at $V_0 = 10 \text{ m/s}$. The values obtained from the two diffusers show good agreement because there was no significant reconfiguration of the self-adaptive flange. However, at $V_0 = 20 \text{ m/s}$, obvious differences in the velocity and pressure coefficients distributions between the two types of flanged diffuser are observed because of the deformation of the self-adaptive flange. Numerical results show good consistency with the experimental results of the velocity and pressure coefficient distributions, although the numerical analysis underestimates the values compared to the experimental results.
CONTENTS

CHAPTER 1 Introduction............................................................................................................. 1
  1.1 General background........................................................................................................... 2
    1.1.1 Development status of wind power in the world .................................................. 2
    1.1.2 Development status of wind power in Japan ......................................................... 5
  1.2 Introduction of diffuser shrouded wind turbine .............................................................. 8
  1.3 Objective and outline ...................................................................................................... 14

Bibliography .......................................................................................................................... 16

CHAPTER 2 Approximate method for the evaluation of the normal force acting on a
flexible plate normal to the wind flow....................................................................................... 20
  2.1 Introduction....................................................................................................................... 21
  2.2 Governing equations ....................................................................................................... 22
    2.2.1 Continuity and momentum equations ..................................................................... 23
    2.2.2 Reynolds-averaged Navier-Stokes equations .......................................................... 25
    2.2.3 Standard $k – \varepsilon$ model ..................................................................................... 26
    2.2.4 Large eddy simulation ............................................................................................. 27
    2.2.5 Nonlinear structural analysis ................................................................................... 29
    2.2.6 Large deflections of plates ...................................................................................... 30
  2.3 Approximate method ....................................................................................................... 39
    2.3.1 Theoretical modeling .............................................................................................. 39
    2.3.2 Determination of $f(\tilde{\theta})$ ............................................................................... 43
2.3.3 Algorithm to solve the average pressure acting on a flexible plate normal to the flow .......................................................... 45

2.4 Numerical simulation tests ................................................................................. 49

2.4.1 Simulation tests for five flexible plates ......................................................... 50

2.4.2 A comparative simulation test ....................................................................... 53

2.5 Results and discussion ....................................................................................... 54

2.6 Summary ........................................................................................................... 62

Bibliography ........................................................................................................... 64

CHAPTER 3 Upgrading a shrouded wind turbine with a self-adaptive flanged diffuser 67

3.1 Introduction ....................................................................................................... 68

3.2 A novel diffuser with a self-adaptive flange .................................................... 69

3.3 Numerical simulations ...................................................................................... 74

3.3.1 Structural mechanics analysis ...................................................................... 76

3.3.2 CFD Analysis ............................................................................................... 78

3.4 Results and discussion ...................................................................................... 81

3.5 Summary .......................................................................................................... 92

Bibliography .......................................................................................................... 94

CHAPTER 4 Experimental study of a small diffuser with a self-adaptive flange ....... 97

4.1 Introduction ....................................................................................................... 98

4.2 A small diffuser with a self-adaptive flange .................................................... 98

4.3 Experiments ..................................................................................................... 99

4.3.1 Manufacture of the mold ............................................................................. 100

4.3.2 Fabrication of diffusers with either a flat flange or a self-adaptive flange. 103

4.3.3 Wind tunnel experiments ............................................................................ 114

4.4 Numerical comparisons .................................................................................... 118

4.5 Results and discussion ..................................................................................... 120
4.6 Summary ........................................................................................................ 133
Bibliography ........................................................................................................ 135
CHAPTER 5 Conclusions and future works .......................................................... 136
  5.1 Conclusions .................................................................................................... 136
  5.2 Future work .................................................................................................... 137
LIST OF FIGURES ............................................................................................. 139
LIST OF TABLES ................................................................................................. 146
LIST OF PUBLICATIONS .................................................................................... 147
ACKNOWLEDGEMENTS .................................................................................... 148
CHAPTER 1

Introduction

This chapter introduces the general background of wind power generation in the world, the development of diffuser augmented wind turbine (DAWT), as well as the main objectives and outline of this dissertation.
1.1 General background

1.1.1 Development status of wind power in the world

With increasing energy consumption and a growing sense of environmental issues, such as pollution and global climate change, the development of clean and sustainable energy sources has attracted considerable attention worldwide. Wind energy is one of the most important sustainable energy sources, and wind turbine systems are an important method for extracting energy from wind. According to the global wind report 2014 [1] from the Global Wind Energy Council (GWEC), the annual installation of wind power in 2014 was 51,473 MW, which was a new record, showing a sharp rise compared to 2013 (Fig. 1-1). At the end of 2014, the new global cumulative wind capacity was 369.6 GW with a growth rate of > 16% (Fig. 1-2).

![Figure 1-1: Global annual installed wind capacity 1977-2014 [1].](image)

Figs. 1-3 and 1-4 show ten countries with the highest numbers of newly installed wind capacity and cumulative wind capacity, respectively, in 2014, as well as their share in the total world amount. China, the largest market for wind power since 2009, led the global
market again in 2014 and maintained the top spot at the largest share of 45.1%, with Germany in the second spot, and USA in the third spot. As for the cumulative wind capacity, six countries crossed 10,000 MW of installed capacity by the end of 2014, including China (114609 MW), the USA (65879 MW), Germany (39165 MW), Spain (22987 MW), India (22645 MW) and the UK (12440 MW).

Figure 1-2: Global cumulative installed wind capacity 1977-2014 [1].

Figure 1-3: Top 10 new installed wind capacity Jan-Dec 2014 [1].
Asia, Europe, and North America are the main markets for wind power generation, as shown in the annual installed capacity by region (Fig. 1-5). Asia, the world’s largest...
regional market since 2008, rapidly developed in wind power in recent years and obtained a sharp rise of wind capacity in 2014 compared to 2013, with a total capacity > 26 GW of which China contributed 23 GW. In contrast, Europe showed steady growth, and added 12,858 MW of new capacity in 2014. However, a slowdown in wind power development in North America is clear, although the annual installed capacity in 2012 was exciting.

**Table 1-1: Installed wind power capacity in Asia (MW) [1].**

<table>
<thead>
<tr>
<th>Country</th>
<th>Capacity until 2013</th>
<th>New capacity in 2014</th>
<th>Total capacity until 2014</th>
</tr>
</thead>
<tbody>
<tr>
<td>PR China</td>
<td>91413</td>
<td>23196</td>
<td>114609</td>
</tr>
<tr>
<td>India</td>
<td>20150</td>
<td>2315</td>
<td>22465</td>
</tr>
<tr>
<td>Japan</td>
<td>2669</td>
<td>130</td>
<td>2789</td>
</tr>
<tr>
<td>Taiwan</td>
<td>614</td>
<td>18</td>
<td>633</td>
</tr>
<tr>
<td>South Korea</td>
<td>561</td>
<td>47</td>
<td>609</td>
</tr>
<tr>
<td>Thailand</td>
<td>223</td>
<td>-</td>
<td>223</td>
</tr>
<tr>
<td>Pakistan</td>
<td>106</td>
<td>150</td>
<td>256</td>
</tr>
<tr>
<td>Philippins</td>
<td>66</td>
<td>150</td>
<td>216</td>
</tr>
<tr>
<td>Other</td>
<td>167</td>
<td>-</td>
<td>167</td>
</tr>
<tr>
<td>Total</td>
<td>115968</td>
<td>26007</td>
<td>141964</td>
</tr>
</tbody>
</table>

Table 1-1 lists the installed wind power capacity of several main countries in Asia. India is the second largest wind market in Asia after China. India achieved new wind energy installations of 2315 MW in 2014, and reached a total of 22465 MW. However, the rest of Asia did not show much development in 2014.

### 1.1.2 Development status of wind power in Japan

In Japan, 130.4 MW of wind capacity was installed in 2014 to reach a cumulative capacity of 2789 MW (Table 1-1), which is only 0.7546% of the total wind global capacity. This share drops to 19th in the world compared to 13th in 2012. According to the statistics
of the power generation capacity from the New Energy and Industrial Technology Development Organization (NEDO) of Japan, after rapid development in the 1990s, the progress of wind power generation shows a slowdown over the last four years in a row, as shown in Fig. 1-6.

![Figure 1-6: Installed wind capacity in Japan 1995-2012 [2].](image)

![Figure 1-7: Transition of power generation capacity in Japan 1952-2012 [3].](image)
Figure 1-8: Increase in rated power and size of wind turbines [4].

Figure 1-9: (a) Onshore wind potential map and (b) offshore wind potential map in Japan [2].
Similarly, the agency for natural resources and energy of Japan reported that the proportion of renewable energy in the total power supply of Japan is merely 0.4% [3]. Clearly, Japan is slow to transform its energy system to allow for a more diverse energy mix including more wind power and other renewable energy, as shown in Fig. 1-7.

Because of the low energy density of wind, a larger wind turbine is expected to extract more wind power and reduce the cost of power generation, based on the theory that the output of the wind turbine is proportional to the square of the diameter of the rotor plane. A clear trend towards larger wind turbines can be observed over the last two decades as depicted in Fig. 1-8, especially in the deployment of offshore wind power. For instance, V164-8MW, which is presently the largest offshore, three bladed, upwind wind turbine, with a rated capacity of 8 MW, and a rotor diameter of 164 m, was developed by Vestas Wind Systems and put into use in 2014.

It is difficult to find suitable land areas, with an annual average wind speed over 7 m/s, for wind power plants in Japan compared to Europe and North America. However, the offshore wind potential is relatively rich in the areas of Northeast, Kanto, and Kyushu, as shown in Fig. 1-9. In addition, the complex terrain and turbulent nature of the local wind limit the development of wind power generation in Japan. Therefore, it is very useful to develop a new wind power system that can produce high power output even in areas with lower wind velocities and complex wind patterns.

1.2 Introduction of diffuser shrouded wind turbine

It is known that the power extractable from the wind by a wind turbine increases with the cube of the wind velocity so that even small increase in wind velocity can lead to a
large increase in wind power generation. Therefore, many researchers have been making efforts to develop new technologies that can effectively accelerate the approaching wind velocity. One of advanced wind turbine designs is the diffuser augmented wind turbine (DAWT). It was first proposed by Lilley and Rainbird in 1956 [5]. Experimental studies were then carried out by Oman et al. [6], Igra [7, 8], Foreman and Gilbert [9], Phillips et al. [10]. Fig. 1-10 presents the view of two typical DAWTs. These experimental results proved that the DAWT can extract much more power from the wind than any similar bare wind turbine with the same rotor diameter. Afterwards, many theoretical and numerical studies [11-16] have been performed to investigate and improve the performance of DAWTs. Most of these researches presented the similar conclusions with the early

**Figure 1-10:** View of the shrouded wind turbines, (a) Ben Gurion University, Israel [8], (b) Vortec 7 prototype DAWT, New Zealand [10].
experimental results. Results from Hasen [11] showed that the power coefficient of the wind turbine with a wing profiled shroud is 1.8 times that of a bare wind turbine with the aid of a wing profiled shroud compared to a bare wind turbine (Fig. 1-11a). Similarly, Bet and Grassmann [12] reported that about a 2-fold power augmentation was achieved by using two wing profiled rings mounted around the rotor using a wing profiled ring (Fig. 1-11b).

![Wind turbine shrouded with cylindrical wing profiled structure](image)

**Figure 1-11:** Wind turbine shrouded with cylindrical wing profiled structure, (a) Hasen [11], (b) F. Bet and H. Grassmann [12].

Computational fluid dynamics (CFD) analysis for a shrouded wind turbine with a simple frustum diffuser was carried out by Jafari [13]. The shrouded wind turbine improved the power extraction over a bare turbine by nearly 1.7 times. Van Bussel [14] demonstrated two power coefficients based on the rotor swept area, and the diffuser exit area, and indicated that the power coefficients of almost all the DAWTs reported based on the diffuser exit area are less than the Betz limit, although the power coefficients based on the
rotor swept area are larger than the Betz limit. The equations for the two power coefficients were expressed as follows:

\[
C_{p,\text{rotor}} = \beta \gamma 4\alpha (1 - \alpha)^2 \quad 1-1
\]

\[
C_{p,\text{exit}} = \gamma 4\alpha (1 - \alpha)^2 \quad 1-2
\]

Where, \(\alpha\) and \(\beta\) are axial induction factor at the exit of the diffuser and the ratio of the diffuser exit area to the rotor plane area, respectively, and \(\gamma\) is the back pressure velocity ratio, which is an unknown function of variable system efficiency of the DAWT. It was reported by Werle and Presz [15] that a thrust increase of nearly 80% above the bare turbine level is attainable with moderate diffusion and exit plane suction pressures. Jamieson [16] derived unified expressions for power coefficient and its maximum value based on the rotor swept area as follows.

\[
C_{p,\text{rotor}} = \frac{4(a - a_0)(1 - a)^2}{(1 - a_0)^2}, \quad 1-3
\]

\[
C_{p,\text{max,rotor}} = \frac{9}{8} (1 - a_{\text{max}}) = \frac{16}{27} (1 - a_0), \quad a_{\text{max}} = \frac{1 + 2a_0}{3}, \quad 1-4
\]

where, \(a_0\) and \(a\) are the axial induction factors at the energy extraction plane in an empty diffuser and in a diffuser with rotor, respectively. They found that the ideal maximum power coefficient of DAWT based on the rotor swept area is 8/9, which is 1.5 times the Betz limit according to Eq. 1-4.

In addition to the above research on DAWTs, Ohya et al. [17, 18] have developed a new shrouded wind turbine with a large flange mounted at the exit of the diffuser and a curved inlet section. A 3 kW wind turbine shrouded with a flanged diffuser and an illustration of the wind flow around the flanged diffuser are shown in Fig. 1-12. They found
that the large flange created a low-pressure region owing to the strong vortices formed behind the flange based on the wind tunnel tests. As a result, much more wind was drawn into the diffuser than in a general DAWT without the flange. Their experimental and numerical results showed that the flange increased the wind velocity in the nozzle of the diffuser by 1.6–2.4 times over the upwind velocity, and by 1.2–1.7 times over the wind velocity in the nozzle of the diffuser obtained by the same diffuser but without the flange, respectively. This acceleration effect of the wind velocity led to an increase in the power extraction by 2–3 times over the bare wind turbine and 1.4–2.1 times over the wind turbine with the same diffuser but without the flange, respectively. After that, extensive experiments and CFD simulations were carried out to investigate the effects of various geometrical parameters of the flanged diffuser on the flow field around the diffuser and the power coefficients [19-22]. It was reported in [19] that a 500 W wind turbine shrouded by a long flanged diffuser showed an increase in power output by 4–5 times compared to the bare wind turbine with the same rotor diameter. Several compact flanged diffusers were

Figure 1-12: (a) A 3 kW wind turbine shrouded by a flanged diffuser and (b) the illustration of wind flow around the flanged diffuser[19].
investigated in [22], and the experiments revealed that a 5kW diffuser shrouded wind turbine obtained 2.5 times augmentation in output power as compared to the bare wind turbine. It was further reported that the power coefficient based on the diffuser exit area is 0.54, and it is still higher than that of the conventional wind turbine without diffuser (0.4). Computations were performed by Mansour and Meskinkhoda [23] to investigate the flow field around flanged diffuser and their numerical results were consistent with those previously reported by Abe et al. [20]. It was further proved that the flange improves on-axis stream velocity at the inlet by 1.18 times over the wind turbine with the same diffuser but without the flange. Two 100 kW wind turbines shrouded by the flanged diffuser have been built at the Ito campus of Kyushu University and put into use, as shown in Fig. 1-13.

**Figure 1-13:** Two 100 kW wind turbines shrouded by a flanged diffuser at the Ito campus of Kyushu University.
In addition to the above research in the flow field around the flanged diffuser, the influence of flanged diffuser on the dynamic behavior of rotating rotor blades of a 3 kW wind turbine with a compact flanged diffuser was investigated by wind tunnel experiments in a recent work [24]. It was found that the rotational speed of the blade of the wind turbine with a flanged diffuser is about 1.35 times that without the flanged diffuser, which led to higher strains at the root of the rotating blades of the wind turbine with a flanged diffuser than those in the rotating blades of the bare wind turbine with the same rotor. According to this recent research, issues involving the wind turbine structure are again recognized. In addition to the advantages, including the acceleration effect of wind velocity, low noise, and kindness to birds, the drawbacks of the wind turbine shrouded by a flanged diffuser are also obvious. An extra and large structure, namely, the flanged diffuser is added to a bare wind turbine, which makes this kind of wind turbine complicated and costly, especially for very large-scale wind turbines.

1.3 Objective and outline

The objective of this dissertation is to develop a diffuser with a self-adaptive flange, which is based on a novel idea of bi-cantilevered plates. This self-adaptive flange can not only accelerate the approaching wind with velocities below the rated wind velocity but can also gradually reduce the wind load acting on the wind turbine structure at high wind velocities.

The outline of this dissertation is listed below:

Chapter 1 introduces the general background of wind power development and the previous research on diffuser shrouded wind turbines.
Chapter 2 describes an approximate numerical method for the evaluation of the normal force acting on a flexible plate normal to the wind flow and the deformation of the plate using the iteration of structural mechanics analysis rather than conventional complex iterations of fluid-structure coupling analysis [25, 26].

Chapter 3 proposes a diffuser with a self-adaptive flange based on a flexible flange idea comprising two cantilevered plates [27]. A 100 kW wind turbine with a typical compact-type flanged diffuser is considered as a model in the present study. Numerical analyses of fluid-structure interaction between the flow and the diffuser with a self-adaptive flange are performed to investigate the flow field around the flanged diffuser and the variation of wind load acting on the flanged diffuser due to the reconfiguration of the self-adaptive flange at various wind velocities.

Chapter 4 describes the manufacture of two small diffusers, one with a self-adaptive flange and the other with a flat and rigid flange, in order to demonstrate the validity of the self-adaptive flange. Wind tunnel experiments are performed to investigate the flow field around the two types of flanged diffusers and the variation of wind loads acting on the diffusers at various wind velocities.

Chapter 5 summarizes the major results of the present dissertation.
Bibliography


CHAPTER 2

Approximate method for the evaluation of the normal force acting on a flexible plate normal to the wind flow

In this chapter, an approximate method is developed for evaluating the normal force acting on a flexible plate normal to the wind flow and the deformation of the plate based on a theoretical modeling. The model describes an approximate relationship between the normal drag coefficient of a rigid curved-plate and the configuration of the plate through a series of numerical analyses of structure and fluid dynamics.
2.1 Introduction

A flexible plate normal to the wind flow is a typical and classic problem of fluid-structure interaction (FSI), of direct relevance to many natural phenomena. For instance, leaves of plant, flexible fiber, and plates roll up in a high wind to reduce the drag and avoid damage as reported by Vogel [1, 2], Alben et al. [3, 4], Schouveiler and Boudaoud [5, 6], and Gosselin et al. [7]. Furthermore, this problem is also of relevance to many practical applications such as commercial plates and turbine blades [8-10]. Many research efforts [11-18] have been contributed to the investigation of the dynamic behaviors of the flow around a rigid-plate, such as drag coefficient, lift coefficient, Strouhal numbers, velocity fluctuation behind the plate, and vortex behaviors. Relatively few studies [1, 2], Alben et al. [3, 4], Schouveiler and Boudaoud [5, 6], and Gosselin et al. [7], Campbell and Paterson [19], Lee and Lee [20] have been focused on the evaluation of the wind pressure acting on a flexible plate and the deformation of the plate. On the other hand, the wind pressure acting on a deformable plate and the deformation of the plate are important parameters in the strength design of the plate and its supportive structure in practical applications.

In general, it is extremely difficult to derive an analytical solution for the evaluation of the wind pressure acting on a flexible plate because of the strong nonlinearity in the coupling of the fluid flow and the plate deformation. Rigorous evaluation requires complex fluid-structure coupling analysis, although many numerical methods have been developed as reviewed by Hou et al. [21] and Degroote [22], such as the Arbitrary Lagrangian-Eulerian (ALE) finite element method [23, 24] and the Boltzman-Lattice method [20, 25]. Complex iteration procedures of numerical calculations related to the alternative fluid and structural analyses are cumbersome and error-prone in the rigorous FSI numerical analysis.
Therefore, developing a relatively simple numerical method to evaluate the wind pressure acting on a deformable plate is quite useful in various practical applications.

In this chapter, we are interested to develop an approximate numerical method for evaluating the normal force acting on a flexible plate normal to the wind flow and the deformation of the plate. The averaged pressure is defined by the normal force divided by the plate area. A theoretical modeling is proposed to approximately describe the relationship between the normal drag coefficient of a rigid curved-plate in the flow and its configuration, with the aid of a series of regular numerical calculations of fluid dynamics and structural mechanics. Based on the theoretical curve of the normal drag coefficient and the configuration of the rigid curved-plate, an approximate method for evaluating the normal force acting on the plate and the deformation of the plate is constructed using iteration of structural mechanics analysis rather than conventional complex iterations of fluid-structure coupling analysis. Based on the proposed approximate method, we can easily predict the averaged pressure acting on the plate and the deformation of the plate using conventional CFD and structural analysis codes independently. Simulation tests for 3D flexible plates with different lengths and different material moduli are conducted. In addition, a comparative simulation test of a 3D flexible plate used in a previous experiment is performed to further confirm the validity and accuracy of the approximate method.

2.2 Governing equations

General governing equations of fluid dynamics and structure mechanics, used in the present approximate method are introduced in this section.
2.2.1 Continuity and momentum equations

The conservation equations of mass and momentum are described in Eqs. 2-1 and 2-2, respectively [26, 27].

\[
\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{u}) = 0, \tag{2-1}
\]

where \( \rho \) is mass density, \( \vec{u} \) is Eulerian fluid velocity, and \( t \) is time, and

\[
\frac{\partial}{\partial t} (\rho \vec{u}) + \nabla \cdot (\rho \vec{u} \vec{u}) = -\nabla p + \rho \vec{f}, \tag{2-2}
\]

\[
\frac{\partial}{\partial t} (\rho \vec{u}) + \nabla \cdot (\rho \vec{u} \vec{u}) = -\nabla p + \nabla \cdot \vec{\tau} + \rho \vec{f},
\]

where \( p \) is hydrostatic pressure, \( \vec{\tau} \) is viscous stress tensor, and \( \vec{f} \) is the body force per unit mass.

Figure 2-1: Elemental Cartesian fixed control volume showing surface forces in the \( x \) direction only.
Fig. 2-1 shows the surface forces acting on the fixed control element volume in a Cartesian coordinate system in the x-direction only to avoid cluttering the drawing. The surface forces are the sum of hydrostatic pressure and viscous stresses. For a Newtonian fluid, the viscous stresses are proportional to the element strain rates and the coefficient of viscosity and are expressed as follows [26, 27]:

\[
\tau_{xx} = 2 \mu \frac{\partial u_x}{\partial x} - \frac{2}{3} \mu \nabla \cdot \vec{u}, \quad \tau_{xy} = 2 \mu \frac{\partial u_y}{\partial y} - \frac{2}{3} \mu \nabla \cdot \vec{u},
\]

\[
\tau_{zz} = 2 \mu \frac{\partial u_z}{\partial z} - \frac{2}{3} \mu \nabla \cdot \vec{u}, \tag{2-3a}
\]

\[
\tau_{yx} = \tau_{xy} = \mu (\frac{\partial u_x}{\partial y} + \frac{\partial u_y}{\partial x}), \quad \tau_{xz} = \tau_{zx} = \mu (\frac{\partial u_x}{\partial z} + \frac{\partial u_z}{\partial x}),
\]

\[
\tau_{yz} = \tau_{zy} = \mu (\frac{\partial u_y}{\partial z} + \frac{\partial u_z}{\partial y}),
\]

where \( \mu \) is the viscosity coefficient. In the present study, the fluid is assumed to be incompressible. Thus, Eq. 2-3a reduces to following Eq. 2-3b.

\[
\tau_{xx} = 2 \mu \frac{\partial u_x}{\partial x}, \quad \tau_{xy} = 2 \mu \frac{\partial u_y}{\partial y}, \quad \tau_{zz} = 2 \mu \frac{\partial u_z}{\partial z},
\]

\[
\tau_{yx} = \tau_{xy} = \mu (\frac{\partial u_x}{\partial y} + \frac{\partial u_y}{\partial x}), \quad \tau_{xz} = \tau_{zx} = \mu (\frac{\partial u_x}{\partial z} + \frac{\partial u_z}{\partial x}), \tag{2-3b}
\]

\[
\tau_{yz} = \tau_{zy} = \mu (\frac{\partial u_y}{\partial z} + \frac{\partial u_z}{\partial y}).
\]

Substituting Eq. 2-3b into Eq. 2-2 yields the differential momentum equations for a incompressible Newtonian fluid with constant density and viscosity:
\[
\rho f_i = \frac{\partial p}{\partial x_i} + \mu \left( \frac{\partial^2 u_i}{\partial x_j \partial x_j} + \frac{\partial^2 u_i}{\partial y^2} + \frac{\partial^2 u_i}{\partial z^2} \right) = \rho \frac{\partial u_i}{\partial t} + \rho u_j \frac{\partial u_i}{\partial x_j}
\]

That is, 2-4

\[
\rho f_x = \frac{\partial p}{\partial x} + \mu \left( \frac{\partial^2 u_x}{\partial x^2} + \frac{\partial^2 u_x}{\partial y^2} + \frac{\partial^2 u_x}{\partial z^2} \right) = \rho \frac{Du_x}{Dt},
\]

\[
\rho f_y = \frac{\partial p}{\partial y} + \mu \left( \frac{\partial^2 u_y}{\partial x^2} + \frac{\partial^2 u_y}{\partial y^2} + \frac{\partial^2 u_y}{\partial z^2} \right) = \rho \frac{Du_y}{Dt},
\]

\[
\rho f_z = \frac{\partial p}{\partial z} + \mu \left( \frac{\partial^2 u_z}{\partial x^2} + \frac{\partial^2 u_z}{\partial y^2} + \frac{\partial^2 u_z}{\partial z^2} \right) = \rho \frac{Du_z}{Dt},
\]

\[
\frac{Df}{Dt} = \frac{\partial f}{\partial t} + u_j \frac{\partial f}{\partial x_j}.
\]

These are the Navier-Stokes (NS) equations for incompressible fluid, named after C. L. M. H. Navier and Sir George G. Stokes. The equations are second-order, nonlinear, partial differential equations, but many solutions have been found for various interesting viscous flow problems.

2.2.2 Reynolds-averaged Navier-Stokes equations

In the turbulent flow, the field properties become random functions of space and time. Hence, the field variables \( u \) and \( p \) can be expressed as the sum of mean and fluctuating parts as [27]:

\[
u_i = U_i + u_i' \quad p = P + p',
\]

where the mean and fluctuating parts satisfy

\[
U_i = \bar{u}_i, \quad \bar{u}_i' = 0, \quad P = \bar{p}, \quad \bar{p}' = 0,
\]

2-5a

2-5b
with the bar denoting the time average.

Substituting the expression of Eq. 2-5a into the continuity and momentum equations of incompressible flow and taking the time average produce the Reynolds-Averaged Navier-Stokes (RANS) equations as follows.

\[
\frac{\partial U_i}{\partial x_i} = 0,  \hspace{1cm} 2-6
\]

\[
\rho \frac{\partial U_i}{\partial t} + \rho U_j \frac{\partial U_i}{\partial x_j} = \rho f_i - \frac{\partial P}{\partial x_i} + \mu \frac{\partial^2 U_i}{\partial x_j \partial x_j} - \rho \frac{\partial \overline{u_i' u_j'}}{\partial x_j}  \hspace{1cm} 2-7
\]

The quantity \( R_{ij} = -\overline{\rho u_i' u_j'} \) is known as the Reynolds stress tensor which is a symmetric tensor of second order and thus has six independent components. The Reynolds-Averaged Navier-Stokes equations govern the transport of the average flow quantities over the entire range of turbulence scales being modeled. The RANS-based modeling approach therefore greatly reduces the required computational effort and resources, and is widely adopted for practical engineering applications.

2.2.3 Standard \( k - \varepsilon \) model

In order to apply the Reynolds-averaged approach to turbulence modeling, the Reynolds stress \( -\overline{\rho u_i' u_j'} \) must be appropriately modeled. As a common method, Reynolds stress is related to the mean velocity gradients based on the Boussinesq hypothesis:

\[
-\overline{\rho u_i' u_j'} = \mu_t \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \rho k \delta_{ij},  \hspace{1cm} 2-8
\]

\[
k = \frac{1}{2} u_i' u_i',
\]
where $\mu_\text{t}$ is the turbulent viscosity and $k$ is the turbulence kinetic energy, and $\delta_{ij}$ is the Kronecker delta.

The Boussinesq hypothesis is used in the $k-\epsilon$ model [28-30]. The standard $k-\epsilon$ model is a semi-empirical model based on two model transport equations for the turbulence kinetic energy $k$ and its dissipation rate $\epsilon$. In the derivation of $k-\epsilon$ model, it is assumed that the flow is fully turbulent and the effects of molecular viscosity are negligible. The two model transport equations of the standard $k-\epsilon$ model in ANSYS Fluent [29] are expressed as follows:

$$\rho \frac{\partial k}{\partial t} + \rho u_j \frac{\partial k}{\partial x_j} = \frac{\partial}{\partial x_j} \left( (\mu + \frac{\mu_\text{t}}{\sigma_k}) \frac{\partial k}{\partial x_j} \right) + G_k + G_b - \rho \epsilon - Y_M + S_k, \quad 2.9$$

$$\rho \frac{\partial \epsilon}{\partial t} + \rho u_j \frac{\partial \epsilon}{\partial x_j} = \frac{\partial}{\partial x_j} \left( (\mu + \frac{\mu_\text{t}}{\sigma_\epsilon}) \frac{\partial \epsilon}{\partial x_j} \right) + C_1 \frac{\epsilon}{k} (G_k + C_3 \epsilon G_b) - C_2 \rho \frac{\epsilon^2}{k} + S_\epsilon, \quad 2.10$$

where $G_k$ and $G_b$ represent the generation of turbulence kinetic energy due to the mean velocity gradients and buoyancy, respectively. $Y_M$ represents the contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate. $G_{1\epsilon}$, $G_{2\epsilon}$ and $G_{3\epsilon}$ are constants. $\sigma_k$ and $\sigma_\epsilon$ are the turbulent Prandtl numbers for $k$ and $\epsilon$, respectively. $S_k$ and $S_\epsilon$ are user-defined source terms for $k$ and $\epsilon$, respectively.

### 2.2.4 Large eddy simulation

The simulation of turbulence by numerically solving Navier-Stokes equations requires to resolve an entire range of time and length scales. In theory, it is possible to directly resolve the entire spectrum of turbulent scales using direct numerical simulation (DNS).
However, DNS is not feasible for practical engineering problems of high Reynolds number flows because of the high computational cost of DNS in resolving the entire range of scales. Large eddy simulation (LES) is a mathematical model for turbulence used in computational fluid dynamics to reduce computational cost by reducing the range of time and length scales. In LES, only large eddies are resolved directly, whereas small eddies are effectively removed.

The governing equations employed for LES are obtained by filtering the time-dependent Navier-Stokes equations. The filtering process effectively filters out eddies with scales smaller than the filter width. The LES model is applicable to compressible flows, and the filtered Navier-Stokes equations are presented as follows [29]:

\[
\begin{align*}
\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i} (\rho \bar{u}_i) &= 0, \quad 2-11 \\
\frac{\partial}{\partial t} (\rho \bar{u}_i) + \frac{\partial}{\partial x_j} (\rho \bar{u}_i \bar{u}_j) &= \frac{\partial}{\partial x_j} (\rho \bar{u}_i \bar{u}_j) \quad 2-12
\end{align*}
\]

where the filtered variables are denoted by an overbar, and \( \sigma_{ij} \) is the stress tensor due to molecular viscosity and is defined by the following equation:

\[
\sigma_{ij} = [\mu (\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i})] - \frac{2}{3} \mu \frac{\partial \bar{u}_i}{\partial x_i} \delta_{ij}, \quad 2-13
\]

and \( \tau_{ij} \) is the subgrid-scale stress defined by:

\[
\tau_{ij} = \rho \bar{u}_i \bar{u}_j - \rho \bar{u}_i \bar{u}_j, \quad 2-14
\]
2.2.5 Nonlinear structural analysis

As described in the introduction of this chapter, large deformation of the plate is considered in the present study in order to make better simulation for the deformation of the self-adaptive flanged diffuser. Thus, the nonlinear structure analysis is conducted. The expression of the force-displacement relation for a nonlinear problem is defined by [31]:

\[ P = K(P, d)d, \]

where, \( P \) is a generalized force vector, \( K \) is the stiffness matrix, and \( d \) is a generalized displacement vector.

For problems in classical linear elastostatics, this relation can be written in the form:

\[ P = Kd, \]

where the stiffness matrix \( K \) is independent of both \( d \) and \( P \).

There are three sources of nonlinearity: material, geometric, and nonlinear boundary conditions. Material nonlinearity results from the nonlinear relationship between stresses and strains and shows different behaviors such as the elasto-plasticity, elasto-viscoplasticity, and creeping. Geometric nonlinearity results from the nonlinear relationship between strains and displacements. Conversely, the nonlinear relationship results from the relationship between stresses and forces. Boundary conditions can also cause nonlinearity. For instance, loads on a structure cause nonlinearity if they vary with the displacements of the structure.

For these nonlinear problems, incremental theories are employed and the formulation of the incremental theories begins by dividing the loading path of the solid body problem into a number of equilibrium states [32]

\[ \Omega^{(0)}, \Omega^{(1)}, \ldots, \Omega^{(N)}, \Omega^{(N+1)}, \ldots, \Omega^{(f)}, \]
where, $\Omega^{(0)}$ and $\Omega^{(f)}$ are the initial and final states of the deformation, respectively, whereas $\Omega^{(N)}$ is an arbitrary intermediate state. It is assumed that all the state variables such as stresses, strains and displacements, together with the loading history, are known up to the $\Omega^{(N)}$ state. Then, an incremental theory is formulated for determining all the state variables in the $\Omega^{(N+1)}$ state, under the assumption that the $\Omega^{(N+1)}$ state is incrementally close to the $\Omega^{(N)}$ state and all the governing equations may be linearized with respect to the incremental quantities. The Lagrangian method, in which the second Piola-Kirchhoff stress tensors and the Green strain tensors are used, is used to formulate the incremental theory in nonlinear structural problems and it can be classified into two categories: the total Lagrangian method and the updated Lagrangian method. In present study, the software MSC. MARC 2010 is used and the updated Lagrangian method is employed in the nonlinear structure analysis.

2.2.6 Large deflections of plates

In order to get a better understanding on the large deflection of a thin plate, the well-known theory [33] of von Karman for the large deflection of thin plates is introduced before conducting the nonlinear structure analysis. The term large deflection indicates that the transverse deflection $w$ of a plate under bending load is no longer small compared with its thickness $h$, that is, $w \approx O(h)$. Thus, the middle surface of the plate begins to stretch and the deformed configuration of the plate differs significantly from the original one (see Fig. 2-2). When these effects become significant one needs to consider the geometrical nonlinearity of the plate deformation. In present study, it is designed that the self-adaptive
flange can undergo large deflection to reduce the wind load at high wind velocities. Therefore, geometrical nonlinearity is considered for the deformation of thin plates.

In the theory of large deflection of thin plates (see Fig. 2-3), there are six basic assumptions as follows:

**Figure 2-2:** Corresponding stresses in the initial and deformed configuration of a plate.

**Figure 2-3:** A thin plate and the coordinate system.

H1. The plate is thin. The thickness $h$ is much smaller than the characteristic dimension $L$ or $W$ of the plate in its plane, i.e., $h \ll L$ or $h \ll W$. 

31
H2. The magnitude of the deflection \( w \) is of the same order of magnitude as the thickness of the plate \( h \) but smaller as compared with the typical plate dimension \( L \).

H3. The slope is small everywhere, \( |\partial w/\partial x| \ll 1, |\partial w/\partial y| \ll 1 \).

H4. The tangential displacements of the middle surface of the plate are infinitesimal. In the strain displacement relations, only those nonlinear terms related to \( \partial w/\partial x \) and \( \partial w/\partial y \) are retained. All other nonlinear terms are neglected.

H5. All strain components are small. Hooke’s law holds.

H6. Kirchhoff’s hypotheses hold; i.e., the tractions on surfaces parallel to the middle surface are negligible. Strains vary linearly within the plate thickness.

In the large deflection theory of thin plate, Lagrangian description is used to describe the motion and deformation of the body. A fixed right-handed rectangular Cartesian frame is used, with the \( x, y \) plane coinciding with the middle surface of the plate in its initial unloaded state (see Fig. 2-3). The displacement components of an arbitrary particle located initially at \( (x, y, z) \) are denoted by \( u_x, u_y, u_z \). In Lagrangian description, the components of the Green’s strain tensor, referred to initial configuration of the plate, are expressed as follows:

\[
E_{xx} = \frac{\partial u_x}{\partial x} + \frac{1}{2}[ (\frac{\partial u_x}{\partial x})^2 + (\frac{\partial u_y}{\partial x})^2 + (\frac{\partial u_z}{\partial x})^2 ],
\]

\[
E_{yy} = \frac{\partial u_y}{\partial y} + \frac{1}{2}[ (\frac{\partial u_x}{\partial y})^2 + (\frac{\partial u_y}{\partial y})^2 + (\frac{\partial u_z}{\partial y})^2 ],
\]

\[
E_{xy} = \frac{1}{2}[ \frac{\partial u_x}{\partial x} \frac{\partial u_y}{\partial y} + \frac{\partial u_y}{\partial x} \frac{\partial u_x}{\partial y} + \frac{\partial u_z}{\partial x} \frac{\partial u_z}{\partial y} + \frac{\partial u_z}{\partial y} \frac{\partial u_z}{\partial x} ].
\]

According to the assumption H6 (Kirchhoff’s hypothesis), we have
\[ u_x = u(x, y) - z \frac{\partial w(x, y)}{\partial x}, \]
\[ u_y = v(x, y) - z \frac{\partial w(x, y)}{\partial y}, \]
\[ u_z = w(x, y), \]

where \( u, v, w \) are displacements at middle surface. Furthermore, according to H4, the higher powers of the derivatives of \( u \) and \( v \) are negligible. Thus, the strain-displacement relation expressed by Eq. 2-17 is reduced to

\[ E_{xx} = \frac{\partial u}{\partial x} - z \frac{\partial^2 w}{\partial x^2} + \frac{1}{2} \left( \frac{\partial w}{\partial x} \right)^2, \]
\[ E_{yy} = \frac{\partial v}{\partial y} - z \frac{\partial^2 w}{\partial y^2} + \frac{1}{2} \left( \frac{\partial w}{\partial y} \right)^2, \]
\[ E_{xy} = \frac{1}{2} \left[ \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} - 2z \frac{\partial^2 w}{\partial x \partial y} + \frac{\partial w}{\partial x} \frac{\partial w}{\partial y} \right]. \]

Therefore, von Karman’s large deflection theory of thin plate differs from the linear theory of thin plates only in retaining certain powers of derivatives \( \partial w/\partial x \) and \( \partial w/\partial y \) in the strain-displacement relationship. In addition, in the Lagrangian description, related to Green strain tensor, the second Piola-Kirchhoff stress tensor \( S_{ij} \) is used. According to H5, \( S_{ij} \) is expressed by \( E_{ij} \) through the Hooke’s law for a linearly elastic material as follows:

\[ S_{ij} = C_{ijkl} E_{kl} \cdot \]

For a linearly elastic and isotropic material, writing the individual components of Eq. 2-21 by the use of Eq. 2-19 and Hooke’s law lead to

\[ S_{xx} = \frac{E}{1 - v^2} \left[ \frac{\partial u}{\partial x} + v \frac{\partial v}{\partial y} - z \left( \frac{\partial^2 w}{\partial x^2} + v \frac{\partial^2 w}{\partial y^2} \right) + \frac{1}{2} \left( \frac{\partial w}{\partial x} \right)^2 + \frac{v}{2} \left( \frac{\partial w}{\partial y} \right)^2 \right], \]
\[
S_{yy} = \frac{E}{1-v^2} \left[ \frac{\partial v}{\partial y} + v \frac{\partial u}{\partial x} - z \left( \frac{\partial^2 w}{\partial y^2} + \nu \frac{\partial^2 w}{\partial x^2} \right) + \frac{1}{2} \left( \frac{\partial w}{\partial y} \right)^2 + \nu \left( \frac{\partial w}{\partial x} \right)^2 \right].
\]

\[
S_{xy} = G \left[ \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} - 2z \frac{\partial^2 w}{\partial x \partial y} + \frac{\partial w}{\partial x} \frac{\partial w}{\partial y} \right],
\]

where \( E \) and \( \nu \) are Young’s modulus and Poisson’s ratio. Based on the above stress components and referred to the initial configuration of unloaded plate, the stress resultants and the stress moments are defined by following expressions:

\[
N_x = \int_{-h/2}^{h/2} S_{xx} \, dz, \quad N_{xy} = \int_{-h/2}^{h/2} S_{xy} \, dz, \quad N_y = \int_{-h/2}^{h/2} S_{yy} \, dz,
\]

\[
M_x = \int_{-h/2}^{h/2} S_{xx} \, z \, dz, \quad M_{xy} = \int_{-h/2}^{h/2} S_{xy} \, z \, dz, \quad M_y = \int_{-h/2}^{h/2} S_{yy} \, z \, dz.
\]

By the institution of Eq. 2-20 into 2-22, we can express the resultants and moments in terms of three displacement components \( u, v \) and \( w \) as follows:

\[
N_x = N_x(u, v, w), \quad N_{xy} = N_{xy}(u, v, w), \quad N_y = N_y(u, v, w),
\]

\[
M_x = M_x(u, v, w), \quad M_{xy} = M_{xy}(u, v, w), \quad M_y = M_y(u, v, w).
\]

For an isotropic material, inserting Eq. 2-23 becomes

\[
N_x = \frac{Eh}{1-v^2} \left[ \frac{\partial u}{\partial x} + v \frac{\partial v}{\partial y} + \frac{1}{2} \left( \frac{\partial w}{\partial y} \right)^2 + \nu \left( \frac{\partial w}{\partial x} \right)^2 \right],
\]

\[
N_y = \frac{Eh}{1-v^2} \left[ \frac{\partial v}{\partial y} + \frac{\partial u}{\partial x} + \frac{1}{2} \left( \frac{\partial w}{\partial x} \right)^2 + \nu \left( \frac{\partial w}{\partial y} \right)^2 \right],
\]

\[
N_{xy} = \frac{1-v}{2} \frac{Eh}{1-v^2} \left[ \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} + \frac{\partial w}{\partial x} \frac{\partial w}{\partial y} \right],
\]

\[
M_x = -\frac{Eh^3}{12(1-\nu)} \left( \frac{\partial^2 w}{\partial x^2} + \nu \frac{\partial^2 w}{\partial y^2} \right),
\]

\[
M_y = -\frac{Eh^3}{12(1-\nu)} \left( \frac{\partial^2 w}{\partial y^2} + \nu \frac{\partial^2 w}{\partial x^2} \right),
\]
\[ M_{xy} = -\frac{Eh^3}{12} \frac{\partial^2 w}{\partial x \partial y}. \]

The equilibrium equations in Lagrangian description are expressed in terms of the second Piola-Kirchhoff stress tensor as follows:

\[
\frac{\partial}{\partial X_j} \left( S_{jk} \frac{\partial X_i}{\partial X_k} \right) + b_{0i} = 0, \tag{2-25}
\]

where \( b_{0i} \) is the body force per unit volume. Identifying \( X_1, X_2, X_3 \) with \( x, y, z \), identifying \( x_1, x_2, x_3 \) with \( x + u_x, y + u_y, z + u_z \), and neglecting any product terms that include \( u \) or \( v \), the equilibrium equations of Eq. 2-25 becomes

\[
\frac{\partial S_{xx}}{\partial x} + \frac{\partial S_{xy}}{\partial y} + \frac{\partial S_{xz}}{\partial z} + b_{0x} = 0,
\]

\[
\frac{\partial S_{xy}}{\partial x} + \frac{\partial S_{yy}}{\partial y} + \frac{\partial S_{yz}}{\partial z} + b_{0y} = 0, \tag{2-26}
\]

\[
\frac{\partial}{\partial x} \left( S_{xx} \frac{\partial w}{\partial x} + S_{xy} \frac{\partial w}{\partial y} + S_{xz} \right) + \frac{\partial}{\partial y} \left( S_{yx} \frac{\partial w}{\partial x} + S_{yy} \frac{\partial w}{\partial y} + S_{zy} \right) + \frac{\partial}{\partial z} \left( S_{zx} \frac{\partial w}{\partial x} + S_{zy} \frac{\partial w}{\partial y} + S_{zz} \right) + b_{0z} = 0.
\]

Multiplying the first two equations of Eq. 2-26 successively by \( dz \) and \( zdz \) and then integrating them from \(-h/2\) to \( h/2\) yield

\[
\frac{\partial N_x}{\partial x} + \frac{\partial N_y}{\partial y} + f_x = 0,
\]

\[
\frac{\partial N_y}{\partial x} + \frac{\partial N_y}{\partial y} + f_y = 0, \tag{2-27}
\]

\[
\frac{\partial M_x}{\partial x} + \frac{\partial M_y}{\partial y} - Q_x + m_x = 0,
\]
\[
\frac{\partial M_{sx}}{\partial x} + \frac{\partial M_{sy}}{\partial y} = Q_y + m_y = 0,
\]

with

\[
f_x = S_{zx} \bigg|_{-h/2}^{h/2} + \int_{-h/2}^{h/2} b_{ox} \, dz,
\]

\[
f_x = S_{zy} \bigg|_{-h/2}^{h/2} + \int_{-h/2}^{h/2} b_{oy} \, dz,
\]

\[
Q_x = \int_{-h/2}^{h/2} S_{xz} \, dz,
\]

\[
Q_y = \int_{-h/2}^{h/2} S_{zy} \, dz,
\]

\[
m_x = \frac{h}{2} S_{zx} \bigg|_{-h/2}^{h/2} + \int_{-h/2}^{h/2} zb_{ox} \, dz,
\]

\[
m_y = \frac{h}{2} S_{zy} \bigg|_{-h/2}^{h/2} + \int_{-h/2}^{h/2} zb_{oy} \, dz.
\]

Finally, integrating the third equation of Eq.2-23 with respect to \( z \) from \(-h/2\) to \( h/2\) gives

\[
\frac{\partial}{\partial x} \left( Q_x + N_x \frac{\partial w}{\partial x} + N_{sy} \frac{\partial w}{\partial y} \right) + \frac{\partial}{\partial y} \left( Q_y + N_{sx} \frac{\partial w}{\partial x} + N_y \frac{\partial w}{\partial y} \right) + q = 0,
\]

where \( q \) is the lateral load per unit area of undeformed middle surface and is expressed by

\[
q = \left( S_{zz} + S_{zx} \frac{\partial w}{\partial x} + S_{zy} \frac{\partial w}{\partial y} \right) \bigg|_{-h/2}^{h/2} + \int_{-h/2}^{h/2} b_{oz} \, dz.
\]

The first and the last terms on the right side of the above equation are the vertical load. The second and third terms represent the contributions to the lateral load due to shear acting on the surface that are rotated in the deformed position. By means of Eq. 2-24, Eq. 2-26 can be rewritten as:
\[
\frac{\partial^2 M_x}{\partial x^2} + 2 \frac{\partial^2 M_{xy}}{\partial x \partial y} + \frac{\partial^2 M_y}{\partial y^2} = -q - \frac{\partial m_x}{\partial x} - \frac{\partial m_y}{\partial y} - N_x \frac{\partial^2 w}{\partial x^2}
\]

2-31

\[
-2 N_{xy} \frac{\partial^2 w}{\partial x \partial y} - N_y \frac{\partial^2 w}{\partial y^2} + f_x \frac{\partial w}{\partial x} + f_y \frac{\partial w}{\partial y}.
\]

Inserting the resultants and moments into the first two equations of Eq. 2-27 and Eq. 2-31 can yield immediately basic three equilibrium equations in terms of \(u, v, w\). For an isotropic material, these three equilibrium equations become:

\[
\frac{\partial}{\partial x} \left[ \frac{\partial u}{\partial x} + v \frac{\partial v}{\partial y} + \frac{1}{2} \left( \frac{\partial w}{\partial x} \right)^2 + \frac{\nu}{2} \left( \frac{\partial w}{\partial y} \right)^2 \right] + \frac{1 - \nu^2}{Eh} \frac{\partial}{\partial x} \frac{\partial u}{\partial x} = 0,
\]

2-32

\[
\frac{\partial}{\partial y} \left[ \frac{\partial v}{\partial x} + \frac{\partial u}{\partial y} + \frac{1}{2} \left( \frac{\partial w}{\partial y} \right)^2 + \frac{\nu}{2} \left( \frac{\partial w}{\partial x} \right)^2 \right] + \frac{1 - \nu^2}{Eh} \frac{\partial}{\partial x} \frac{\partial v}{\partial y} = 0,
\]

\[
\frac{\partial^4 w}{\partial x^4} + 2 \frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4} = \frac{1 - \nu^2}{Eh^3} \left[ q + \frac{\partial m_x}{\partial x} + \frac{\partial m_y}{\partial y} + N_x \frac{\partial^2 w}{\partial x^2} \right]
\]

+ \frac{2N_{xy}}{\partial \partial x \partial y} + N_y \frac{\partial^2 w}{\partial y^2} - f_x \frac{\partial w}{\partial x} - f_y \frac{\partial w}{\partial y}.
\]

In the third equation of Eq. 2-32 the resultants \(N_x, N_{xy}, N_y\) are expressed in terms of \(u, v, w\), referring to Eq. 2-24. Alternatively, a stress function \(F(x, y)\) can be introduced by

\[
N_x = \frac{\partial^2 F}{\partial y^2} - \int f_x (x, y) dx, \quad N_y = \frac{\partial^2 F}{\partial x^2} - \int f_y (x, y) dy, \quad N_{sy} = -\frac{\partial^2 F}{\partial x \partial y}.
\]

2-33

These resultants expressed by the stress function \(F(x, y)\) satisfy the first two equations of Eq. 2-27 identically. Furthermore, eliminating \(u, v\) from Eq. 2-24 and using the above Eq. 2-33 give the compatibility condition as follows:
\[
\frac{\partial^4 F}{\partial x^4} + 2 \frac{\partial^4 F}{\partial x^2 \partial y^2} + \frac{\partial^4 F}{\partial y^4} = Eh \left\{ \left( \frac{\partial^2 w}{\partial x \partial y} \right)^2 - \frac{\partial^2 w}{\partial x^2} \frac{\partial^2 w}{\partial y^2} \right\} + \frac{\partial^2}{\partial x^2} \left[ \int f_x(x, y) dx \right] - \nu \int f_y(x, y) dy + \frac{\partial^2}{\partial x^2} \left[ \int f_x(x, y) dx - \nu \int f_y(x, y) dy \right].
\]

These two equations of Eq. 2-34 and the third equation of Eq. 2-32 are famous von Karman equations for large deflection of plates. In most practical problems, \( f_x, f_y, m_x, m_y \) are zero, these two equations then reduce to

\[
\frac{\partial^4 w}{\partial x^4} + 2 \frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4} = \frac{1 - \nu^2}{Eh} \left[ q + \frac{\partial^2 F \partial^3 w}{\partial y^2 \partial x^2} - 2 \frac{\partial^2 F \partial^2 w}{\partial x \partial y \partial x \partial y} + \frac{\partial^2 F \partial^3 w}{\partial x^2 \partial y^2} \right]
\]

\[
\frac{\partial^4 F}{\partial x^4} + 2 \frac{\partial^4 F}{\partial x^2 \partial y^2} + \frac{\partial^4 F}{\partial y^4} = Eh \left\{ \left( \frac{\partial^2 w}{\partial x \partial y} \right)^2 - \frac{\partial^2 w}{\partial x^2} \frac{\partial^2 w}{\partial y^2} \right\}
\]

The most important features of the above two equations are that they are coupled and nonlinear. When the nonlinear terms are neglected, these equations reduce to the corresponding equations of the small deflection theory.

It is pointed that the above equations of large deflection theory are not directly used in the numerical analysis based on the finite element method (FEM). Alternatively, in order to construct the basic equations in FEM [32], the principle of virtual work expressed by

\[
\int_{V_0} S_{ij} \delta E_{ij} dV = \int_{V_0} b_{0i} \delta u_i dV + \int_{A_0} T_{0i} \delta u_i dA
\]

is used. Where, \( V_0 \) is the volume of the body, \( b_{0i} \) are the body force components, \( T_{0i} \) are the prescribed external force components in initial configuration. The details of FEM formulation are omitted here.
2.3 Approximate method

In this section, a theoretical formulation to describe the relationship between the normal drag coefficient of a rigid curved-plate and the configuration of the plate is first derived. Then, based on the theoretical formulation, an approximate numerical method is constructed to evaluate the normal force acting on a flexible plate and the deformation of the plate using only the iteration of structural mechanics analysis rather than conventional complex iterations of fluid-structure coupling analysis.

2.3.1 Theoretical modeling

A flexible plate with upper end fixed normal to the wind flow is considered, as shown in Fig. 2-4a. The length, width, and thickness are denoted by \( L \), \( W \), and \( T \), respectively. The plate subjected to wind pressure tends to bend towards the flow direction to reduce drag. Dashed line denotes the plate before deformation. The displacements of free end are denoted by \( u_x \) and \( u_y \). The deformed configuration is described by the chord angle \( \tilde{\theta} \) between the vertical line and the line connecting the fixed end and the free end of the plate. In general, this is a complex fluid-structure coupling problem, the pressure acting on the plate is complicatedly distributed on the plate surfaces and fluctuates around its mean value over time [11, 13, 20]. In present study, as the first approximation, it is assumed that the deformed plate in uniform and steady wind flow is under a quasi-static state and that the mean pressure is uniformly distributed on the plate surface. Therefore, the deformed plate under the quasi-static state in the wind flow can be considered as an equivalent rigid curved-plate. Furthermore, from Gosselin et al. [7], the skin friction drag on a rigid plate parallel to the flow is 2 or 3 orders of magnitude smaller than the pressure drag measured.
on a rigid plate perpendicular to the flow. Therefore, the effect of the viscous drag along
the tangential direction of the plate surface due to the fluid viscosity on the deformation of
the plate is considered to be very slight compared to the wind pressure normal to the surface
so that only the normal force is considered in the present study. In addition, for the sake of
simplicity, the effect of gravity on the deformation of plate is not considered hereafter.

\[ V_N(\theta) = V \cos \theta \]
\[ V_T(\theta) = V \sin \theta \]

**Figure 2-4:** The schematic of a flexible plate normal to wind flow.

Based on the above assumptions, consider the wind flow acting on an arbitrary
infinitesimal element $ds$ of a rigid curved-plate, as shown in Fig. 2-4b. $V$ is the wind
velocity. $V_N$ and $V_T$ are the velocity components normal to the curved surface and along
the tangential direction of the curved surface, and $s$ is the Lagrangian coordinate defined
along the mid-plane of the plate from its fixed end to its free end, respectively. Therefore,
the total normal force produced by the wind flow with velocity $V$ on the curved plate can be expressed by:

$$ F_{N\text{-theory}} = \frac{1}{2} \int_{0}^{L} \rho (V \times \cos \theta)^2 WC_D ds. $$  \hspace{1cm} 2-38

where $\rho$ is the density of air, $\theta$ is the tangential angle of the curved plate at location $s$, and $C_D$ is the drag coefficient of an equivalent rigid flat-plate ($L \times W \times T$) normal to the flow. $C_D$ can be find from books of fluid mechanics [26] for regular 3D plates or can be obtained from fluid dynamic analysis using a CFD codes. Generally, $\theta (\theta(F_{N\text{-theory}}, s))$ is a function of the normal force and the location $s$ along the curved plate, and that this equation is actually an integral equation of fluid-structure coupling. It is difficult to obtain the exact solution. In order to express the total normal force $F_{N\text{-theory}}$ in an explicit formulation, we assume that the total normal force $F_{N\text{-theory}}$ can be approximately expressed by the following:

$$ F_{N\text{-theory}}(\tilde{\theta}) = \frac{1}{2} \rho V^2 AC_D f(\tilde{\theta}) \cos^2 \tilde{\theta}, \quad A = L \times W, $$ \hspace{1cm} 2-39

where $A$ denotes the area of the plate, $\tilde{\theta}$ denotes the chord angle of the curved plate (Fig. 2-4a) and is used to characterize the configuration of the curved plate, and $f(\tilde{\theta})$ is a correction function, which is used to correct the errors caused by the above simplification because Eq. 2-39 is not a rigorous solution of the integral equation of Eq. 2-38. Then the average pressure $p_{\text{theory}}$ acting on the curved plate can be calculated by the following equation:

$$ p_{\text{theory}} = \frac{F_N(\tilde{\theta})}{A} = \frac{1}{2} \rho V^2 C_D f(\tilde{\theta}) \cos^2 \tilde{\theta}. $$ \hspace{1cm} 2-40
From Eq. 2-40, the normal drag coefficient of the curved plate can be expressed as follows:

\[ C_{N\text{-theory}}(\tilde{\theta}) = \frac{p_{\text{theory}}(\tilde{\theta})}{0.5 \rho V^2} = \frac{F_N(\tilde{\theta})}{0.5 \rho V^2 A} = C_D f(\tilde{\theta}) \cos^2 \tilde{\theta}. \]  \hspace{1cm} 2-41

And the conventional drag coefficient is expressed by the following:

\[ C_{d\text{-theory}}(\tilde{\theta}) = \frac{F_d(\tilde{\theta})}{0.5 \rho V^2 A} = \frac{F_N(\tilde{\theta}) \times \cos \tilde{\theta}}{0.5 \rho V^2 A} = C_{N\text{-theory}}(\tilde{\theta}) \times \cos \tilde{\theta} = C_D f(\tilde{\theta}) \cos^3 \tilde{\theta}. \]  \hspace{1cm} 2-42

Observing Eq. 2-41 and Fig. 2-4a, it is recognized that \( \tilde{\theta} = 0 \) and \( \tilde{\theta} = \pi / 2 \) correspond to the two special cases of a rigid flat-plate normal and parallel to the wind flow, respectively. Therefore, according to fluid mechanics, the normal drag coefficient \( C_{N\text{-theory}}(\tilde{\theta}) \) should satisfy \( C_{N\text{-theory}}(0) = C_D \) and \( C_{N\text{-theory}}(\pi / 2) = C_D \). The correction function \( f(\tilde{\theta}) \) should satisfy \( f(0) = 1 \) and \( f(\pi / 2) < \infty \). As a result, \( f(\tilde{\theta}) \) is assumed here by the following equation:

\[ f(\tilde{\theta}) = 1 + \beta \sin^2 \tilde{\theta}, \]  \hspace{1cm} 2-43

where parameter \( \beta \) is a constant, which is determined from the comparison between the curve \( C_{N\text{-theory}}(\tilde{\theta}) \) obtained from Eq. 2-41 and the curve \( C_{N\text{-exp}}(\tilde{\theta}) \) obtained from experiment or \( C_{N\text{-CFD}}(\tilde{\theta}) \) obtained from a series of CFD calculations of rigid curved-plates with chord angles \( \tilde{\theta}_i (0 < \tilde{\theta}_i < \pi / 2; i = 1, 2, \cdots, k) \). In the present study, CFD calculations are employed to determine the curve \( C_{N\text{-CFD}}(\tilde{\theta}) \). The geometries of rigid curved-plates with chord angles \( \tilde{\theta}_i (i = 1, 2, \cdots, k) \) used in the CFD calculations are determined from a series of structure calculations of a flexible plate subjected to a series of uniform pressures as follows.
2.3.2 Determination of $f(\tilde{\theta})$

In order to determine the correction function $f(\tilde{\theta})$, a series of numerical calculations of structural mechanics and CFD calculations are conducted using commercially available codes of MSC Marc2010 and ANSYS Fluent 13.0, respectively. First, a flexible plate subjected to a series of uniform pressures are conducted to determine a series of related reconfigurations of the plate. Large deformation, namely, the geometrical nonlinearity of the deformation is considered. The pressure applied to the flexible plate is given by the following equation:

$$p_i = \frac{1}{2} \rho V_i^2 C_D, \quad (0 < V_i \leq V_{\max}; i = 1, 2, \cdots, k),$$  \hspace{1cm} (2-44)

where $V_i (i = 1, 2, \cdots, k)$ denote a series of given wind velocities, $V_{\max}$ is the maximum velocity specified according to the design requirement of the flexible plate, and $C_D$ is the drag coefficient of a rigid flat-plate normal to the flow as mentioned earlier. Then, applying $p_i (1, 2, \cdots, k)$ of Eq. 2-44 to a given flexible plate and conducting the analysis of structural mechanics, we can obtain a series of self-similar geometries $\tilde{\theta}_i (i = 1, 2, \cdots, k)$ of curved plates. According to the experimental facts reported in many references as mentioned in introduction, it is well known that the value of the real normal drag coefficient $C_{N_i-\text{real}}$ of a flexible flat-plate normal to the flow is always smaller than $C_D$, because the deformation of the flexible plate reduces the drag force. Therefore, for a given $V_i$, the real averaged pressure $p_{i-\text{real}}$ acting on a flexible plate is always lower than $p_i$ calculated by Eq. 2-44 because of that $C_{N_i-\text{real}} < C_D$. In other words, the $p_i$ gives the upper bound of $p_{i-\text{real}}$ for
Similarly, the chord angle \( \tilde{\theta}_i \) of the deformed plate, obtained from the above structural analysis related to \( p_i \), is also not equal to the real chord angle \( \tilde{\theta}_{i\text{-real}} \) for a given \( V_i \). The real chord angle \( \tilde{\theta}_{i\text{-real}} \) is always smaller than \( \tilde{\theta}_i \), namely \( 0 < \tilde{\theta}_{i\text{-real}} < \tilde{\theta}_i \) because of that \( 0 < p_{i\text{-real}} < p_i \) for a given \( V_i \). Therefore, \( \tilde{\theta}_i \) also gives the upper bound of \( \tilde{\theta}_{i\text{-real}} \) for a given \( V_i \).

Secondly, we use these curved plates with chord angles \( \tilde{\theta}_i \) \((i = 1, 2, \ldots, k)\) as a series of rigid curved-plates in the CFD calculations to solve the corresponding normal drag coefficients \( C_{Ni\text{-CFD}}(\tilde{\theta}_i) \) \((i = 1, 2, \ldots, k)\). That is, we obtain a curve of \( C_{Ni\text{-CFD}}(\tilde{\theta}_i) \) \((i = 1, 2, \ldots, k)\) related to a series of rigid curved-plates with chord angles \( \tilde{\theta}_i \) \((i = 1, 2, \ldots, k)\). On the other hand, according to Eqs. 2-41 and 2-43 with \( \beta = 0 \), we can obtain the theoretical formulation of normal drag coefficient \( C_{N\text{-theory}}(\tilde{\theta})\big|_{\beta=0} \). In consequence, we obtain two curves of \( C_{N\text{-CFD}}(\tilde{\theta}_i) \) and \( C_{N\text{-theory}}(\tilde{\theta})\big|_{\beta=0} \). Plotting these two curves together and comparing them with each other reveal the deference between these two curves. Finally, a proper value of \( \beta \) is selected through a process of trial and error to make the difference between these two curves as small as possible. That is, \( \beta \) is determined to make the following equation:

\[
C_{N\text{-theory}}(\tilde{\theta})\big|_{\beta} = C_D f(\tilde{\theta})\big|_{\beta} \cos^2 \tilde{\theta} \approx C_{N\text{-CFD}}(\tilde{\theta}). \tag{2-45}
\]
2.3.3 Algorithm to solve the average pressure acting on a flexible plate normal to the flow

Assuming that the geometry and material properties of a flexible plate are known, the velocity of the uniform and steady wind inflow is $V_0$, and the plate normal to the flow is at a quasi-static state. Then, the real average pressure in the sense of FSI can be expressed by the following equation:
$p_{-real} = p_{-theory}(\tilde{\theta}_{bend}), \quad 2-46a$

if the following equation is satisfied.

$p_{-theory}(\tilde{\theta}_{bend}) = p_{-bend}, \quad 2-46b$

where $p_{-bend}$ denotes the pressure applied to the flexible plate in the structural analysis, $\tilde{\theta}_{bend}$ is the corresponding chord angle of the deformed plate, and $p_{-theory}(\tilde{\theta}_{bend})$ is the theoretical average pressure obtained from Eq. 2-40. It is noted that the theoretical average pressure is approximately equal to the average pressure obtained from the CFD analysis of a rigid curved-plate with the chord angle $\tilde{\theta}_{bend}$, based on the preceding theoretical modeling. In other words, Eq. 2-46a means that the real pressure equals the theoretical pressure obtained from Eq. 2-40, when the pressure applied to the flexible plate in the structure calculation equals the theoretical pressure acting on the rigid curved-plate with the chord angle $\tilde{\theta}_{bend}$. Then, based on Eq. 2-46, an algorithm is developed to solve the real pressure $p_{i-real}$, chord angle $\tilde{\theta}_{i-real}$, and normal drag coefficient $C_{N-real}(\tilde{\theta}_{i-real})$ for a flexible plate normal to the wind flow of $V_0$, using only iterative simulations of structural analysis as follows. $V_0$ is an arbitrary given wind velocity.

The flowchart of calculation procedures of the approximate method is illustrated in Fig. 2-6. At the first iteration, nonlinear bending calculation of the flexible plate subjected to a given pressure is conducted. The initial pressure is calculated by Eq. 2-44, that is:

$p_{1-bend} = \frac{1}{2} \rho V_0^2 \times C_D, \quad 2-47$
where the drag coefficient $C_D$ of a rigid plate normal to the wind flow is obtained from the CFD analysis for the rigid plate with the same geometry as the flexible plate. Consequently, a reconfiguration $\tilde{\theta}_{\text{bend}}$ of the flexible plate is obtained from the bending analysis. Inserting this $\tilde{\theta}_{\text{bend}}$ into Eqs. 2-40 and 2-41 produces $p_{1\_\text{theory}}(\tilde{\theta}_{\text{bend}})$ and $C_{N1\_\text{theory}}(\tilde{\theta}_{\text{bend}})$, respectively. According to the preceding subsections, $p_{1\_\text{bend}}$ and $\tilde{\theta}_{\text{bend}}$ give the upper bounds of the real pressure $p_{V_0\_\text{real}}$ and real reconfiguration $\tilde{\theta}_{V_0\_\text{real}}$, respectively. Therefore, $p_{1\_\text{theory}}$ obtained from Eq. 2-50 using $\tilde{\theta}_{\text{bend}}$ gives the lower bound of the real pressure. It is obvious that $p_{1\_\text{bend}}$ is not equal to $p_{1\_\text{theory}}$. Thus, we move to the second iteration using $p_{1\_\text{theory}}$ as the bending load applied to the flexible plate as follows.

$$p_{2\_\text{bend}} = p_{1\_\text{theory}}. \quad 2-48$$

Similar to the calculation procedures at the first iteration, $\tilde{\theta}_{2\_\text{bend}}$ is obtained from the bending calculation, and then $p_{2\_\text{theory}}(\tilde{\theta}_{2\_\text{bend}})$ and $C_{N2\_\text{theory}}(\tilde{\theta}_{2\_\text{bend}})$ can be obtained from Eqs. 2-40 and 2-41. It is observed that

$$\therefore \quad \tilde{\theta}_{1\_\text{bend}} > \tilde{\theta}_{2\_\text{bend}} > 0$$
$$\therefore \quad C_{N1\_\text{theory}}(\tilde{\theta}_{1\_\text{bend}}) < C_{N2\_\text{theory}}(\tilde{\theta}_{2\_\text{bend}}) < C_D \quad 2-49$$
$$\therefore \quad p_{1\_\text{bend}} > p_{2\_\text{theory}}(\tilde{\theta}_{2\_\text{bend}}) > p_{2\_\text{bend}}.$$
For a flexible plate normal to wind flow of given velocity $V_0$

\[ p_{1\text{-bend}} = 0.5 \rho V_0^2 C_D \]
\[ (p_{1\text{-bend}} > p_{V_0\text{-real}}) \]
\[ p_{k\text{-bend}} = p_{k\text{-bend}} \]

Applying $p_{k\text{-bend}} (k=1,2,...)$ to the flexible plate to solve $\tilde{\theta}_{k\text{-bend}}$ by nonlinear FEM structural analysis

By using Eqs. (2-40) and (2-43)

\[ \tilde{\theta}_{k\text{-bend}} \Rightarrow p_{k\text{-theory}}(\tilde{\theta}_{k\text{-bend}}) \]

\[ \Delta p_k = p_{k\text{-bend}} - p_{k\text{-theory}}(\tilde{\theta}_{k\text{-bend}}) \]
\[ \Delta p_k \leq 0.5\%? \]

Yes \quad END

No

\[ p_{k\text{-bend}} = p_{(k+1)\text{-bend}} \]

\[ p_{(k+1)\text{-bend}} = p_{k\text{-theory}} \]

Figure 2-6: Flowchart of the calculation sequence of the present approximate method.

Assuming that $p_{2\text{-theory}}(\tilde{\theta}_{2\text{-bend}})$ is still not equal to $p_{2\text{-bend}}$, and that the difference between the both values is still significant, we move to the third iteration using the following equation:

\[ p_{3\text{-bend}} = p_{2\text{-theory}} \]

2-50

as the pressure applied to the flexible plate in the bending calculation of the third iteration.

Similar to preceding iterations, $\tilde{\theta}_{3\text{-bend}}$ is obtained from bending calculation, and then a set of $p_{3\text{-theory}}(\tilde{\theta}_{3\text{-bend}})$, and $C_{N3\text{-theory}}(\tilde{\theta}_{3\text{-bend}})$ are obtained using Eqs. 2-40 and 2-41, respectively. It is observed that:
\[ \therefore \tilde{\theta}_{1\text{-bend}} > \tilde{\theta}_{3\text{-bend}} > \tilde{\theta}_{2\text{-bend}} \]
\[ \therefore C_{N1\text{-theory}}(\tilde{\theta}_{1\text{-bend}}) < C_{N3\text{-theory}}(\tilde{\theta}_{3\text{-bend}}) < C_{N2\text{-theory}}(\tilde{\theta}_{2\text{-bend}}) \]
\[ \therefore p_{1\text{-bend}} > p_{3\text{-bend}} > p_{3\text{-theory}}(\tilde{\theta}_{3\text{-bend}}) > p_{2\text{-bend}}. \]

We must move to the next iteration if the difference between \( p_{3\text{-bend}} \) and \( p_{3\text{-theory}}(\tilde{\theta}_{3\text{-bend}}) \) is still significant. Similar to the preceding iterations, iteratively repeat the computational procedures as described above until the difference between the theoretical pressure \( p_{k\text{-theory}}(\tilde{\theta}_{k\text{-bend}}) \) obtained from Eq. 2-40 and the pressure \( p_{k\text{-bend}} \) applied to the flexible plate in the bending calculation at the \( k \)-th iteration is equal to or less than a given small value. In this study, the iterative calculation is completed when this difference satisfies the following criterion:

\[
\Delta p = \frac{|p_{k\text{-theory}} - p_{k\text{-bend}}|}{|p_{k\text{-theory}}|} = \frac{|p_{k\text{-theory}} - p_{k-1\text{-theory}}|}{|p_{k\text{-theory}}|} \leq 0.5\%. \tag{2-52}
\]

Therefore, based on Eq. 2-52 the real pressure acting on the given flexible plate normal to the wind flow of \( V_{\alpha} \) is defined by following equation:

\[
p_{V_{\alpha}\text{-real}} = p_{k\text{-theory}}. \tag{2-53}
\]

### 2.4 Numerical simulation tests

In order to demonstrate the validity of the proposed approximate method, numerical simulation tests for several 3D flexible plates with different geometries and different material moduli are conducted using the approximate method. Furthermore, simulation for a 3D flexible plate studied by Gosselin et al. \[7\] is also carried out for a comparison.
2.4.1 Simulation tests for five flexible plates

A 3D model of flexible plates normal to the wind flow is depicted in Fig. 2-7. The upper end of the model is clamped and the lower end is free. The length, width and thickness of the plate model are denoted by $L$, $W$ and $T$, respectively. Five plates with different lengths or different material constants are used in the simulation tests. Tables 2-1 and 2-2 give the geometries of five plates and the material constants of three kinds of materials. PP, PE and PET are the abbreviations of polypropylene, polyethylene and polyethylene terephthalate.

Figure 2-7: A 3D simulation model.
Table 2-1: Geometries of five flexible thin plates.

<table>
<thead>
<tr>
<th>Plate</th>
<th>L (mm)</th>
<th>W (mm)</th>
<th>T (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>PP-100</td>
<td>100</td>
<td>100</td>
<td>1</td>
</tr>
<tr>
<td>PP-300</td>
<td>300</td>
<td>100</td>
<td>1</td>
</tr>
<tr>
<td>PP-500</td>
<td>500</td>
<td>100</td>
<td>1</td>
</tr>
<tr>
<td>PE-300</td>
<td>300</td>
<td>100</td>
<td>1</td>
</tr>
<tr>
<td>PET-300</td>
<td>300</td>
<td>100</td>
<td>1</td>
</tr>
</tbody>
</table>

Table 2-2: Material constants of three materials.

<table>
<thead>
<tr>
<th>Material</th>
<th>Young's Modulus (GPa)</th>
<th>Poisson’s ratio</th>
<th>Density ($10^3$kg/m$^3$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>PE</td>
<td>0.392266</td>
<td>0.41</td>
<td>0.91</td>
</tr>
<tr>
<td>PP</td>
<td>1.569064</td>
<td>0.41</td>
<td>0.91</td>
</tr>
<tr>
<td>PET</td>
<td>3.010642</td>
<td>0.39</td>
<td>1.39</td>
</tr>
</tbody>
</table>

Following the computational procedures described in Section 2.3, the determination of the theoretical curve of $C_{N-theory}(\bar{\theta})_\beta \approx C_{N-CFD}(\bar{\theta})$ is first performed based on the theoretical modeling with the aid of a series of structural and CFD analyses. Subsequently, iterative calculations of bending analysis are conducted to solve the real averaged pressure acting on the flexible plate and the deformation of the plate for a series of given wind velocities, according to the algorithm of the approximate method. Both of structure and
CFD analyses are ordinary calculations with definite boundary conditions and are relatively simple compared to complex conventional iterative fluid-structure coupling calculations.

**Figure 2-8:** (a) CFD computational domain and (b) grid around the plate.

In the CFD analysis, the fluid flow is assumed to be uniform and incompressible. The computation domain of CFD analysis is 42L×24L×11W with the origin of the coordinate located at the center of the plate, as shown in Fig. 2-8. The inlet, top, and bottom boundary walls are set at the distance of 12L from the center of the plate, respectively. The outlet is set at 30L downstream from the plate center. A uniform velocity flow is specified at the inlet, zero pressure is specified at outlet, slip condition is specified along the top and bottom boundary walls, and no-slip boundary are specified on the plate surface. The large eddy simulation method is utilized in the CFD analysis. According to ANSYS Fluent 13.0 User’s Guide (2010), in present CFD analysis, the Reynolds number $Re$, the turbulence intensity $I$, the turbulent kinetic energy $k$, and the turbulent length scale $l$ are calculated.
Effect of the number of grids on the calculation accuracy is investigated first using the geometry of PP-500 plate with different numbers of grids from 1,133,500 to 5,269,000. The plate is assumed to be rigid and normal to the wind flow. The drag coefficient is calculated for each case of grid number. As the result, the number of grids around 2,500,000 is adopted for all the plates based on the considerations of acceptable computational accuracy and cost. Fig. 2-8b demonstrates a typical grid distribution around the rigid curved-plate of PP-300 and the total number of grids of the computational domain is 2,266,000.

2.4.2 A comparative simulation test

A comparative simulation test of a 3D flexible plate used in a previous wind tunnel experiment by Gosselin et al. [7], as shown in Fig. 2-9, is also conducted based on the present approximate method to confirm the validity of the approximate method. The length and width of the plate are 100 mm and 35 mm, respectively. The flexural rigidity of the

![Figure 2-9: Simulation model for the comparative test.](image)
plate is $10^{-6}$ Nm. The Young’s modulus $E = 3.1$ GPa and Poisson’s ratio $\nu = 0.3$ are used. In order to keep the same flexural rigidity as that used in the experiment of Gosselin et al. [7], the thickness $h$ of the plate is 0.0152 mm determined from the following equation:

$$\frac{Eh^3}{12(1-\nu^2)} = 10^{-6} \text{ (Nm)}.$$  \hfill (2-54)

The computational domain for the comparative simulation test is a square section duct with 0.180 m in width and 4 m in length according to the experiment of Gosselin et al. [7]. The boundary conditions on the inlet, outlet, plate surface and boundary walls are the same as described in Fig. 2-8. The calculation procedures of the simulation test are the same as those described in subsection 2.3.

### 2.5 Results and discussion

Numerical results of simulation tests are presented in the following figures. The effect of the number of grids on the drag coefficient of a 3D vertical rigid plate of PP-500 are described in Fig. 2-10. It is seen that the drag coefficient varies slowly and tends to converge a stable value when the number of grids is larger than 2,000,000. As the result, the grid number around 2,500,000 was adopted for the CFD analyses of all plates based on the considerations of acceptable computational accuracy and cost.
Figure 2-10: Effect of the number of grids on the calculation results.

Typical velocity contour and velocity vector around a rigid plate normal to the flow are shown in Fig. 2-11. Nearly symmetric velocity field is observed around the vertical plate, and asymmetrically separated flow starts at the far wake from the plate. On the other hand, the velocity field is asymmetric around the rigid curved-plate as shown in Fig. 2-12. A long and narrow low velocity region is formed near the rear of the curved-plate.

Figure 2-11: Velocity contour and velocity vector around a vertical rigid plate with length of 500mm at 10 m/s: (a) velocity contour and (b) velocity vector.
Following the analysis procedures as described in section 2.3, the theoretical curve $C_{N_{theory}}(\bar{\theta})$ with $\beta = 0.5$ is determined with the aid of structure and CFD analyses of five flexible plates listed in Table 2-1. The curve and the normal force coefficients obtained from the CFD analyses of five plates are presented in Fig. 2-13. The normal force coefficients obtained from the CFD analyses of five plates are displayed by five kinds of marked points and the solid curve is the theoretical curve. It is obvious that the theoretical curve agrees well with the results obtained from the CFD analyses of five plates with different length or different modulus, although relatively large difference can be found for large chord angle beyond 60 degree. Errors between CFD results and theoretical curve are less than 7% for the chord angles smaller than 60 degree and over 10% for chord angles larger than 60 degree. Therefore, it is reasonable that using the proposed theoretical modeling method to predict the relationship between the normal force coefficient and the chord angle of rigid curved-plate.
We can calculate the normal force acting on a rigid curved-plate based on the theoretical curve if we know the chord angle of the curved-plate. In the case of large chord angle, which means large bending deformation of the plate, the theoretical curve overestimates the normal force. For a more accurate prediction of the normal force in the chord angle range of larger than 60 degree, a different parameter $\beta$ may be selected. That is, the theoretical curve may be defined in two ranges of chord angle: $0 \leq \theta < 60$ and $60 \leq \theta \leq 90$, selecting different $\beta$ for different range. This issue is a further research subject, which is ongoing now.

Results of the normal forces acting on the flexible plates and the deformations of the plates are presented versus to wind velocities in following figures. Effect of the plate length on the normal force acting on the three flexible plates at various wind velocities are
The CFD analysis results of normal force acting on the plates PP-100, PP-300 and PP-500 are compared with those obtained from the present approximate method. Here the results of CFD analyses are obtained from the analysis of 3D rigid curved-plates, which have the same geometries obtained from the approximate method. An
enlarged draft for the data at low wind velocities is also presented below. Marked points denote the CFD results and solid, dashed and dotted curves denote the results of approximate method. It is seen that, when the wind velocity is below 5 m/s, the longest plate of PP-500 gives the largest normal force because of its largest area normal to the wind flow. However, because the wind velocity increases, the normal forces acting on the relatively short plates of PP-100 and PP-300 increase more rapidly and successively exceed the normal force acting on the longest plate of PP-500. The normal force acting on the shortest plate of PP-100 has the largest value at the wind velocities higher than 22 m/s and the longest plate of PP-500 gives the smallest one.

Furthermore, it is seen that both results obtained from the present approximate method and the CFD analyses are in good agreement, especially in the case of PP-100 plate. In the cases of PP-300 and PP-500 plates, the difference between both results increases with the increase of wind velocity. The largest errors for PP-300 and PP-500 plates are 10.2% and 16.8% at 30 m/s, respectively. These features reflect the effect of the bending deformations of flexible plates on the normal force acting on the plates. Longer plate is easily bended by the wind flow than shorter one at the same wind velocity. Larger bending chord angle leads to lower normal force acting on the bended plate. For a more accurate prediction of the normal force acting on the flexible plate with a large chord angle, a different parameter $\beta$ may be selected, as mentioned above. In addition, the average pressure acting on the plates can be calculated by the normal force divided by the plate area, referring to Eq. 2-40.

Effect of material modulus on the normal force acting on the three flexible plates with the same geometry but different moduli is described in Fig. 2-15. The normal forces acting on a rigid plate having the same geometry are also depicted in the figure for a comparison.
Similar to Fig. 2-14, the results obtained from the present approximate method agree well with those obtained from the CFD analyses. The normal force acting on the PET-300 which has the highest modulus increases quickly and shows small error of 1.4% at 30 m/s. In contrast, the normal force acting on the PE-300 which has the lowest modulus increases slowly and shows relatively large error of 35% at 30 m/s. Plate with high modulus is hard to be bended by wind flow compared to the plate with low modulus. Large bending deformation reduces the normal force acting on the plate. These results are in consistence with the nature phenomena frequently observed from the bended trees under high wind.

The chord angles of five bended flexible plates at various wind velocities, obtained from the present approximate method, are described in Fig. 2-16. It is clear that the flexible plate with low modulus or large length has larger chord angle which means large bending deformation.

**Figure 2-15:** Effect of material modulus on the normal force acting on the flexible plates at various wind velocities.
Figure 2-16: Chord angles of five flexible plates at various wind velocities.

Figure 2-17: Comparison of drag force obtained from the previous experiment and the present approximate method.

Finally, the results of drag force obtained from the comparative simulation are compared with the previous experimental results [7] in Fig. 2-17. The drag force of a rigid
and flat plate obtained from the previous experiment is also depicted for a contrast. The drag values obtained from the present approximate method are well consistent with the previous experiment results and the errors are less than or around 10%. These comparative simulation results further confirm the validity and accuracy of the present approximate method.

2.6 Summary

Based on the theoretical modeling, the construction of the approximate method, and the numerical simulation tests, this study leads to following conclusions are obtained.

1. An explicit theoretical formulation for the prediction of the curve of the normal drag coefficient of a rigid curved-plate versus to its chord angle is derived through a theoretical modeling with the aid of a series of ordinary numerical analyses of nonlinear structure mechanics and fluid dynamics. The formulation is simple and gives good accuracy of prediction. Further study is needed to improve the prediction accuracy by selecting different correction parameters for different chord angle ranges.

2. An approximate method is constructed for evaluating the normal force acting on a flexible flat-plate normal to the wind flow and the deformation of the plate using the present theoretical formulation and the iteration of structural mechanics analysis rather than conventional complex iterations of fluid-structure coupling analysis.

3. Results of the numerical simulation tests for the plates with different geometries and material moduli demonstrate the validity and accuracy of the approximate method. Moreover, the comparative simulation test of a 3D flexible plate used in a previous wind tunnel experiment further confirms the accuracy of the present approximate method.
Therefore, it is considered that the present approximate method is relatively simple compared to conventional complex fluid-structure coupling analysis and is useful for the evaluation of the normal force acting on the flexible plate normal to the wind flow and the deformation of the plate in the practice applications.
Bibliography


CHAPTER 3

Upgrading a shrouded wind turbine with a self-adaptive flanged diffuser

In this chapter, a self-adaptive flange is proposed for the wind turbine shrouded by a flanged diffuser to reduce the wind loads acting on the flanged diffuser at high wind velocities. The self-adaptive flange can maintain the advantages of the flanged diffuser at wind velocities lower than the rated velocity while reducing the wind loads acting on the diffuser and blades at higher wind velocities. Numerical analyses of fluid-structure interactions are performed to investigate the flow field around the diffuser with a self-adaptive flange as well as the variation of wind load acting on the diffuser because of the reconfiguration of the self-adaptive flange at various wind velocities.
3.1 Introduction

In addition to the earlier experimental researches on DAWTs by Lilley and Rainbird [1], Oman et al. [2], Igra, Foreman and Gilbert, Phillips et al. [3, 4], and corresponding numerical studies [5-10], Ohya et al. developed a new shrouded wind turbine with a large flange mounted at the exit of the diffuser and a curved inlet section, and they found that the large flange created a low-pressure region owing to the strong vortices formed behind the flange based on the wind tunnel tests and numerical results [11-16]. As a result, much more wind was drawn into the diffuser than in a general DAWT without the flange. Their experimental and numerical results showed that the flange increased the wind velocity in the nozzle of the diffuser by 1.6–2.4 times over the upwind velocity, and by 1.2–1.7 times over the wind velocity in the nozzle of the diffuser obtained by the same diffuser but without the flange, respectively. This acceleration effect of the wind velocity led to an increase in the power extraction by 2–3 times over the bare wind turbine and 1.4–2.1 times over the wind turbine with the same diffuser but without the flange, respectively. However, in addition to the advantages of shrouded wind turbine with a flanged diffuser, one of drawbacks of the wind turbine shrouded by a flanged diffuser is also obvious. An extra and large structure, namely, the flanged diffuser is added to a bare wind turbine, which makes this kind of wind turbine complicated and costly, especially for very large-scale wind turbines. It is easily imagined that a large flanged diffuser brings higher wind loads to the supporting structures of wind turbine than a bare wind turbine, although a large flange can lead to high wind velocity and then high wind power extraction. For this reason, it is difficult to develop very large-scale wind turbines with a flanged diffuser due to the structure and cost issues, although a large wind turbine is much more cost effective in wind
power extraction than a small wind turbine. Thus, it is of interest to develop a smart flanged diffuser which flanged diffuser, which can maintain the acceleration function at low wind velocities and meanwhile can reduce the wind loads at high wind velocities. This is a new challenge for upgrading wind turbines shrouded by flanged diffusers.

In this study, a diffuser with a self-adaptive flange is proposed based on a novel idea of bi-cantilevered plates. This self-adaptive flange can not only maintain the advantage of accelerating the approaching wind velocity at velocities lower than the rated wind velocity but can also gradually reduce the wind loads acting on the wind turbine structure at higher wind velocities. Furthermore, this proposed flange is completely self-adaptive and works without the aid of extra electrical or mechanical devices. That is, no electric energy is consumed for the control of this self-adaptive flange. In this chapter, numerical analyses of fluid-structure interaction are performed to investigate the flow field around the diffuser with the self-adaptive flange as well as the variation in wind load acting on the diffuser because of the reconfiguration of the self-adaptive flange at various wind velocities.

3.2 A novel diffuser with a self-adaptive flange

It is useful to recognize that how the wind load acts on the wind turbine shrouded by a flanged diffuser is affected by the flanged diffuser before considering a self-adaptive flange. According to the previous results [17] of wind tunnel experiments for a 500 W wind turbine with and without a flanged diffuser and CFD analysis using the CFD code of Star-CD, wind loads acting on a wind turbine with a typical compact flanged diffuser are about 3.2 times the wind loads acting on a bare wind turbine with the same rotor diameter, as shown in Fig. 3-1. In the other words, about 69% of the total wind load acting on the
wind turbine with a flanged diffuser is caused by the flanged diffuser. It is noted that in Fig. 3-1 the towers of two kinds of wind turbines with and without flanged diffuser are assumed to have the same dimensions, although the tower of wind turbine with a flanged diffuser generally has larger diameter than that of a bare wind turbine because of higher wind load caused by the flanged diffuser. For this reason, a large-scale wind turbine with flanged diffuser requires a much more complicated and stronger supporting structure, including the tower and the supporting structure of flanged diffuser, than a bare wind turbine to assure the safety of the wind turbine at limited high wind velocity. As a result, a higher cost of a wind turbine with a flanged diffuser is inevitable. Therefore, it is important to reduce the wind load caused by the flanged diffuser for the development of large-scale and practicable wind turbine with a flanged diffuser.

\[\text{Figure 3-1: Wind loads acting on two kinds of wind turbines with and without a flanged diffuser at 60 m/s for various rated powers.}\]

Referring to Fig. 1-12, the flanged diffuser comprises two main parts: cylindrical part and flange. Thus the wind load acting on the flanged diffuser can be divided into two parts:
one partial wind load caused by the cylindrical part and another partial wind load caused by the flange. In order to reduce the above wind loads significantly, challenges to the structures of these two parts are necessary. As a first trial, this study takes on the challenge of developing a smartly adaptive flange. Here a smartly adaptive flange means a flange that satisfies two requirements. The first one is that the flange can maintain the original function of accelerating the approaching wind at a velocity lower than a specified wind velocity and the second one is that the flange can reduce the wind load gradually with the increase of wind velocity at velocities higher than the specified velocity. In the present study, the rated velocity $V_r$ of the wind turbine is considered as the specified wind velocity. Based on today’s updated control technologies, it is not difficult to make a smartly adaptive flange using extra and active electric or mechanical control devices. However, extra and active electric and mechanical control devices not only further increase the manufacturing cost of the wind turbine, but also consume electric energy. Therefore, in present study, a self-adaptive flange which does not use any extra and active electric or mechanical devices, is considered. Learning from natural leaves of plant and trees tells us that a flexible flange structure can gradually reduce the wind load acting on it by its deformation, as illustrated in Fig. 3-2. The flange in the conventional wind turbine with a flanged diffuser is rigidly connected with the cylindrical part (see Fig. 3-2(a)) and has a small deformation even at the limiting high wind velocity, which leads to a high wind load acting on the flange. In the above flexible flange idea (see Fig. 3-2(b)), the flange is like a cantilevered plate with a fixed upper edge and a free lower edge so that it can gradually reduce the wind load acting on it by its deformation. However, this idea only satisfies the second requirement of a smartly adaptive flange and the first one is not satisfied because the flexible flange
also deforms at wind velocities lower than the rated velocity $V_r$, as depicted by the solid curve in Fig. 3-2(c). A smartly adaptive flange requires that the open gap be zero or nearly zero at low wind velocities and gradually increase as wind velocity increases to high velocities. Therefore, improvement of the flexible flange idea is necessary to make a real smartly adaptive flange.

In order to overcome the shortcomings of the above flexible flange, a flexible flange idea of two cantilevered plates is considered, as described in Fig. 3-3. Two cantilevered plates may have different flexural rigidity, which depends on the design of the interval between two plates, the length of the two plates, the contact requirement at rated velocity, and the gap opening at the limited high wind velocity. It is understood that the deformation of the rear plate is zero or nearly zero before the front plate contacts the rear plate because the wind load acting on the rear plate is negligible. Once the front plate contacts the rear plate, the rear plate will deform gradually together with the front plate with the increase of wind velocity due to the contact force. Obviously, this idea of a bi-cantilever flange satisfies the two requirements mentioned above for a smartly adaptive flange and thus this

\begin{figure}
\centering
\includegraphics[width=\textwidth]{fig3-2.png}
\caption{A general flexible flange idea: (a) conventional rigid flanged diffuser; (b) general flexible flange idea; (c) curves of open gap vs. wind velocity.}
\end{figure}
bi-cantilever flange is a completely self-adaptive flange. As the deformation of the bi-cantilever flange is not controlled by any extra and active electric or mechanical devices but rather adapts passively to the wind load, in the study the bi-cantilever flange is named hereinafter a self-adaptive flange.

![Diagram of bi-cantilever flange](image)

**Figure 3-3:** Illustration of the idea of a bi-cantilever flange.

![Diagram of compact-type diffuser with self-adaptive flange](image)

**Figure 3-4:** A compact-type diffuser with a self-adaptive flange.

A diffuser with a self-adaptive flange is proposed based on the above self-adaptive flange concept. A 100 kW wind turbine with a typical compact-type flanged diffuser [17] is considered as a model in the present study. Geometry and dimensions of the compact-type flanged diffuser and the structure of the self-adaptive flange are depicted in Fig. 3-4.
The outer diameter of the flange is 15.438 m, the height of the flange is 0.6475 m, the throat diameter of the cylindrical part is 12.95 m, and the length on the flanged diffuser in the flow direction is 1.777 m. In the case of original 100 kW unit built at the Ito campus of Kyushu University, the cylindrical part and the flange were made of sandwich shell structure consisting of glass fiber reinforced plastic (GFRP) skin and porous plastic core. The thickness of the sandwich shell is about 30 mm, and the ribs were made of 9 mm in thickness steel plate.

To simplify the present numerical simulation, the cylindrical part, ribs and other structural parts of the wind turbine are considered as rigid structures, and only the flange is considered as a flexible structure. The flange is uniformly divided into 40 parts along the circumference and each one of the 40 parts is further equally divided into two subparts along the radial direction based on the original dimensions of the flange and the fabrication view of the subparts, as shown in Fig. 3-4b. Two rings made of rectangular steel pipe are added in the outer and middle diameters of the flange to fix the upper edges of all the subparts of the divided flange. The rings, ribs, and cylindrical part form a relatively rigid structure. Each subpart comprises two cantilevered plates. The details of the models used in the analyses of the structure and CFD are described in next section.

### 3.3 Numerical simulations

To investigate the availability and feasibility of above proposed self-adaptive flange applied to the wind turbine with a flanged diffuser, numerical simulations of the interaction between wind flow and the self-adaptive flange are performed based on the iterative analyses of CFD and structural mechanics. In the iterative calculations, the CFD analysis
gives the pressure acting on the flange. The structural analysis gives the deformation of the flange. The flowchart of the iterative calculations of fluid-structure coupling is described in Fig. 3-5.

**Figure 3-5:** Flowchart of the iterative CFD and structure coupling analyses.

It is noted that, in general, the exact pressure acting on the flexible flange is quite complicated due to the interaction between the fluid and deformation of the flange and fluctuates around its mean value over time [18, 19]. The pressure distribution on the flange near the outer and inner boundary of the flange is not uniform due to the influence of boundary conditions. However, on the other hand, according to the recent wind tunnel experiments [18], the fluctuation of total pressure acting on a flexible plate is about 5% of the mean pressure if the plate has the proper flexure rigidity. The regions with abrupt pressure change near the outer and inner diameters of the flange are about 5.3% and 6.7%, respectively, of the whole area of the plate surface so that the influence of non-uniform
pressure on the deformation of the plate is limited. Therefore, in the present analysis it is assumed that the flexible flange in the uniform and steady wind flow is approximately under a quasi-static equilibrium state and the pressure acting on the flange is uniformly distributed on the front surface of the flange. Here, the quasi-static equilibrium state means that the deformed flange in the flow is like a rigid curved-flange under this state. The iterative analyses of CFD and structural mechanics is completed if the difference between the two averaged pressures, namely, the pressure applied to the flat flange in structural analysis to solve the configuration of deformed curved-flange and that obtained from the CFD analysis of the curved-flange, is less than 0.5%. Details of structure and CFD analyses are described in the following subsections. It is pointed that the more advanced FSI method of coupling simultaneously unsteady CFD with structural dynamics and moving objects using the immersed boundary method [20] is useful in further FSI analysis to obtain more accurate evaluation for the present flexible self-adaptive flange.

3.3.1 Structural mechanics analysis

According to the symmetry of the flange structure with respect to the rotor central axis, one of the 40 parts is used as the flange model in the structural mechanics analysis of the self-adaptive flange and its geometry and dimensions are described in Fig. 3-6.

One subpart comprises two cantilevered plates with fixed upper edges and free lower edges. Plates are sectorial and the widths of upper and lower arcs are about 1.111 m and 1.213 m, respectively. The length of plates is about 0.324 m. The thickness is 1.25 mm for the front plate and 0.5 mm for the rear plate, as given in Table 3-1.
Figure 3-6: Self-adaptive flange model used in the structural analysis: (a) front view and dimensions, and (b) 3-dimensional view and loading and boundary conditions.

Table 3-1: Stacking sequences and thickness of the front and rear plates.

<table>
<thead>
<tr>
<th>Plate</th>
<th>Layers</th>
<th>Thickness (mm)</th>
<th>Laying direction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Front plate</td>
<td>10</td>
<td>1.25</td>
<td>[45/90/−45/0/45]_s</td>
</tr>
<tr>
<td>Rear plate</td>
<td>4</td>
<td>0.5</td>
<td>[75/−75]_s</td>
</tr>
</tbody>
</table>

The pressure obtained from the CFD analysis is uniformly applied on the front plates of the flange model and a frictionless contact condition is defined between the front and rear plates. Carbon fiber reinforced plastic (CFRP) laminates are used for the front and rear plates because the laminate flexure rigidity is easily adjusted by changing the fiber direction of plies [21]. The CFRP laminates consist of multiple unidirectional CFRP laminas. The material properties of the unidirectional CFRP lamina are listed in Table 3-2.

Table 3-2: Material properties of the unidirectional CFRP lamina.

<table>
<thead>
<tr>
<th>Young’s Modulus (GPa)</th>
<th>Poisson’s Ratio</th>
<th>Shear Modulus (GPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_1 = 142$</td>
<td>$v_{12} = 0.32$</td>
<td>$G_{12} = 4.2$</td>
</tr>
<tr>
<td>$E_2 = 8.8$</td>
<td>$v_{23} = 0.27$</td>
<td>$G_{23} = 3.7$</td>
</tr>
<tr>
<td>$E_3 = 8.8$</td>
<td>$v_{31} = 0.02$</td>
<td>$G_{31} = 4.2$</td>
</tr>
</tbody>
</table>
The stacking sequence and thickness of the front and rear plates are determined according to the conditions required by the proposed self-adaptive flange. That is, the front plate must not be in contact with the rear plate at \( V < V_r \), but must be in contact with the rear plate at \( V \geq V_r \), and the open gap (Fig. 3-3) gradually increases with the increase of wind velocity at \( V > V_r \). Based on the numerical calculations of fluid-structure interaction at a given rated wind velocity \( V_r \), the stacking sequence and thickness of the front and rear plates are obtained and listed in Table 1. The nonlinearity of the finite deformation of the flange plates is considered in the structural mechanics analysis by the nonlinear finite element method (FEM) and the numerical analysis is performed using commercially available MSC/Marc2010 codes. The rated velocity \( (V_r) \ 12 \text{ m/s} \) is used in the present simulations hereafter.

3.3.2 CFD Analysis

According to the structural symmetry of the flanged diffuser, an axisymmetric model of flanged diffuser is used in the iterative CFD analyses. Computational domains for the model of a flanged diffusers with a rigid and flat flange is described in Fig. 3-7a. A typical model of a flanged diffuser with a deformed self-adaptive flange is described in Fig. 3-7b. Grids around two flanged diffusers are also depicted in Fig. 3-7, respectively. It is noted that the bended shapes of two front plates of the self-adaptive flange are used as the configuration of the deformed flange in Fig. 3-7b. The diffuser with a rigid and flat flange, shown in Fig. 3-7a, is used in the first CFD analysis to determine the averaged pressure acting on the flange for the successive structural analysis. After the first CFD analysis, applying the averaged pressure obtained from the CFD analysis to the self-adaptive flange
Figure 3-7: Computational domains, boundary conditions and grids: (a-1) diffuser with a rigid and flat flange, (a-2) related grids around the flanged diffuser, (b-1) diffuser with a typical deformed self-adaptive flange, and (b-2) related grids around the flanged diffuser.

before deformation and conducting structural analysis solve a configurations of deformed self-adaptive flange. Consequently, the deformed self-adaptive flange is considered as a rigid curved-flange and is used in the next CFD analysis, as shown in Fig. 3-7b, to solve the averaged pressure acting on the deformed self-adaptive flange for the next structural analysis. Repeat these iterative CFD and the structure analyses until the convergent condition is satisfied, as described in the flowchart in Fig. 3-5. In the CFD analyses, a uniform and steady flow $V$ is set at the inlet boundary, no-slip boundary condition is specified on the surface of the flanged diffuser and the $y^+$ values are in the range of 23 to 381, zero-pressure condition is prescribed at the outlet boundary, slip boundary condition
is applied to the top boundary, and axisymmetric condition is applied to the bottom boundary (the axisymmetric axis). The flow field is expressed by the continuity and incompressible Reynolds-averaged Navier–Stokes equations, the standard $k–\varepsilon$ model is used in the analysis of turbulent flow, and a second-order numerical scheme is used to compute the convection and diffusion terms. Iterative CFD analyses are performed using a CFD code of ANSYS Fluent 13.0, which is a useful CFD code for modeling fluid flow in complex geometry [22].

Figure 3-8: Computational domain used by a previous research [13] for the comparative simulation ($V_0 = 5$ m/s, $D = 0.2$ m).

To investigate the effect of grid number and Reynolds number on the simulation results of velocity and pressure coefficient distributions along the axisymmetric axis, two different grid systems and two different Reynolds numbers were investigated first in the CFD analysis using the flanged diffuser model shown in Fig. 3-7a. In addition, for a further confirmation of the numerical accuracy of present CFD analysis, a comparative simulation is conducted for the model of a rigid flanged diffuser used in a previous research [13], as shown in Fig. 3-8. The boundary conditions, the grid number, and the grid distribution are
as same as those used for the CFD analysis of model shown in Fig. 3-7a. $V_0$ is the approaching wind velocity of a uniform and steady flow at the inlet.

### 3.4 Results and discussion

The effects of grid number and Reynolds number on the simulation results are described in Fig. 3-9. The distribution of streamwise velocity and the pressure coefficient $C_p$ along the axisymmetric axis are depicted in Figs. 3-9a and 3-9b for two grid numbers (38,770 and 66,520) and two Reynolds numbers (10,634,703, and 53,173,516), respectively. Here $C_p$ is defined by the following equation:

$$C_p = (p - p_\infty)/(0.5\rho V_0^2),$$

where $V_0$ is the approaching wind velocity of a uniform and steady flow at the inlet, $p$ is the static pressure along the axisymmetric axis, and $p_\infty$ is the static pressure in a far upstream field. The transverse axis $x/D_b = 0$ denotes the position of the diffuser entrance. As shown in the figure, the difference between the results of two different grid numbers or two different Reynolds numbers is insignificant. In addition, the difference between two drag coefficients for grid numbers 38,770 and 66,520 with $Re = 10,634,703$ is about 1.6%. Therefore, all the following simulations use the grid number of 66,520. Numerical results of the comparative simulation are presented together with those obtained from the previous research [13] in Fig. 3-10. The difference of both results is within about 5%, which confirms the accuracy of the present CFD analysis.
Figure 3-9: Effects of different grid numbers and Reynolds numbers on streamwise velocity and pressure coefficient distributions along the axisymmetric axis in the case of the diffuser with a rigid and flat flange (Fig. 3-7a): (a) streamwise velocity and (b) pressure coefficient.
Figure 3-10: Streamwise velocity and pressure coefficient along the axisymmetric axis in the present comparative simulation are compared with those obtained from the previous research Abe et al. [13] \( (V_0 = 5 \text{ m/s}, D = 0.2 \text{ m}, L = 1.5D, h = 0.5D, \phi = 4^\circ): \) (a) streamwise velocity, and (b) pressure coefficient.

Numerical results of flow field (velocity contour and streamline) around the diffuser with a rigid and flat flange (Fig. 3-7a) at 30 m/s are presented in Fig. 3-11 – Fig. 3-13. In Fig. 3-11, a representative time history of aerodynamic drag coefficient of the flanged diffuser, obtained by the present numerical calculation, is illustrated. The drag coefficient shows an sinusoid-like fluctuation with the period \( T_v \) of 0.54/s which corresponds the period from the generation, release and then regeneration of the vortex behind the flange. Fig. 3-12 shows the velocity contour around the flanged diffuser in the duration of a period. Therefore, it is understood that the periodic fluctuation of the drag coefficient corresponds to the vortex generation and release behind the flange.
Figure 3-11: A representative time history of the drag coefficient for the diffuser with rigid and flat flange at 30 m/s.

Figure 3-12: Velocity contour around the flanged diffuser in one period at 30 m/s.

Fig. 3-13 presents clear images of velocity contour and streamline for the case at 30 m/s. It is seen in Fig. 3-13a that acceleration of the wind velocity occurs in the inner region of the flanged diffuse, especially near the diffuser wall. Because the power output is proportional to the cube of wind velocity as mentioned earlier, a significant augmentation of the power output is expected. In addition, as shown in Fig. 3-13b, two vortices are
generated behind the flange, which cause a low-pressure region behind the flange and increase the pressure difference between the entrance and exit of the flanged diffuser. These results are consistent with previous reports [13, 14]. The streamlines converge near the diffuser entrance, as indicated by the arrow, and more air is drawn into the diffuser.

Figure 3-13: Flow fields around the flanged diffuser with a rigid flange (Fig. 3-7a) at 30 m/s: (a) velocity contour and (b) streamline.

Fig. 3-14 presents the comparison of the drag forces and the time-averaged drag coefficients along the flange, the cylindrical part, and the total diffuser obtained from the CFD analyses for the diffuser with a rigid and flat flange (Fig. 3-7a) at various velocities. The drag coefficient is defined by the following:

\[ C_d = \frac{drag}{0.5 \rho V^2 A}, \quad A = \pi D_b^2 / 4. \]  

3-2

The reference area \( A \) is the circular area of outer diameter of the flange [17]. The relationship between the drag force and the velocity obeys the parabolic law. It is seen that approximately 60% of the total drag acting on the flanged diffuser is deduced from the flange.
Figure 3-14: Comparison of the drag forces (a) and time-averaged drag coefficients (b) among the flange, the cylindrical part, and the total flanged diffuser obtained from the CFD analyses for the diffuser with a rigid and flat flange (Fig. 3-7a) at various velocities.

Numerical results related to the diffuser with a self-adaptive flange are presented in Figs. 3-15 – 3-20. Velocity contour and streamline around the diffuser with the bended self-adaptive flange at 30 m/s are shown in Fig. 3-15. Compared to the rigid flange in Fig. 3-13, the velocity in the velocity field of the bended self-adaptive flange is lower because the deformed self-adaptive flange allows some wind to flow through the open gaps. Consequently, the acceleration effect of the flanged diffuser decreases and the reduction in the wind load acting on the diffuser and wind turbine blades is expected due to the open gaps of the deformed self-adaptive flange. Similar behaviors are observed from the streamline field shown in Fig. 3-15b. The streamlines at the entrance of flanged diffuser in Fig. 3-15b become smoother than those in Fig. 3-13, a lot of streamlines flow through the open gaps, and the separation region behind the flange is much smaller than that in Fig. 3-13. As a result, the pressure difference between the entrance and exit of the diffuser decreases.
as the open gaps gradually become large and the wind drawn into the self-adaptive flanged diffuser is less than that in Fig. 3-13.

![Flow fields around the diffuser with a bended self-adaptive flange at 30 m/s: (a) velocity contour and (b) streamline.](image)

**Figure 3-15**: Flow fields around the diffuser with a bended self-adaptive flange at 30 m/s: (a) velocity contour and (b) streamline.

![Time history of the drag coefficient for the diffuser with the self-adaptive flange at 30 m/s.](image)

**Figure 3-16**: Time history of the drag coefficient for the diffuser with the self-adaptive flange at 30 m/s.

Similarly, the variation of the drag coefficient with time is given in Fig. 3-16 for the diffuser with the self-adaptive flange at 30 m/s, and it is found that the time-averaged drag
coefficient decreased compared to that in Fig. 3-11 due to the bending deformation of the flange. It should be indicated that the vortex period $T_v$ is 0.84s and is larger than 0.54s in Fig. 3-11.

![Diagrams showing streamwise velocity and pressure coefficient distributions along the axisymmetric axis in the flanged diffusers with a rigid and flat and with a bent self-adaptive flange.](image)

**Figure 3-17**: Streamwise velocity and pressure coefficient distributions along the axisymmetric axis in the flanged diffusers with a rigid and flat and with a bent self-adaptive flange: (a) streamwise velocity, and (b) pressure coefficient.

In Fig. 3-17 streamwise velocity and pressure coefficient distributions along the axisymmetric axis in the diffusers with either a rigid and flat flange or a bent self-adaptive flange are depicted together for a comparison. Solid and dotted curves denote the results of the diffuser with a rigid and flat flange and those of the diffuser with a bent self-adaptive flange, respectively. The maximum velocity increment occurs near the rotor plane for the two kinds of flanged diffusers. It is seen that the maximum velocity increment and the absolute value of the pressure coefficient in the diffuser with a bent self-adaptive flange decrease by 27.87% and 28.5%, respectively, due to the deformation of the self-adaptive flange, compared to those in the diffuser with a rigid and flat flange.
Figure 3-18: Wind velocity distributions on the rotor planes, at 60 m/s, of two flanged diffusers with either a rigid and flat flange or a bended self-adaptive flange, respectively.

To demonstrate the distribution of wind velocity along the radial direction of the flanged diffuser, wind velocity distributions on the rotor planes, at 60 m/s, of two flanged diffusers with either a rigid and flat flange or a bended self-adaptive flange are depicted in Fig. 3-18, respectively. It is seen that slow increase in the wind velocities in the rotor planes of two kinds of flanged diffuser is seen in the region of $2r/D_t < 0.6$ and fast increase is seen near the diffuser wall. These results are consistent with those shown in Fig. 3-13a and Fig. 3-15a. Furthermore, it is again seen that the wind velocity in the rotor plane of the diffuser with a self-adaptive flange is lower than that in the rotor plane of the flanged diffuser with a rigid and flat flange, except for a very small region near the diffuser wall. These results again confirm the availability of the self-adaptive flange for the reduction in the wind load acting on the flanged diffuser and wind turbine blades at high wind velocities.
Figure 3-19: Configurations of deformed upper front plate of the self-adaptive flange at various wind velocities.

Fig. 3-19 presents the configurations of the deformed upper front plate of the self-adaptive flange under various velocities. The small draft in the upper-right of the figure shows an image of the self-adaptive flange at 60 m/s. It should be noted that the deformation of the lower front plate of the self-adaptive is similar to that of the upper front plate because of the similar geometry and the same boundary conditions. It can be seen that at the rated velocity of 12 m/s, the displacement of the free end of the plate is 34.6 mm in the X direction, which is close to the gap (35 mm) between the front and rear plates. That is, the front plate almost touches the rear plate at the rated velocity. After that, the gaps in both the X and Y directions increase and the open gap in Y direction reaches 130 mm at 60 m/s. A full image of the diffuser with a self-adaptive flange at 60 m/s is presented in Fig. 3-20. The deformation of the self-adaptive flange creates two ring-like open gaps. As a result, much more wind can flow through the gaps so that the wind loads acting on the flange and the wind turbine blades decrease at high velocities.
Numerical results related to the drag forces acting on the flanged diffuser, the cylindrical part, and the self-adaptive flange are depicted in Fig. 3-21. Here $F_d$ denotes a drag and $F_{d-0}$ denotes the drag acting on the diffuser with a rigid and flat flange. The solid line shows the drag ratio of the total drag acting on the diffuser with a self-adaptive flange over $F_{d-0}$, dotted curve with triangle marks shows the drag ratio for the flange part, and dashed curve with diamond mark gives the drag ratio for the cylindrical part. It is seen that the drag ratio of the cylindrical part decreases slightly with the increase of wind velocity, which is reasonable because the cylindrical part is assumed as rigid part and does not deform. Besides, it is seen that the drag ratios show almost no decrease at wind velocities below 12 m/s. These results further confirm that the self-adaptive flange can maintain the acceleration function at the wind velocities below the rated velocity. On the other hand, the values of drag ratio of the whole flanged diffuser and the flange decrease significantly from 0.996 to 0.65 and 0.618 to 0.341, respectively, with the increase of the wind velocity from 12 to 60 m/s. The wind loads acting on the whole flanged diffuser, on the flange, and on the cylindrical part at 60 m/s are reduced by approximately 34.5%, 45.1%, and 18.4%,
respectively, because of the deformation of the self-adaptive flange. These results further prove that the present proposed diffuser with a self-adaptive flange is available for the reduction of the wind load acting on the flanged diffuser and is applicable for the middle and large-scale wind turbines with a flanged diffuser.

![Figure 3-21: Drag ratio of the diffuser with a self-adaptive flange at various wind velocities.](image)

3.5 Summary

In this study, a self-adaptive flange is proposed for the wind turbine shrouded by a flanged diffuser to reduce the wind loads acting on the flanged diffuser at high wind velocities. The self-adaptive flange can maintain the advantages of the flanged diffuser at wind velocities lower than the rated velocity and reduce the wind load acting on the diffuser and blades at higher wind velocities. Numerical analyses of fluid–structure interaction between the flow and the diffuser with a self-adaptive flange are carried out to
investigate the flow field around the flanged diffuser and the variation of wind load acting on the flanged diffuser due to the reconfiguration of the self-adaptive flange at various wind velocities. Based on the above numerical analyses and discussions the following conclusions are obtained:

1. Approximately 60% of the total wind load acting on the total flanged diffuser of a wind turbine shrouded by a typical compact flanged diffuser is caused by the wind load acting on the flange. Therefore, reduction of the wind load acting on the flange is important in the development of middle and large scale wind turbines shrouded by a flanged diffuser.

2. The presented self-adaptive flange without any extra energy consumption is a viable solution. The structure of the self-adaptive flange is rather simple and practically feasible for the application to the middle and large scale wind turbines shrouded by a flanged diffuser to reduce the wind load acting on the wind turbine structure at high velocities.

3. The presented numerical results demonstrate that the wind load acting on the total flanged diffuser at 60 m/s can be reduced by approximately 34.5% using the proposed self-adaptive flange. Furthermore, the reduction in the wind loads acting on the wind turbine blades is also expected at high wind velocities beyond the rated value because the wind velocity in the rotor plane is reduced due to the deformation of the self-adaptive flange.
Bibliography


CHAPTER 4

Experimental study of a small diffuser with a self-adaptive flange

Based on the diffuser with a self-adaptive flange proposed in Chapter 3, two small flanged diffusers are manufactured, one with a rigid and flat flange and the other one with a self-adaptive flange. The cylindrical part of the flanged diffuser component is fabricated by a CFRP laminate and the flange is made by plastic sheets. Wind tunnel experiments are performed to investigate the flow field around the two diffusers as well as the variation of wind loads acting on the diffuser with a self-adaptive flange due to the reconfiguration of the self-adaptive flange at various wind velocities.
4.1 Introduction

In Chapter 3, a self-adaptive flange [1] is proposed for the wind turbine shrouded by a flanged diffuser to reduce the wind loads acting on the flanged diffuser at high wind velocities. Numerical analyses of fluid-structure interactions are performed to investigate the flow field around the diffuser with a self-adaptive flange.

In this chapter, to further confirm the feasibility of the diffuser with a self-adaptive flange, two small flanged diffusers, one with a rigid and flat flange and the other one with a self-adaptive flange, are manufactured. Wind tunnel experiments are carried out to investigate the effect of the self-adaptive flange on the change of the flow field around the diffuser and the variation of the wind load acting on the diffuser.

4.2 Design of a small diffuser with a self-adaptive flange

Based on the concept of the self-adaptive flange described in the previous chapter, a small diffuser with a self-adaptive flange is fabricated. A 300 W wind turbine with a typical compact-type flanged diffuser [2, 3] is considered in the present study. The geometry and dimensions of the compact-type flanged diffuser, and the structure of the self-adaptive flange are depicted in Fig. 4-1. The outer diameter of the flange is 380 mm, the height of the flange is 30 mm, the throat diameter of the cylindrical part is 300 mm, and the length on the flanged diffuser in the flow direction is 30 mm. In the present experiments, the cylindrical part and the flange are made by CFRP laminate and plastic sheet, respectively. The flange is uniformly divided into 60 parts along the circumference as shown in Fig. 4-1b. A ring made from rectangular plastic fixed rings (PET) plates is added in the outer
diameter of the flange to fix the upper edges of the divided flange. Each subpart comprises two cantilevered plates.

**Figure 4-1**: A small compact-type diffuser with a self-adaptive flange.

### 4.3 Experiments

In this study, diffusers with either a rigid and flat flange or a self-adaptive flange are fabricated and tested. Experiments contain three sections. First, a mold for the formation of diffuser components is designed and manufactured. The flanged diffuser is divided into four identical parts, each comprising a 90° arc. Second, four components of the flanged diffuser are fabricated by CFRP prepregs based on the same mold, and are cured using an autoclave. Thirdly, wind tunnel experiments are performed for the flanged diffusers with either a rigid and flat flange or a self-adaptive flange to investigate the effect of the self-
adaptive flange on the change of flow field around the flanged diffuser and the variation of wind load acting on the flanged diffuser.

4.3.1 Manufacture of the mold

Fig. 4-2 shows the schematic of constructing a mold for the formation of one fourth part of the flanged diffuser. The concept is to construct a three dimensional (3D) mold from multiple two dimensional (2D) quarter circle plates and two side plates. The thickness of

Figure 4-2: Schematic of the mold for one-fourth of the flanged diffuser: (a) sectional division of the flanged diffuser, (b) construction of the mold skeleton, (c) design view of the mold, (d) dimensions for bolt holes and side plate.
each quarter circle plate is 1 mm, and the radius of each plate is calculated based on the geometry of the diffuser. Total 30 quarter circle plates with different radius can form a piecewise-linear curvy surface to simulate the curvy surface of the cylindrical part of the flanged diffuse along the flow direction, as shown in Figs. 4-2b and 4-2c. An additional quarter circle plate with the outer radius of the diffuser is added for the formation of the flange part, which is numbered as 31st in Fig. 4-2a. By stacking the 31 quarter circle plates together constructs the mold for the formation of one fourth part of the diffuser (see Figs. 4-2b and 4-2c). Two side plates, with lengths of the same size as the outer radius of the diffuser and widths of the same size as the total thickness of the 31 quarter circle plates, are attached to the two sides of these circular plates. Fig. 4-2c gives the image skeleton of the finished mold in the design. Four bolts are used to fix the multiple quarter circle plates. Dimensions of side plates and the locations of bolt holes are described in Fig. 4-2d. Finally, the piecewise-linear curve is modified into a smooth curve using plaster. The details of making the mold is described as follows.

The procedures for the manufacturing the quarter circle plates is shown in Fig. 4-3. A plastic sheet of UNILATE (1000 × 1000 × 1 mm) with a density of $1.63 \times 10^3$ kg/m$^3$ and a 1 mm thick aluminum plate are selected as the materials for the mold. Good machining properties and high temperature (230°C) resistance of the UNILATE sheet meet the experimental requirements, because the UNILATE sheet needs to be cut into many quarter circle plates and the stacked CFRP prepregs based on the mold are cured at a high temperature (135°C) in an autoclave. The as-received UNILATE plastic sheet is cut into multiple rectangular plates (Fig. 4-3 Step-1). Then, the quarter circle plates are cut along
the arc track marked in the paper, which is attached to the rectangular plate using glue (Fig. 4-3 Step-2). Fig. 4-3 Step-3 shows part of the obtained quarter circle plates.

Figure 4-3: Construction of the mold skeleton for one-fourth of the diffuser.
All the quarter circle plates are stacked long the right-angle sides and bundled using adhesive tape in order to ensure that there is no relative movement between the adjacent plates (Fig. 4-3 Step-4 and Step-5). Subsequently, four holes are drilled for bolts using a drill according to the dimensions in Fig. 4-2d. The assembled mold is shown in Fig. 4-3 Step-6. Note that the quarter circle plates from 1 to 30 were made from unilate plastic sheet, but the 31st plate and two side plates were made from the aluminum plate.

4.3.2 Fabrication of diffusers with either a flat flange or a self-adaptive flange

From Fig. 4-2c and Fig. 4-3 Step-6, it can be observed that there are two problems with the assembled mold. The first one is that the curvy surface constructed from the plates is ladder-like surface. A smooth curvy surface is necessary for the formation of the flanged diffuser with smooth surface. Therefore, the ladder-like surface of the assembled mold must be modified. The second one is the poor heat conductivity of the plastic plate, which will influence the surface quality of the final CFRP structure. Good heat conductivity of the mold ensures good flowability of the resin during the curing process, which is necessary to form a smooth surface and homogeneous structure for the CFRP component.

Plaster is used for the modification of the ladder-like surface. Wet plaster is pressed on the ladder-like surface of the mold and an approximately uniform plaster layer is formed on the surface of the mold. Subsequently, the plaster covered mold is heated in a vacuum drying oven at 135°C for 8 h to remove water from the plaster. A #1500 sandpaper is used to polish the dry plaster layer further until a uniformly smooth surface is achieved. To improve the heat conductivity of the surface, a commercial aluminum film with thickness
of 17 μm is laid over the surface of the plaster layer. Fig. 4-4 shows the schematic of the layout of the cross-section of a vacuum bag mold before cure and the details of the layer construction can be observed from the enlarged view (Fig. 4-4b). It is difficult to attach the aluminum film to the surface of the dry plaster layer. It is also difficult to fix the releasing film which is laid between the mold surface and the CFRP prepregs to separate the cured CFRP structure from the mold. To solve these problems, two types of hand-made combined layers are proposed as presented in Fig. 4-4c and 4-4d. Combined layer 1 is constructed by pasting four layers to the curved surface of the dry plaster layer, in the sequence of double-sided tape, aluminum film, double-sided tape, and releasing film. Similarly, combined layer 2 is constructed by releasing film, double-sided tape, and aluminum film. With the aid of the two combined layers, the aluminum and releasing films can be laid smoothly.
along the curved surface. CFRP prepregs with a certain stacked sequence are laid between the two combined layers. Commercially available unidirectional CFRP prepreg of P3252S-12 (T700SC/#2592) (Toray) is used in this study. The longitudinal modulus is 230.3 GPa and the longitudinal strength is 4900 MPa. The fiber volume fraction ($V_f$) and thickness of the prepreg are $V_f = 67\%$ and 0.125 mm, respectively. Then, the molds with the combined layers and CFRP prepregs are placed on a steel plate and covered with breather layers. Finally, they are enclosed in a vacuum bag and sealed with sealant tape.

Typical images of fabrication procedures as described above are shown in Fig. 4-5. A uniformly smooth curved surface of dry plaster is presented in Fig. 4-5 (Step-2). Fig. 4-5 Step-3 shows the assembled mold after attaching combined layer 1 to the dry plaster layer. Subsequently, the aluminum side plates, bottom plate, and top plate are attached to the mold using bolts and adhesive tape (Step-4). Because the surface of the mold is covered by an aluminum plate and film, good heat conductivity can be ensured during curing process. The stacked CFRP prepreg layers, as given in Table 4-1, are placed on the curved surface, as presented in Step-5. Two kinds of bended edges are applied to the CFRP structure at the ends and the center, as shown in Fig. 4-5 Step-5, these edges are used to connect the other quarter parts of the flanged diffuser. After attaching combined layer 2 onto the upper surface of the CFPR prepregs, the entire mold is placed on a steel plate and covered with a

<table>
<thead>
<tr>
<th>Structure</th>
<th>Layers of laminate</th>
<th>Thickness mm</th>
<th>Laying direction</th>
</tr>
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<tr>
<td>Diffuser</td>
<td>8</td>
<td>1</td>
<td>$[0/90]_{2s}$</td>
</tr>
</tbody>
</table>
Figure 4-5: Surface modification of the mold and fabrication preparation for one-fourth of the diffuser.
breather layer. Then, the assembled mold with CFRP layer is enclosed in a vacuum bag equipped with a vacuum mouth and sealed with sealant tape. After drawing vacuum using a pump, the entire mold is covered closely with a breather layer and vacuum bag (Step-6).

Figure 4-6: Curing one-fourth of the CFRP diffuser, (a) autoclave used for curing one-fourth of the diffuser, and (b) cure cycle used in fabrication.
Figure 4-7: Fabricated CFRP diffuser, (a) one-fourth of the diffuser, and (b) semi-finished total diffuser.
The CFRP part of the one fourth of the flanged diffuser are cured using an autoclave as shown in Fig. 4-6a, and the cure cycle recommended by manufacturing of CFRP prepregs is presented in Fig. 4-6b. The pressure inside the autoclave is increased to 0.3 MPa at first, then the temperature is raised from room temperature to isothermal dwell at 135°C at a heating rate of 2.25°C min, successively, the temperature stays at 135°C for 120 min. At the end of the curing dwell, the pressure inside the autoclave is released and the autoclave is cooled to 60°C in 30 min. The cured CFRP structure then is removed from the autoclave and cooled under ambient conditions.

Fig. 4-7 shows the photographs of CFRP parts and assembled CFRP structure of the flanged diffuser. Two CFRP parts of one-fourth of the diffuser with three bolt holes after trimming the edges and drilling the holes are presented in Fig. 4-7a. Four of these CFRP structures are manufactured by repeating the fabrication procedures described in Figs. 4-5 and 4-6 four times. Then, connecting these four CFRP parts together constructs a full CFRP structure of the flanged diffuser, as shown in Fig. 4-7b.

A support system for the diffuser is constructed from the base, pole, fixture nut and trusses as depicted in Fig. 4-8. For a simplification, a fixture nut with four holes, as shown in Fig. 4-8b, is used as a center hub to fix the trusses. The CFRP structure of the diffuser is installed onto the trusses using eight small nuts.
Figure 4-8: Construction of support system for the diffuser, (a) total structure, (b) fixture nut, (c) pole and base, and (d) truss.
Therefore, the remaining work is to make the flange parts. In the rigid and flat flange case, the flange parts are made by plastic sheet of PET of 2mm in thickness. Four quarters of the flat flange are cut from the PET sheet following the shape marked by the pre-drawing line, as shown in Fig. 4-9a. Then, the four plastic quarters of the flange are attached to the CFRP structure of the diffuser using bolts. A full image of the flanged diffuser with rigid and flat flange is given in Fig. 4-9c.

**Figure 4-9:** Construction of the diffuser with rigid flat flange, (a) semi-installed flanged diffuser, (b) plastic flat flange, and (c) finished diffuser with rigid flat flange.
As for the diffuser with a self-adaptive flange, Fig. 4-10 shows the schematic for the fabrication of the self-adaptive flange. Firstly, four parts of one fourth of the ring, as shown in Fig. 4-10a and 4-10b, are made by 5-mm-thick PET plate in order to fix the upper end of the self-adaptive flange. A quarter ring with multiple small holes of 1 mm in diameter (Fig. 4-10a and 4-10b) is attached to the CFRP structure using two bolts at both ends of the ring as illustrated in Fig. 4-10c. The self-adaptive flange comprises multiple bi-cantilevered plates, as described before. The front and rear plates are cut from the plastic sheets of PVC and PP (0.2 mm in thickness), respectively. The material properties of PVC and PP are listed in Table 4-2. Fig. 4-11 shows the details of making the self-adaptive flange. The upper ends of the front and rear plates of the self-adaptive flange are attached to the front and back surfaces of the fixed ring, as shown in the enlarged view of Fig. 4-10c. The front and rear plates are fixed onto the ring by threading a wire through the holes. A full image of the diffuser with a self-adaptive flange is shown in Fig. 4-11d.

<table>
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<tr>
<th>Material</th>
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<th>Poisson’s ratio</th>
<th>Thickness (mm)</th>
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<tr>
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Figure 4-10: (a) Model of one-fourth of the semi-finished diffuser with self-adaptive flange, (b) shape of one-fourth of the ring with multiple holes, and (c) construction of the diffuser with self-adaptive flange.
4.3.3 Wind tunnel experiments

The large wind tunnel of the Research Institute for Applied Mechanics, Kyushu University, was used. The tunnel has a measurement section of 3600 (width) × 2000 (height) × 1500 (length) mm with a maximum wind velocity of 30 m/s. The flanged diffuser with the support system is placed on a platform at the center of the wind tunnel section. A three component force detector (Nissho Electric Works Co., Ltd., LMC-3501), which can measure the horizontal components of the forces and moment in the vertical direction, is employed to measure the wind load acting on the flanged diffuser in flow direction. The
photographs presented in Fig. 4-12 show the experimental set-up for measuring the wind load. The force detector is set under the platform. A metal fixture is tightly fixed on the detector and is connected to the base of the flanged diffuser by four bolts. As shown in Fig. 4-12 b and d, the flanged diffuser is set on a platform of 1800 (width) × 500 (height) × 2400 (length) mm. To avoid the direct action of the wind load on the detector and fixture, a specific cover is used to surround the force detector, as shown in Fig. 4-12c. The wind loads at the wind velocities from 2 to 25 m/s were measured. The data sampling frequency

![Figure 4-12](image)

**Figure 4-12:** Experimental set-up of force detector and flanged diffuser, (a) three-component force detector connected with fixture, (b) semi-finished set-up, (c) specific cover for force detector, and (d) finished set-up for measuring.
Figure 4-13: Experimental arrangement for measuring wind velocity and pressure coefficient, (a) traverse system equipped with static pressure tube and hot-wire probe, and (b) schematic images of measurement positions.

is 100 Hz and the wind load in a duration of 30 seconds are recorded for each testing case. Furthermore, distributions of wind velocity and static pressure along the horizontal and
radial direction on two vertical planes, namely, the rotor plane and a parallel plane 40 mm behind the rotor plane, are measured using an I-type hot-wire and a static-pressure tube with a diameter of 3 mm, respectively, with the aid of a transversely moving system. Fig. 4-13 gives the experimental arrangement for measuring the wind velocity and static pressure. As shown in Figs. 4-1a and 4-13b, the diameters of the rotor plane and the fixture nut for the diffuser are 300 mm and 30 mm, respectively. The hot-wire probe and static pressure tube are mounted on the transversely moving system to measure the wind velocity and static pressure along radial direction on the two vertical planes. The static-pressure coefficient \( C_p \) is defined as \( C_p = \frac{(p - p_\infty)}{(0.5 \rho V_0^2)} \), where \( p \) is the static pressure measured along the red arrows, and \( p_\infty \) is the static pressure in a far upstream field. Two wind velocities of 10 and 20 m/s are used for the measurement of the wind velocity and pressure coefficient. The origin of the coordinate system \( x-r \) is located at the entrance plane of the diffuser and with a distance of 5 mm from the fixture nut, as shown in Fig. 4-13b. The measurement ranges are marked by the red arrows. The first one is a linear distance of 110 mm with the starting position of \((x = 10 \text{ mm}, r = 0)\) in the coordinate, and the other one is a linear distance of 170 mm with the starting position of \((x = 50 \text{ mm}, r = 0)\). All the velocity and static pressure data were measured from the starting position to the end position with an interval of 10 mm. In order to reduce the blockage effect in the wind tunnel testing, the side walls of the wind tunnel in the central testing section were opened.
4.4 Numerical investigations

Numerical investigations are performed for the small flanged diffusers, with either a flat and rigid flange or a self-adaptive flange, used in the wind tunnel experiments based on the approximate method mentioned in Chapter 2 for a comparison. The procedures for the calculations based on the approximate method are described in section 2.3.

![Diagram of a self-adaptive flange model](image)

**Figure 4-14:** Self-adaptive flange model used in the structural analysis.

According to the symmetry of the flange structure described in Fig. 4-1, one 60th of the flanged diffuser is used as the flange model of the self-adaptive flange in the structural mechanics analysis, as shown in Fig. 4-14. Each subpart comprises two cantilevered plates with fixed upper edge and free lower edge. The pressure obtained from the CFD analysis is uniformly applied on the front plate of the flange model and a frictionless contact condition is defined between the front and rear plates. Plastic sheets used for the front and rear plates are the same as those used in the experiments. Similarly, the front plate must not be in contact with the rear plate at \( V < V_r \), but must be in contact with the rear plate at \( V \geq V_r \). Moreover, the open gap gradually increases with the increase of wind velocity at
\( V > V_r \). The nonlinearity of the finite deformation of the flange plates is considered in the structural mechanics analysis by the nonlinear finite element method, and the numerical analysis is performed using commercially available MSC Marc2010 codes. The rated velocity \( (V_r) \) is 12 m/s in the present simulations.

Computational domains of CFD analysis for the models of the diffusers with either a rigid and flat flange or a deformable self-adaptive flange are described in Figs. 4-15a and 4-15b, respectively, together with the related grids around each flanged diffuser. The boundary conditions of the domains are the same as shown in Fig. 3-7. The diffuser with a rigid and flat flange shown in Fig. 4-15a is used in the first CFD analysis to determine the averaged pressure acting on the flange for the successive bending calculation of the flexible plate. The bended shapes of the self-adaptive flange obtained from the structural analysis are used as the configuration of the deformed flange in Fig. 4-15b, to determine the corresponding normal drag coefficients of these bended flanges. Similarly, by plotting the two curves of \( C_{N-CFD}(\tilde{\theta}) \) and \( C_{N-theory}(\tilde{\theta}) \big|_{\beta=0} \), and comparing them with each other, the correction function \( \tilde{f(\tilde{\theta})} \) can be determined according to Eq. 2-45. Subsequently, the normal forces acting on the self-adaptive flange and the deformations of the flange at various wind velocities are evaluated using the approximated method described in subsection 2.3.3. Finally, CFD analyses are performed to simulate the flow field around the diffuser with a deformed flange whose configuration is obtained from the final iteration calculation of the structure analyses for the flange at the specific wind velocity and the wind load acting on the diffuser and flange.
4.5 Results and discussion

In order to confirm the accuracy of numerical analysis, the effects of the grids number and Reynolds number on the calculation results of the drag coefficients of the diffuser with a rigid and flat flange are presented in Table 4-3 for three grid numbers (25,150, 48,565 and 72,900) and three Reynolds numbers (41,075, 246,451 and 410,752), respectively. The drag coefficient is defined as follows:
\[ C_d = drag \cdot (0.5 \rho V^2 A) \quad A = \pi D_b^2 / 4. \]

The difference between the results of different grid numbers or Reynolds numbers is insignificant. Therefore, all the following simulations use the grids number of 72,900.

**Table 4-3:** Effects of different grid numbers and Reynolds numbers on the calculation results.

<table>
<thead>
<tr>
<th>Case</th>
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<th>( C_d )-flange</th>
<th>( C_d )-cylinder</th>
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<td>0.56</td>
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<tr>
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<td>410,752</td>
<td>0.799</td>
<td>0.559</td>
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</tr>
</tbody>
</table>

**Figure 4-16:** Comparison of the drag forces of the flange, the cylindrical part, and the total flanged diffuser obtained from the CFD analyses and wind tunnel experiment for the diffuser with a rigid and flat flange (Fig. 4-15a) at various velocities.

Fig. 4-16 shows the comparison of the drag forces acting on the flanged diffuser, on the flange part, and the cylindrical part, in the case of a rigid and flat flange, obtained from
wind tunnel experiments and CFD analyses. The relationship between the drag force and the velocity obeys the parabolic law. The drag force acting on the flange obtained from the CFD analysis is approximately 70% of the total drag force. Because the support system for the diffuser is not considered in the CFD simulation as shown in Fig. 4-15, it is reasonable that the total drag force obtained by CFD analysis is relatively smaller than that obtained from the experiments.

**Figure 4-17:** Flow fields around the diffuser with a rigid flange (Fig. 4-15a) at 20 m/s: (a) velocity contour and (b) streamline.

Numerical results of the flow field (velocity contour and streamline) around the diffuser with a rigid and flat flange at 20 m/s are shown in Fig. 4-17. Fig. 4-17a shows that the acceleration of the wind velocity occurs in the inner region of the flanged diffuser, especially, near the diffuser wall, although the velocity does not change so much in the radial direction. Because the power output is proportional to the cube of the wind velocity as mentioned earlier, a significant augmentation of the power output is expected. Similarly, as shown in Fig. 4-17b, two vortices are generated behind the flange, which cause a low-pressure region behind the flange and increase the pressure difference between the entrance and exit of the flanged diffuser. These results are in accordance with the previous reports.
The streamlines converge as they approach the diffuser entrance, which indicates that more air is drawn into the diffuser. The numerical and experimental results related to the diffusers with either a rigid and flat flange or a self-adaptive flange are presented in Figs. 4-18 to 4-24. The velocity contour and streamline around the diffuser with the bended self-adaptive flange at 20 m/s are shown in Fig. 4-18. Compared to the rigid flange, the values of velocity contour are lower than those shown in Fig. 4-17 because the deformed self-adaptive flange allows some wind to flow through the open gaps. Therefore, the acceleration effect of the flanged diffuser decreases, and a reduction in the wind loads acting on the diffuser and wind turbine blades is expected due to the open gaps of the bended self-adaptive flange. Similar behaviors are observed in the streamline field shown in Fig. 4-18b. The streamlines at the entrance of the self-adaptive flanged diffuser become smoother than those of the diffuser with a rigid flange (Fig. 4-17). In addition, many streamlines flow through the open gaps of the self-adaptive flanged diffuser and the separation region behind the flange is much smaller than that of the diffuser with a rigid

**Figure 4-18:** Flow fields around the diffuser with a bended self-adaptive flange at 20 m/s: (a) velocity contour and (b) streamline.
flange. As a result, the pressure difference between the entrance and exit of the self-adaptive flanged diffuser decreases as the open gaps gradually increase, and the wind drawn into the self-adaptive flanged diffuser is lesser than that of the diffuser with a rigid flange.

**Figure 4-19:** Configurations of deformed upper front plate of the self-adaptive flange at various wind velocities.

Fig. 4-19 shows the configurations of deformed front plate of the self-adaptive flange at various velocities, obtained from numerical analysis. The upper-right graph of the figure shows an image of the self-adaptive flange at the rated velocity 12 m/s. At 12m/s, the displacement of the free end of the plate is 4 mm in the $x_2$ direction, which is close to the interval between the front and rear plates, *i.e.*, the front plate almost touches the rear plate at the rated velocity. Subsequently, the gaps in both $x_2$ and $y_2$ directions increase and the open gap in $y_2$ direction reaches 2.58 mm at 25 m/s. Full images of the diffuser with a self-
adaptive flange at various velocities are presented in Fig. 4-20, images obtained from wind tunnel experiment are compared with those obtained from CFD simulation. In Fig. 4-20a and 4-20d, no deformation is observed at a velocity of 2 m/s, because at low velocities only the front plate of the self-adaptive flange bends slightly and does not contact the rear plate.

**Figure 4-20:** Full image of the diffuser with a self-adaptive flange at various velocities obtained from experiments and simulations: (a) experiment: 2 m/s, (b) experiment: 10 m/s, (c) experiment: 24 m/s, (d) simulation: 2 m/s, (e) simulation: 10 m/s, (f) simulation: 24 m/s, and (g) schematic of the diffuser with bended flange.
When the velocity increases to 10 m/s, the front plate tends to contact the rear plate. Therefore, a slight deformation of the flexible flange can be observed in Fig. 4-20b and 4-20e. When the velocity increase to 24 m/s, the deformation of the self-adaptive flange creates a ring-like open gap, as shown in Figs. 4-20c and 4-20f. A schematic of the diffuser with bended flange is given in Fig. 4-20g. As a result, much more wind can flow through the ring-like open gap and the wind load acting on the flange and the wind turbine blades decreases as the open gap increases at high velocities.

The drag forces acting on the total flanged diffuser, the cylindrical part, and the self-adaptive flange, obtained from experiments and simulations are depicted in Fig. 4-21. The solid lines in Fig. 4-21a show the comparison of drag forces acting on the diffusers with either a rigid and flat flange or self-adaptive flange at various velocities obtained from wind tunnel experiments. All drag forces increase with the increase in wind velocity and
Figure 4-21: The variation of drag force acting on the diffuser with self-adaptive flange versus to different velocities: (a) experiment, (b) CFD simulation.

are almost identical at wind velocities below 12 m/s. These results confirm that the self-adaptive flange can maintain the acceleration function at wind velocities below the rated velocity. Then, the difference of the results between the rigid flange and self-adaptive flange is observed because of the reconfiguration of the flange, and the difference increases with the increase of wind velocity. Similar results are produced by the CFD analysis, as shown in Fig. 4-21b. Dashed lines with different marks indicate the numerical results of the drag forces acting on the rigid and flat flanged diffuser and the self-adaptive flanged diffuser as well as the drag forces acting on the cylindrical part and the self-adaptive flange (for the self-adaptive flanged diffuser). Difference is also observed between the drag forces of the two diffusers at wind velocities >12 m/s, and the difference increases with the increase of wind velocity. These results confirm that the self-adaptive flange can maintain the acceleration function at wind velocities below the rated velocity and can reduce the wind load at high wind velocities. Similarly, the drag forces acting on the cylindrical part
and the self-adaptive flange decrease compared to those of the diffuser with a rigid and flat flange (Fig. 4-16). The reduction in the wind loads acting on the entire flanged diffuser is approximately 17.9% and 12.7% at 25 m/s, respectively, according to the estimations of wind tunnel experiment and CFD simulation. The open gap generated by the deformation of the self-adaptive flange at 25 m/s is not so large because of the relatively short length and large modulus of the flange, as shown in Fig. 4-20. Based on these results and if proper material and flange geometry are selected, it is reasonable to expect that the present self-adaptive flange can reduce the wind load acting on flanged diffusers and is applicable for mid- and large-scale wind turbines with flanged diffusers.
Figure 4-22: Experimental comparison for velocity and pressure coefficient distributions along the radial direction for diffusers with either a rigid and flat flange or self-adaptive flange at $V_0 = 10 \text{ m/s}$: (a) velocity, (b) pressure coefficient.

The experimental results for the wind velocity and pressure coefficient distributions along the radial direction of the flanged diffuser are given in Figs. 4-22 to 4-24. Fig. 4-22 shows the distributions of velocity and pressure coefficient distributions in the radial direction for diffusers with either a rigid and flat flange or a self-adaptive flange at 10 m/s. A slow increase in the wind velocities on the two measurement planes ($x = 0.01 \text{ m}$ and $x = 0.05 \text{ m}$) of the two types of flanged diffusers is observed in the region of $r/r_0 < 0.6$, whereas a fast increase is observed near the diffuser wall. Afterward, the wind velocity measured on the plane $x = 0.05 \text{ m}$ sharply decreases in the region of $0.9 < r/r_0 < 1.3$ due to the effect of the flange. These results are consistent with the velocity contour shown in Fig. 4-17. Fig. 4-22b indicates a large separation between the pressure coefficient distributions measured on the two measurement planes ($x = 0.01 \text{ m}$ and $x = 0.05 \text{ m}$) in the range of very small $r/r_0$. 

129
because of the fixture nut which changes the flow field significantly near the fixture nut. The pressure coefficients at the measurement plane ($x = 0.05$ m) for both types of flanged diffusers decrease with the increase of the radial dimension $r$ and show a low-pressure region when $0.9 < r/r_0 < 1.3$, namely, behind the flange. Overall, the experimental results of both flanged diffusers at 10 m/s show good agreement. It is again confirmed that the diffuser with the self-adaptive flange possesses the same acceleration advantage as the rigid flange at wind velocities below the rated velocity.

Fig. 4-23 shows the wind velocities and pressure coefficient distributions at the wind velocity of 20 m/s and shows similar distributions as Fig. 4-22 in the region of $r/r_0 < 0.9$. However, a significant difference is observed in $0.9 < r/r_0 < 1.3$. Fig. 4-23a shows that at the measurement plane $x = 0.05$ m, the minimum velocity of the diffuser with a rigid and flat flange is obtained at approximately $r/r_0 = 1$, whereas it is observed at $r/r_0 = 1.1$ for the diffuser with a self-adaptive flange. Moreover, the minimum velocity behind the rigid and flat flanged diffuser is higher than that behind the diffuser with a self-adaptive flange. It is believed that this is caused by the deformation of the flange, this deformation changes the flow field at the exit of the flanged diffuser such as the position of the vortex. This feature was confirmed by the CFD analysis, as shown in Figs. 4-17 and 4-18. The absolute values of the pressure coefficients for the diffuser with a bended flange decrease compared to those of the diffuser with a rigid and flat flange at almost all the measured points, especially, in the region of $0.8 < r/r_0 < 1.3$. Therefore, the pressure difference between the entrance and exit of the flanged diffuser decreases so that less air is drawn into the diffuser, and the reduction in the wind load acting on the flanged diffuser is achieved.
Figure 4-23: Experimental comparison for velocity and pressure coefficient distributions in the radial direction for the diffusers with either a rigid and flat flange or self-adaptive flange at $V_0 = 20$ m/s: (a) velocity, (b) pressure coefficient.
For the comparison with the above experimental results at 20 m/s, CFD simulation results are presented in Fig. 4-24. These CFD results are roughly consistent with the experimental results shown in Fig. 4-23, although some differences between the velocities and pressure coefficients obtained from simulations and experiments are observed. For instance, there is no difference between the results on the two planes of \(x = 0.01\) m and \(x = 0.05\) m near \(r = 0\) (the fixture nut), which is different from Fig. 4-23, because the support system is not considered in the CFD simulation. In addition, the CFD analysis seems to underestimate the values at almost all the measured positions compared to the experimental results, especially, \(0.8 < \frac{r}{r_0} < 1.3\).
4.6 Summary

In this chapter, small diffusers with either a rigid and flat flange or self-adaptive flange are manufactured. The flow field around the diffusers and the variation of wind load acting on the diffusers are investigated by wind tunnel experiments. Numerical simulations are performed for a comparison with the experimental results. The main conclusions obtained from the research of this chapter are listed as follows.

1. Wind tunnel experimental results indicate that the wind load acting on the diffuser with a self-adaptive flange shows no significant variation compared to the diffuser with a rigid and flat flange at wind velocities below the rated velocity, because there is no open gap. When the wind velocity exceeds the rated velocity, drag forces acting on the flange,
the cylindrical part, and the total diffuser gradually decrease due to the deformation of the flange. At 20 m/s, the drag reductions for the diffuser with a self-adaptive flange are found to be 17.9% and 12.7% at 25 m/s in the wind tunnel experiment and simulation, respectively.

2. Wind velocity and pressure coefficient distributions in the radial direction of the diffusers with either a rigid and flat flange or self-adaptive flange obtained from experiment at 10 m/s show good agreement because no significant reconfiguration of the self-adaptive flange occurs. However, at 20 m/s, differences between the wind velocity and pressure coefficient distributions of the two flanged diffusers are observed, especially, in the region of $0.8 < r/r_0 < 1.3$. The absolute values of the pressure coefficients behind the flange of the diffuser with a bended self-adaptive flange decrease compared to those behind the diffuser with a rigid and flat flange.

3. The present numerical simulation results show good consistency with the experimental results of the velocity and pressure coefficient distributions, although the CFD analysis underestimates the values compared to the experimental results.
Bibliography


CHAPTER 5
Conclusions and future works

This chapter summarizes the main results of this dissertation and predicts the main research topics that may develop from this study in the future.

5.1 Conclusions

In this dissertation, a self-adaptive flange is proposed for the wind turbine shrouded by a flanged diffuser to reduce wind loads acting on the diffuser at high wind velocities. An approximate method is developed to approximately evaluate the interaction of wind flow and self-adaptive flange including the wind loads acting on the flanged diffuser with the self-adaptive flange and the deformation of the self-adaptive flange. Numerical analyses, fabrication of two small flanged diffusers with a self-adaptive flange or a rigid and flat flange, and wind tunnel experiments are performed to investigate the feasibility and validity of the proposed flanged diffusers with a self-adaptive flange. Based on above research, the following conclusions are obtained:

Based on this work, the following conclusions are obtained:

1. The present approximate method is relatively simple compared to conventional complex fluid-structure coupling analysis and is useful for the approximate evaluation of the normal force acting on a flexible plate normal to the wind flow and the deformation of
the plate in practical applications. The numerical results and the comparison with the previous experimental results demonstrate the validity of the proposed approximate method for the evaluation of the approximate deformation of a flexible plate and the averaged wind load on the plate.

2. A self-adaptive flange without using any extra active mechanical or electric devices is proposed. Numerical analysis is carried out to investigate the validity of the proposed self-adaptive flange. Numerical results demonstrate that the proposed self-adaptive flange is available for mid- and large-scale wind turbines shrouded by a flanged diffuser and that the total wind load can be reduced by approximately 34.5% using the proposed self-adaptive flange.

3. The manufacturing test of two small flanged diffusers, namely, one with a self-adaptive flange and the other with a flat and rigid flange, demonstrates the feasibility of the self-adaptive flange. Wind tunnel experiments and numerical simulations are performed to investigate the flow field around the two types of flanged diffusers and the variation of wind loads acting on the diffusers at various wind velocities. Experimental results indicate that the drag reduction for the diffuser with a self-adaptive flange is 17.9% at 25 m/s. Numerical simulation results show good consistency with the experimental results of the velocity and pressure coefficient distributions.

5.2 Future work

Further simulations should be considered for the presence of the rotor in future research, such as assuming a porous actuator disk in the CFD analysis.
In addition, experimental research is necessary to investigate the effect of the self-adaptive flange on the power generation performance of wind turbines with flanged diffusers.
LIST OF FIGURES

Figure 1-1: Global annual installed wind capacity 1977-2014 [1]. .............................................. 2

Figure 1-2: Global cumulative installed wind capacity 1977-2014 [1]. ........................................ 3

Figure 1-3: Top 10 new installed wind capacity Jan-Dec 2014 [1]. .............................................. 3

Figure 1-4: Top 10 cumulative wind capacity Jan-Dec 2014 [1]. .............................................. 4

Figure 1-5: Annual installed capacity by region 2006-2014 [1]. .............................................. 4

Figure 1-6: Installed wind capacity in Japan 1995-2012 [2]. ..................................................... 6

Figure 1-7: Transition of power generation capacity in Japan 1952-2012 [3]. ......................... 6

Figure 1-8: Increase in rated power and size of wind turbines [4]. ........................................... 7

Figure 1-9: (a) Onshore wind potential map and (b) offshore wind potential map in Japan [2]. ................................................................................................................................. 7

Figure 1-10: View of the shrouded wind turbines, (a) Ben Gurion University, Israel [8], (b) Vortec 7 prototype DAWT, New Zealand [10]. ......................................................... 9

Figure 1-11: Wind turbine shrouded with cylindrical wing profiled structure, (a) Hasen [11], (b) F. Bet and H. Grassmann [12]. ............................................................................. 10

Figure 1-12: (a) A 3 kW wind turbine shrouded by a flanged diffuser and (b) the illustration of wind flow around the flanged diffuser [19]. ................................................. 12

Figure 1-13: Two 100 kW wind turbines shrouded by a flanged diffuser at the Ito campus of Kyushu University ...................................................................................... 13

Figure 2-1: Elemental Cartesian fixed control volume showing the surface forces in the x direction only ........................................................................................................... 23
Figure 2-2: Corresponding stresses in the initial and deformed configuration of a plate. 31

Figure 2-3: A thin plate and the coordinate system. ............................................................... 31

Figure 2-4: The schematic of a flexible plate normal to wind flow. ................................. 40

Figure 2-5: Flowchart of the theoretical modeling. ............................................................ 45

Figure 2-6: Flowchart of the calculation sequence of the present approximate method.
............................................................................................................................................... 48

Figure 2-7: A 3D simulation model. ...................................................................................... 50

Figure 2-8: (a) CFD computational domain and (b) grid around the plate. ..................... 52

Figure 2-9: Simulation model for the comparative test. ...................................................... 53

Figure 2-10: Effect of the number of grids on the calculation results. ............................. 55

Figure 2-11: Velocity contour and velocity vector around the vertical rigid plate with
length of 500mm at 10 m/s: (a) velocity contour and (b) velocity vector. ........................ 55

Figure 2-12: Velocity contour and velocity vector around a rigid curved-plate with
length of 500mm at 10 m/s: (a) velocity contour and (b) velocity vector. ......................... 56

Figure 2-13: Theoretical curve of normal force coefficient vs. chord angle of rigid
curved-plates .......................................................................................................................... 57

Figure 2-14: Effect of the plate length on the normal force acting on the flexible plates
at different wind velocities .................................................................................................. 58

Figure 2-15: Effect of material modulus on the normal force acting on the flexible
plates at various wind velocities ......................................................................................... 60

Figure 2-16: Chord angles of five flexible plates at various wind velocities. ................. 61
Figure 2-17: Comparison of drag force obtained from the previous experiment and the present approximate method. ................................................................. 61

Figure 3-1: Wind loads acting on two kinds of wind turbines with and without a flanged diffuser at 60 m/s for various rated powers. ................................................................. 70

Figure 3-2: A general flexible flange idea: (a) conventional rigid flanged diffuser; (b) general flexible flange idea; (c) curves of open gap vs. wind velocity. ............... 72

Figure 3-3: Illustration of the idea of a bi-cantilever flange. ...................................................... 73

Figure 3-4: A compact-type diffuser with a self-adaptive flange. ........................................ 74

Figure 3-5: Flowchart of the iterative CFD and structure coupling analyses. ............ 75

Figure 3-6: Self-adaptive flange model used in the structural analysis: (a) front view and dimensions, and (b) 3-dimensional view and loading and boundary conditions. .... 77

Figure 3-7: Computational domains, boundary conditions and grids: (a-1) diffuser with a rigid and flat flange, (a-2) related grids around the flanged diffuser, (b-1) diffuser with a typical deformed self-adaptive flange, and (b-2) related grids around the flanged diffuser. ............................................................................................................................. 79

Figure 3-8: Computational domain used by a previous research [13] for the comparative simulation (V0 = 5 m/s, D = 0.2 m). ................................................................. 80

Figure 3-9: Effects of different grid numbers and Reynolds numbers on streamwise velocity and pressure coefficient distributions along the axisymmetric axis in the case of the diffuser with a rigid and flat flange (Fig. 3-7a): (a) streamwise velocity and (b) pressure coefficient. ................................................................................................................................. 82

Figure 3-10: Streamwise velocity and pressure coefficient along the axisymmetric axis in the present comparative simulation are compared with those obtained from the previous research Abe et al. [13] (V0 = 5 m/s, D = 0.2 m, L = 1.5D, h = 0.5D, φ = 4°): (a) streamwise velocity and (b) pressure coefficient. ................................................................. 83
Figure 3-11: A representative time history of the drag coefficient for the diffuser with rigid and flat flange at 30 m/s. ................................................................. 84

Figure 3-12: Velocity contour around the flanged diffuser in one period at 30 m/s. ..... 84

Figure 3-13: Flow fields around the flanged diffuser with a rigid flange (Fig. 3-7a) at 60 m/s: (a) velocity contour and (b) streamline. ................................................................. 85

Figure 3-14: Comparison of the drag forces (a) and drag coefficients (b) among the flange, the cylindrical part, and the total flanged diffuser obtained from the CFD analyses for the diffuser with a rigid and flat flange (Fig. 3-7a) at various velocities. .... 86

Figure 3-15: Flow fields around the diffuser with a bended self-adaptive flange at 60 m/s: (a) velocity contour and (b) streamline. ................................................................. 87

Figure 3-16: Time history of the drag coefficient for the diffuser with the self-adaptive flange at 30 m/s. ........................................................................................................ 87

Figure 3-17: Streamwise velocity and pressure coefficient distributions along the axisymmetric axis in the flanged diffusers with a rigid and flat and with a bended self-adaptive flange: (a) streamwise velocity and (b) pressure coefficient. ................. 88

Figure 3-18: Wind velocity distributions on the rotor planes, at 60 m/s, of two flanged diffusers with either a rigid and flat flange or a bended self-adaptive flange, respectively. ........................................................................................................................................ 89

Figure 3-19: Configurations of deformed upper front plate of the self-adaptive flange at various wind velocities. ........................................................................................................ 90

Figure 3-20: Full image of the diffuser with a self-adaptive flange at 60 m/s. .......... 91

Figure 3-21: Drag ratio of the diffuser with a self-adaptive flange at various wind velocities. ........................................................................................................ 92

Figure 4-1: A small compact-type diffuser with a self-adaptive flange. ............... 99
Figure 4-2: Schematic of the mold for the one-fourth of the flanged diffuser: (a) sectional division of the flanged diffuser, (b) construction of the mold skeleton, (c) design view of the mold, (d) dimensions for bolt holes and side plate.......................... 100

Figure 4-3: Construction of the mold skeleton for the one-fourth of the diffuser........... 102

Figure 4-4: Schematic of construction for fabrication, (a) Sectional view of the structure, (b) enlarged view for the detail construction, (c) combined layer 1, (d) combined layer 2.................................................................................................................. 104

Figure 4-5: Surface modification of the mold and fabrication preparation for one-fourth of the diffuser................................................................................................................................. 106

Figure 4-6: Curing one-fourth of the CFRP diffuser, (a) autoclave used for curing one-fourth of the diffuser, and (b) cure cycle used in fabrication................................................................. 107

Figure 4-7: Fabricated CFRP diffuser, (a) one-fourth of the diffuser, and (b) semi-finished total diffuser................................................................................................................................. 108

Figure 4-8: Construction of support system for the diffuser, (a) total structure, (b) fixture nut, (c) pole and base, and (d) truss.................................................................................................................. 110

Figure 4-9: Construction of the diffuser with rigid flat flange, (a) semi-installed flanged diffuser, (b) plastic flat flange, and (c) finished diffuser with rigid flat flange. 111

Figure 4-10: (a) Model of one-fourth of the semi-finished diffuser with self-adaptive flange, (b) shape of one-fourth of the ring with multiple holes, and (c) construction of the diffuser with self-adaptive flange.. .................................................................................. 113

Figure 4-11: Construction of the diffuser with self-adaptive flange, (a) four fixed ring quarters, (b) front and rear plates cut from plastic sheets (PVC and PP 0.2mm), (c) four self-adaptive flange quarters, and (d) finished diffuser with self-adaptive flange.. ...... 114
Figure 4-12: Experimental set-up of force detector and flanged diffuser, (a) three-component force detector connected with fixture, (b) semi-finished set-up, (c) specific cover for force detector, and (d) finished set-up for measuring. ........................................ 115

Figure 4-13: Experimental arrangement for measuring wind velocity and pressure coefficient, (a) a traverse system equipped with static pressure tube and hot-wire probe, (b) schematic images of measurement positions. ................................................................. 116

Figure 4-14: Self-adaptive flange model used in the structural analysis ......................... 118

Figure 4-15: Computational domains, boundary conditions, and grids: (a-1) diffuser with a flange before deformation, (a-2) related grids around the flanged diffuser, (b-1) diffuser with deformed flange plates, and (b-2) related grids around the flanged diffuser. ................................................................................................................................. 120

Figure 4-16: Comparison of the drag forces of the flange, the cylindrical part, and the total flanged diffuser obtained from the CFD analyses and wind tunnel experiment for the diffuser with a rigid and flat flange (Fig. 4-15a) at various velocities .................. 121

Figure 4-17: Flow fields around the flanged diffuser with a rigid flange (Fig. 4-15a) at 20 m/s: (a) velocity contour and (b) streamline ................................................................. 122

Figure 4-18: Flow fields around the diffuser with a bended self-adaptive flange at 20 m/s: (a) velocity contour and (b) streamline ................................................................. 123

Figure 4-19: Configurations of deformed upper front plate of the self-adaptive flange at various wind velocities ................................................................. 124

Figure 4-20: Full image of the diffuser with a self-adaptive flange at various velocities obtained from experiments and simulations: (a) experiment: 2 m/s, (b) experiment: 10 m/s, (c) experiment: 24 m/s, (d) simulation: 2 m/s, (e) simulation: 10 m/s, (f) simulation: 24 m/s, and (g) schematic of the diffuser with bended flange .................. 125

Figure 4-21: The variation of drag force acting on the diffuser with self-adaptive flange versus to different velocities: (a) experiment, (b) CFD simulation .......................... 127
**Figure 4-22:** Experimental comparison for velocity and pressure coefficient distributions on the radial direction for the diffusers with either a rigid and flat flange or self-adaptive flange at $V_0 = 10$ m/s: (a) velocity, (b) pressure coefficient.............. 129

**Figure 4-23:** Experimental comparison for velocity and pressure coefficient distributions in the radial direction for the diffusers with either a rigid and flat flange or self-adaptive flange at $V_0 = 20$ m/s: (a) velocity, (b) pressure coefficient.................. 131

**Figure 4-24:** Numerical comparison of velocity and pressure coefficient distributions in the radial direction for the diffusers with either a rigid and flat flange and or self-adaptive flange at $V_0 = 20$ m/s: (a) velocity, (b) pressure coefficients.............................. 133
LIST OF TABLES

Table 1-1: Installed wind power capacity in Asia (MW) [1]. ......................................................... 5

Table 2-1: Geometries of five flexible thin plates.. ................................................................. 51

Table 2-2: Material constants of three materials........................................................................ 51

Table 3-1: Stacking sequences and thickness of the front and rear plates......................... 78

Table 3-2: Material properties of the unidirectional CFRP lamina.. ................................. 78

Table 4-1: The stacking sequence of CFRP laminates for flanges.. ................................ 106

Table 4-2: Material properties of the plastic sheets used...................................................... 113

Table 4-3: Effects of the grids number and Reynolds number on the calculation results.. ................................................................. 122
LIST OF PUBLICATIONS


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