The value of production from livestock industry in the U.S. has been growing every year and the poultry industry is prominent among the livestock industries. Though the annual production has been increasing every year, it has been reported that the broiler industry continues to face issues which tend to make the production process less economical. The most common problems faced by the industry are bird mortality and retarded growth due to heat-stress or degraded animal environment. Another rising concern to the poultry industry is the emissions from animal feeding operations (AFOs). Ventilation plays an important role in establishing a suitable environment and in quantifying emissions. Numerous field and lab-scale studies have been carried out to determine the optimum ventilation rate. However, the poultry industry is far from achieving an optimal growth environment. This study evaluated the performance of a dedicated Poultry Engineering Chamber complex designed to conduct studies on ventilation, air quality and animal welfare. The performance evaluation of the Poultry Engineering Chamber complex was carried out by:

1. Evaluating the ventilation rate and repeatability of the predesigned chambers for poultry welfare and air quality studies.
2. Evaluating the velocity distribution in the animal occupied region at different operating configurations.
3. Modeling the air flow using CFD technique to simulate, virtually, the indoor microclimate and validating the models using experimental data.

The flow rate and pressure drop measurements across each chamber at six different blower RPMs indicated the effect of structural geometry on air flow. The average difference in flow rates among chambers was 5.06% at 600 RPM. Chamber 1 and chamber 3 had slightly higher flow rates. However, ANOVA showed no significant difference in mean flow rates among different chambers. Chambers 5 and 6 had lower flow rates compared to the rest of the chambers.
Both, chambers 5 and 6 had similar flow rates and hence, the RPM could be linearly increased to obtain constant flow rates across all chambers. The damper flow rates were uniform for all chambers indicating an equal amount of fresh air intake. 3-D velocity measurements in the core chamber showed higher velocities at bird height near the inlet and lower values away from the inlet due to the interference by the feeders.

Further, the flow in the core chamber was simulated using CFD and the results were validated using field measurements. Different boundary conditions were considered in the study. Two different blower rotary speeds viz. 600 and 1200 RPM were simulated. The velocity inlet and pressure outlet boundary conditions were found to simulate the air velocity with least error. Statistical analysis (ANOVA) showed no significant difference between measured and simulated results. Out of 20 measurement points 18 points had error ($E_b$) value less than or equal to 10%. 50% of the points measured had error values less than or equal to 5%. The position (LB-Left Back) away from the inlet at bird height (0.20 m) had the highest $E_b$ of 13.59 %. This was due to the effect of the feeders which are placed in the line of flow. The normalized mean square error (NMSE) was 0.007 indicating good agreement between the simulated and measured values. Bird models (with 11 and 30 birds) were simulated as simple spheres with heat generation using CFD to assess the qualitative effect of birds on air flow inside the core chamber. Certain regions in the animal occupied zone had lower air velocities on an average and therefore a higher mean surface temperature of bird models in those regions. More accurate bird models would provide a better understanding of the heat exchange between the birds and the surroundings.
Flow Testing and CFD Modeling of Poultry Engineering Chamber

by

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DEDICATION

To my grandfather
BIOGRAPHY

Aditya Shivkumar received his Bachelor’s degree in 2012 in Civil and Environmental Engineering from Visvesvaraya Technological University, India. Later, he began his graduate studies at North Carolina State University in the department of Biological and Agricultural Engineering. His topics of interest include air pollution modeling, computational methods in environmental control and environmental data analysis. In his spare time, Aditya enjoys reading philosophy and economics.
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CHAPTER 1 INTRODUCTION

In the United States (US), the value of production from livestock industry was estimated to be $71 billion in 2013 (USDA, 2013). Greater production rates are being achieved by the implementation of modern techniques in animal welfare and animal housing facilities which make them one of the most important aspects of livestock industry. Figure 1 shows the growth in production of various livestock industries.

According to the National Chicken Council (NCC, 2014) the annual production (ready to cook basis) of broilers in 2013 was estimated to be 16.9 megatons with an annual growth rate of 2.1%. Consequently, for the industry to continue to remain sustainable, high production rates have to be achieved by striving to establish optimal conditions for promotion of broiler growth.

Animal welfare study aims at establishing suitable conditions for optimal growth of animals by monitoring closely, various responses of animals to the surrounding environment. Precise observations and analysis of animal responses can be made in a facility with controlled environment, which will eventually help animal welfare studies and increase animal production.
An important factor which comes to picture, in the context of growing animal production and animal housing facilities, is the consequent air pollutant emissions. Studies have shown that the generation of pollutants in the form of air emissions from the animal housing facility have the potential to compromise animal performance, cause human health problems and also degrade atmospheric air quality. Animal production facilities are under pressure to adopt mitigation strategies that reduce air emissions. Hence, precisely quantifying the emissions from the animal production activities, assessing their effects and developing mitigation measures require reliable emission data and emission characteristics. This can be obtained under strictly controlled conditions of ventilation rates, temperature, humidity and other environmental variables.

1.1 Thermal environment and animal welfare

Animals are affected by the existing thermal environment. The thermal conditions prevailing in the animal occupied zone greatly affect animal performance. In case of poultry, which are homoeothermic, when the effective environmental temperature (EET) is within the thermo-neutral zone (TNZ), core body temperature \( t_b \) of adult chickens is maintained between 41.2 °C and 42.2 °C by thermoregulatory mechanisms with minimal effort. According to Tao and Xin, (2003) EET is the result of integrating the environmental factors, including air temperature \( T \), relative humidity \( RH \), air velocity \( V \), solar radiation and precipitation. An air temperature of about 21.1 °C is considered to be the thermoneutral zone for broilers (Dozier et al., 2007). If the thermoregulation mechanism is insufficient to maintain homoeothermy, \( t_b \) begins to rise and eventually leads to death from heat exhaustion. This phenomenon is usually termed as heat stress.

Heat stress is an important problem in the poultry industry. A past study by St-Peirre et al., (2003) reported the economic losses incurred by different livestock industries in the US. The study estimated that the total annual economic loss to the US livestock industry was in the range of $1.69 to $2.36 billion out of which $128 to $165 million occurred in poultry industry.
Heat stress not only causes bird loss, but in less severe cases, also causes slower and less efficient growth of birds, resulting in reduced average daily bird weight gain.

A recent study by Mack et al., (2013) showed that birds subjected to heat stress conditions spend less time feeding, more time drinking and panting, as well as more time with their wings elevated, less time moving or walking, and more time resting. Heat stress can affect the reproductive function of poultry in different ways. In females, heat stress can disrupt the normal status of reproductive hormones at the hypothalamus which in turn produces hormones which affect the ovary, leading to reduced systemic levels and functions (Lucas et al., 2013). Another recent study by Sohail et al. (2012) reported that broilers subjected to chronic heat stress had significantly reduced feed intake (16.4%), lower body weight (32.6%), and lower feed conversion ratio (+25.6%) at 42 days of age. Studies by Dai et al. (2012) and Imik et al., (2012) demonstrated that the heat stress is associated with depreciated meat chemical composition and lower broiler quality.

In general, animal welfare is a multifactorial condition based on healthy and disease free surroundings, ability to perform specific behaviors and to cope with social and environmental conditions. Lucas et al., (2013) reported that heat stress is often times not experienced in isolation but is aggravated by other stressors like past experience and social interactions. Several recent studies have reported the role of genetics in determining the intensity of heat stress (Mack et al., 2013; Soleimani et al., 2011; Felver-Gant et al., 2012). Nevertheless, more studies are required in this area to ascertain the influence of genetics (Lucas et al., 2013). Currently, the most convenient and effective way to deal with heat stress is by maintaining an environment suitable for broiler growth.

It is evident that an estimate of physiological response of broilers to high temperatures is needed to better understand the impacts of heat stress. This will eventually help increase production and animal welfare through improved design of production facilities (Yanagi et al., 2002). Among several environmental indices used to assess the effects of thermal conditions on poultry and other livestock animals, Temperature and Humidity Index (THI) is the most popular and has been extensively studied and developed for various
species (Tao et al., 2003). Purswell et al., (2012) reported that bird performance declined significantly as THI exceeded 21°C in case of heavy broilers (body weight greater than 3.2 kg).

However, when the ambient temperature and humidity are high, evaporative cooling systems fail to regulate the temperature of the incoming ventilation air and thus do not help in reducing heat stress. Thus, a uniform and high air velocity may be very effective in enhancing the thermal comfort of the birds under high humidity condition. With high air velocity it is possible to achieve a higher convective heat transfer (Simmons et al., 1997). To investigate the relative importance of air temperature, humidity and velocity temperature-humidity-velocity index (THVI) was developed (Tao et al., 2003). However, the THVI developed was based on the classification of acute conditions and hence, does not consider the sub-lethal effects that can harm the bird performance over time.

1.2 Air quality in animal housing facilities

The main pollutants of air within the animal house are particulate matter (PM), ammonia (NH₃), carbon dioxide and carbon monoxide (Ross Manual, 2009). Ammonia is the primary aerial pollutant in poultry production houses, resulting from biodegradation of bird feces. Also, NH₃ is considered to be a particulate precursor due to which poultry and other livestock industries have been drawing the attention of local and federal air quality regulatory bodies (Xin et al., 2003). Consequently, animal production facilities are under pressure to adopt mitigation strategies that reduce air emissions. Air pollutant emissions as well as odor emission from animal buildings can affect the well-being of animals, workers in the buildings and neighbors. Many other studies have demonstrated that NH₃ is the main component of emissions from animal production houses and a significant contributor to health and equipment deterioration (Bull and Sutton, 1998; Cupr et al., 2005; Webb et al., 2005). Ammonia concentrations beyond 25 ppm inside poultry facilities reduce the growth rate of broilers. In smaller concentrations NH₃ causes respiratory diseases and irritation in the eyes of broilers hampering overall growth (Ross manual, 2009).
In poultry production, litter is the source of volatilizing \( \text{NH}_3 \) and its management is one of the key factors affecting emission rate and bird health. The most common technique adopted to mitigate \( \text{NH}_3 \) emissions is by maintaining litter moisture levels.

Hence, RH plays an important role in maintaining air quality and therefore has to be controlled. Suitably designed ventilation systems are used to control the litter moisture levels and also the RH of the surrounding. More importantly, ventilation management plays a significant role in controlling moisture. Determination of \( \text{NH}_3 \) emissions requires accurate measurement of \( \text{NH}_3 \) concentration in the air leaving the building and the volume flow rate of air being discharged. While this seems simple in concept but in practice, both concentration and ventilation are difficult to measure accurately under commercial poultry house conditions. Despite the advances in ventilation systems inside poultry houses, it is difficult to analyze their performance, mainly due to changes in ventilation patterns and use of different equipment in the same house (Vranken et al., 2005; Casey et al., 2008). The uncertainty in the measurement of gas emissions from the houses is mainly due to the ventilation rate (Gates et al., 2009). Other authors have reported bird age, management practices and nutrition as factors contributing to the variation in gas emissions from poultry houses (Freguson et al., 1998; Casey et al., 2004; Wheeler et al., 2006). It can be seen that an environment with higher degree of control over RH, temperature and air flow rates is required to estimate the air emissions associated with animal production facilities. With accurate ventilation rate measurements and gas emission concentrations, emission rates may be calculated to support regulatory concerns.

1.3 Commercial broiler housing

Generally, the existing animal houses are large and ventilated either naturally or more commonly by fans. Commercial broiler houses are, usually, 152.4 m-167.64 m (400-600 feet) in length and 12.19 m-15.34 m (40 to 60 feet) in width. Numbers of birds vary from 18000 to 32000 based on housing dimension and season.

In order to achieve required in-house thermal conditions through high air velocity
(‘Wind chill’ effect) broiler houses are generally equipped with tunnel ventilation systems consisting of multiple fans of known capacity on end walls or side walls or both which are controlled by environmental controller that measure house temperatures.

Evaporative cooling pads are used in mechanically ventilated houses to lower the temperature of the incoming air in case of high ambient temperatures. In case of naturally ventilated houses curtains and screens are installed to control the air flow and foggers and misters are used to control temperature.

Air velocity distributions in commercial farms have been evaluated (Boon and Battams, 1988; Lee et al., 2003; Wheeler et al., 2003). However, there are several issues with animal houses:

- Air velocity and hence the temperature uniformity is essential for maintaining the required thermal environment as a whole. None uniformity encourages the animals to migrate towards ventilated zones which may already be crowded and thus eventually increase heat stress (Taler et al., 2002). In case of broiler houses, which usually are large structures, the air is, in general non-uniform. There is little or no control over the environment in case of naturally ventilated houses.

- The large structure of the animal houses also makes it necessary to have high number of monitoring points in order to obtain a reliable measurements of distributions of environmental parameters (T, RH and V) making the study less economical.

- It is very difficult to establish sensors at bird level due to the bird density and bird movement inside large commercial broiler houses.

Thus building designs and control systems to create suitable thermal environment (specifically air velocity) in animal occupied zones in a commercial farm has not been completely achieved (Bustamante et al., 2013).

To obtain more specific and accurate measurements, researchers have used dedicated lab facilities and research chambers to conduct studies on livestock animals. These facilities allow greater degree of control over the microenvironment inhabited by the species (broilers or other livestock animals). A research chamber makes it convenient to simulate conditions close to real world scenarios.
1.4 Existing poultry research chambers

As a suitable environment is required to quantify and optimize the variables such as air T, RH and V to create a specific thermal environment, many attempts have been made to simulate the conditions similar to a commercial farm but with a greater degree of control (Begermann et al., 2006). A number of research chambers have been designed to house the birds and maintain a controlled environment (Begermann et al., 2006; Leftcourt, 2001). A research chamber is an advantageous tool in multiple scientific fields. Chambers enables researchers to create unique circumstances that simulate real-world conditions. Challenges faced in the field or real world can be recreated and fully controlled in enclosed chambers allowing specific variables to be monitored and assessed as other factors fluctuate or change. More literature regarding the existing research chambers is provided in chapter 2.

However, it has been observed that all the research chambers used were designed either for measuring certain physiological stimuli alone or during transportation of broilers (Mitchelle et al., 2001; Hamrita et al., 1998; Brown-Brandl et al., 2003, 2005). The chambers designed so far for research purposes can accommodate very few birds and hence neglect the effect of flocking of birds and their behavior. Although, most of the chambers have environmental control systems they lack the ability to deliver and study the effects of uniformity of air flow.

Thus, considering the shortcomings of certain existing chamber designs the current study aims to assess a poultry chamber complex which can simulate the environment close to that in commercial broiler houses with greater control over the environmental parameters, namely, T, RH and V. Also, the study evaluated the performance by modeling of air flow through the chamber at desired thermal and air speed configurations.

1.5 Application of CFD in animal housing studies

In order to realize wholly the advantages of a controlled environment it is necessary to have in place a precise quantitative understanding of the environmental variables to achieve optimal results.
A number of studies have quantified the environmental parameters such as T, RH and V, with fair degree precision, for the optimal growth of broiler chickens (Lott et al., 1998; May et al., 2000; Yahav et al., 2001; Simmons et al., 2003). The amount of information required to fully quantify these variables is dependent on the physical experimentation and the precision of the tools used. Though direct measurements are most reliable, it is increasingly complex and time consuming to do a comprehensive analysis which often involves high cost implications. Also, a large number of monitoring points are required for the measurements to be representative. Mathematical models and numerical modeling methods such as Computational Fluid Dynamics (CFD) can be effectively used to precisely quantify various environmental variables in simulated environment. Various ventilation configurations can be examined and designed by simulating these conditions virtually. However, these numerical models can only supplement direct measurements and hence, should be validated with field measurements before using the models as tools for decision making (Blanes-Vidal et al., 2008; Bustamante et al., 2012).

Computational fluid dynamics techniques are increasingly being used in bio-systems and agricultural engineering (Norton et al., 2007). CFD can be used to numerically simulate the ventilation conditions and hence develop spatial and time dependent solutions for pressure, velocity, temperature and pollutant transport. Various conditions and variables can be tested using CFD which makes it easy to understand and study potential design techniques.

Studies have included the use of CFD to simulate indoor air flow in agricultural and livestock buildings (Norton et al., 2007). It is also used to analyze the air flow in animal occupied zones (AOZ) (Wu and Gebermedhin, 2001) such as in case of experimental piggeries (Zhang et al., 1999; Morsing et al., 2002). Mistriots et al., (1997) demonstrated the use of CFD in commercial broiler houses. Pawar et al., (2007) and Zajicek et al., (2012) used CFD techniques to compare different air flow patterns and ventilation designs in the context of air velocity and pollution concentration. Blanes-Vidal et al., (2008) used CFD to supplement experimental air flow measurements in a commercial poultry building.
Various benefits of using numerical modeling have been summarized by Norton et al., (2007). More literature regarding the application of CFD in agriculture is provided in Chapter 3.

In this study, an attempt was made to model the air flow in a research chamber using CFD technique to supplement field observations. Suitable boundary conditions were used to simulate the effect of environmental parameters set for the optimal growth of broiler chickens.

1.6 Research objectives

Two things are desirable for carrying out animal air quality and animal welfare study efficiently viz. (1) an environment with necessary control over temperature, humidity and air velocity and a viable system to quantifying these parameters precisely and with ease (2) Simulating environment similar to commercial poultry houses by using compact research chambers which allow testing over a flock rather than one or few test specimens.

This study assesses such an environmentally controlled research chamber by testing the stability of the chamber system for different flow regimes, studying the effect of varying velocities and structural features on flow characteristics. An attempt was made to model the air flow in the research chamber using CFD technique to supplement animal welfare studies and air quality studies. Suitable boundary conditions were used to simulate the effect of environmental parameters (T, RH and V). The effect of chamber design and its configuration in establishing a uniform flow field was assessed.

The specific objectives of this study can be listed as follows:

1. Evaluate the utility and effectiveness of a predesigned chamber complex for poultry welfare and air quality studies.
2. Evaluate the indoor climate at different operating configurations (ventilation rates, velocity distribution, etc.).
3. Numerically model the air flow using CFD technique to simulate the indoor microclimate at different operating configurations and validate using experimental data.
The simulated results after being validated and refined will potentially be used to set up a fully automated system for the chamber. The bigger picture is to establish an automated control system for the chamber which can, effectively, be used to conduct research studies on poultry environmental control, air quality, animal welfare.
CHAPTER 2 PERFORMANCE EVALUATION OF THE POULTRY ENGINEERING CHAMBERS: FIELD TESTING

2.1 Introduction

The importance of obtaining accurate field measurements has been emphasized repeatedly in the literature. As described in chapter 1 there are certain limitations for carrying out field studies but nevertheless field data is required to bolster the results obtained from other indirect methods. Measuring the ventilation rate and the velocity in the proximity of the test subject with the help of sensors or hand held instruments is a common methodology adopted by the research community. Research chambers or lab scale control systems can help negate the use of a large number of sensors and thereby making the study more economical. With a controlled environment it is possible to obtain information most accurately. This enables researchers to carry out control studies and analyze specific physiological responses by changing the circumstances and environment in the chamber. In case of poultry, studies which reflect the phenomena in a commercial poultry farm can replicated in a controlled environment.

Many attempts have been made to simulate the conditions similar to a commercial farm but with a greater degree of control (Begermann et al., 2006). A number of research chambers have been designed to house the birds and maintain a controlled environment (Begermann et al., 2006; Leftcourt, 2001). Challenges faced in the field or real world can be recreated in enclosed chambers allowing specific variables to be monitored and assessed as other factors fluctuate or change.

Environmental chambers have been used to study the physiological studies of poultry mainly to measure the deep body temperature in response to various external factors such as ambient temperature and relative humidity (Hamrita et al., 1998; Lacey et al., 2000b). Brown-Brandl et al. (1998) provided description of a ‘within-chamber’ type of test room which was designed to act as an indirect calorimeter. The outer room had an air unit that
could create an air temperature range from 0 to 100°C. The inner room was divided into three small chambers with heating units. The total animal space was 1.2 m³ all chambers put together. The walls were built of Plexiglas and the chambers housed a humidifier. Thus, the attempt was to establish more control over temperature and humidity. This system was further improved by Brown-Brandl et al. (2005) that controlled both the temperature and humidity with air handling units. The chambers were thoroughly insulated with sandwiched panels (R-22) using fiberglass reinforced panels and Styrofoam. The physical layout of the system had 4 environmental chambers laid side by side. The compartments were designed to be movable and easy to maneuver. The compartments also included a weighing feeder and a cup waterer.

Mitchell and Kettlewell (1998), Mitchell et al. (2001a) and Hunter et al. (1999) reported on chamber studies with a view to study the physiological responses of poultry during transport. All the three studies used a setup similar to transport crates with wire mesh on the top. Mitchell and Kettlewell (1998) controlled the temperature between 22- 30°C. Mitchell et al. (2001a) established temperature ranges between 10-35°C with a setup similar to the basic setup. Hunter et al., (1999), in addition to 4 temperature set points, included an air velocity of 0.7 m/s throughout the experiment representing partially ventilated transport systems.

Chepete and Xin (2000) used a large environmental chamber with a separate air conditioning unit to study the effect of cooling laying hens by intermittent partial surface sprinkling. The chamber was essentially used to circulate conditioned fresh air to the two animal compartments through air ducts. Air distribution panel were installed before the compartments. The compartments included a spray nozzle for sprinkling the birds and also, the sensors were placed at bird height for more accurate data. An infrared camera was also installed to monitor the birds. Yanagi et al., (2002a, b,c) conducted chamber studies to assess the effect of heat stress in poultry. Three large environmental chambers were used out of which one housed two testing chambers and the other two served as air conditioning units. The two testing units were placed downwind of a wind tunnel. The setup included an air-
straightener and a variable speed fan to change air velocity. An infrared camera was used to capture thermal footprint of the birds to study the behavioral responses.

Aerts et al., (2003) used respiration chambers to study the heat production in broilers and the factors like lighting and temperature. Six chambers of dimensions 0.3 by 0.55 by 0.5m were used in the study. Temperature was measured to an accuracy of 0.2°C with modern instruments like platinum resistant temperature detectors.

It can be seen that all the research chambers used were designed either for measuring certain physiological stimuli alone or during transportation of broilers. The chambers designed so far for research purposes can accommodate very few birds and hence, neglect the effect of flocking and their collective behavior. Nonetheless, most of the chambers have environmental control systems, but they lack the ability to deliver and study the effects of uniform air flow.

Thus, considering the short comings of the existing chambers a Poultry Engineering chamber complex was designed and built at North Carolina State University (NCSU) which can simulate conditions similar to commercial broiler houses with greater control over the environment (Wang-Li, 2013). The performance evaluation of this chamber complex was undertaken in this study. Both field measurements and CFD modeling were adopted to study the chamber characteristics. The current chapter reports the field measurements of the chamber system flow rates, ventilation rates and velocity profiles in the animal occupied zone.

A number of instruments are available to measure air flow and direction with high accuracy. Hot-wire anemometers, vane anemometers, sonic anemometers and particle image velocimetry are some of the most common instruments used to measure air velocity. Hot-wire and hot film anemometers are fragile whereas vane anemometers cannot measure low velocities. Latest technologies involving laser and visualization methods like PIV are too expensive. Thus, sonic anemometers are the best choice when it comes to obtaining high accuracy economically. 2-D and 3-D sonic anemometers allow measurement of different components of velocity.
The sonic anemometer was developed to measure surface layer wind and temperature turbulence. More recently 3-D ultrasonic anemometers were extensively used to measure air flow in green houses and livestock buildings. A 3-D ultrasonic anemometer works based on the principle of sound speed in air. The speed of sound is increased in the direction of the wind or air. Thus, the time of flight of an ultrasonic pulse between two transducers is recorded and then, the velocity is calculated using the travel times along the path length.

However, using specific sensors depends on various factors including desired accuracy, economic feasibility, sensor dimensions and availability. Hot-wire anemometer and sonic anemometers were used to carry out flow measurements in this study. A complete list of instruments used is provided in the instrumentation section.

2.2 Methodology

2.2.1 Description of the Poultry Engineering Chambers

There were 6 chambers built side by side and this group of chambers was defined as a chamber complex. Each chamber could be operated individually and was equipped with separate controls and sensors. The chambers were identical. The figure 2.1 shows the longitudinal cross section of a chamber system.
The following are the main components of the chamber:

- **Blower house:** The structure housed a backward curved blade belt drive centrifugal blower with variable speed motor powered by 10 HP motor. Detail specifications of the blower are provided in Appendix-I. The blower speed was operated inclusively between 200-1200 RPM (10-60 Hz). The blower exhaust continued as a smooth conduit which enters the chamber. A damper was provided for intake and circulation of fresh air. The damper opening was controlled manually at 10%, 30%, 50% and 70% opening. (Figure 2.1)
• **Conditioning chambers:** The conditioning chamber was provided to house heating/cooling units and humidifiers (to be installed in the bottom part of the chamber in the future) for conditioning the incoming air. No air conditioning units were present during this study. The top part of the conditioning chamber serves as an exhaust duct.

• **Turnaround:** This component was used to direct the flow towards the core chamber and has a smooth transition giving rise to minimum turbulence.

• **Core chamber:** This is the chamber where animals are kept. The chamber has accessories like feeders, waterlines, lights etc. The chamber can be accessed through maintenance door.

  • **Exhaust duct:** Air from the core chamber flows towards the blower through the circular exhaust duct. Flow testing (velocity, pressure and gas sampling) was performed by measuring air flowing through this chamber.

  The dimensions of the core chamber air inlet and outlet openings were 2.43 m by 0.91 m (8ft by 3ft). The core chamber was essentially a cube with a length of 2.43m (8ft). The bedding material used was fresh wood chips and the thickness of the bedding was approximately 6.5cm (2.5 inches). The core chamber was equipped with 4 feeders and a water line with six drinkers and a water level regulator. The lighting was similar to a commercial poultry house with automatic switches. The core chamber also had a set of thermocouples to measure dry and wet bulb temperatures continuously. A data acquisition system was used to process the signals from the sensors from all the chambers. The blower controls and the data acquisition system were housed in a separate control room adjacent to the chamber complex.
2.2.2 Instrumentation

Detailed description of all the instruments used is shown in Table 2.1.

Table 2.1 Description of the instrumentation

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Sensor description</th>
<th>Vendor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air velocity</td>
<td>Hot film anemometer</td>
<td>Dwyer Inc.</td>
</tr>
<tr>
<td></td>
<td>Range -0 to 75 m/s, Accuracy +/- 2%</td>
<td></td>
</tr>
<tr>
<td>Air velocity – 3D</td>
<td>Ultrasonic Anemometer, Range 0-40m/s, Accuracy +/- 1 to +/-3%</td>
<td>R.M. Young Co.</td>
</tr>
<tr>
<td>System pressure drop</td>
<td>Digital manometer kit, 0-10 in of water, Accuracy +/- 1%</td>
<td>Dwyer Inc.</td>
</tr>
<tr>
<td></td>
<td>Magnehelic gauge, 0-4 In. of water, Accuracy +/- 2%</td>
<td></td>
</tr>
<tr>
<td>Temperature &amp; Relative humidity (RH) (core chambers)</td>
<td>Thermocouples, T type HOBO U-23-004 T/RH sensors, Range- 0-50 °C Accuracy- +/- 0.2°C</td>
<td>Onset Computer Corp.</td>
</tr>
</tbody>
</table>
2.2.3 Experimental design

2.2.3.1 Phase 1 testing

Phase-1 testing was performed to develop a set of chamber system flow curves that describe the operating characteristics of the chamber flow system. The flow rates and differential pressure across the chamber system were obtained for the operating configurations of the blower rotary speed (RPMs) and damper opening combination given in table 2.2. Figure 2.2 shows flow measurement instrumentation and location. The results were tabulated in the form of a database and the curves were developed using the recorded data.

<table>
<thead>
<tr>
<th>Damper Opening (%)</th>
<th>Flow Area Covered (%)</th>
<th>Blower RPM</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>0</td>
<td>200,400,600,800,1000,1200</td>
</tr>
<tr>
<td></td>
<td>25</td>
<td>200,400,600,800,1000,1200</td>
</tr>
<tr>
<td></td>
<td>50</td>
<td>200,400,600,800,1000,1200</td>
</tr>
<tr>
<td></td>
<td>75</td>
<td></td>
</tr>
<tr>
<td>30</td>
<td>0</td>
<td>200,400,600,800,1000,1200</td>
</tr>
<tr>
<td></td>
<td>25</td>
<td>200,400,600,800,1000,1200</td>
</tr>
<tr>
<td></td>
<td>50</td>
<td>200,400,600,800,1000,1200</td>
</tr>
<tr>
<td></td>
<td>75</td>
<td></td>
</tr>
<tr>
<td>50</td>
<td>0</td>
<td>200,400,600,800,1000,1200</td>
</tr>
<tr>
<td></td>
<td>25</td>
<td>200,400,600,800,1000,1200</td>
</tr>
<tr>
<td></td>
<td>50</td>
<td>200,400,600,800,1000,1200</td>
</tr>
<tr>
<td></td>
<td>75</td>
<td></td>
</tr>
<tr>
<td>70</td>
<td>0</td>
<td>200,400,600,800,1000,1200</td>
</tr>
<tr>
<td></td>
<td>25</td>
<td>200,400,600,800,1000,1200</td>
</tr>
<tr>
<td></td>
<td>50</td>
<td>200,400,600,800,1000,1200</td>
</tr>
<tr>
<td></td>
<td>75</td>
<td></td>
</tr>
</tbody>
</table>

The various system pressure drops were created using a set of wooden strips designed. Figure 2.3 shows the setup. The strips were of equal size and the pressure drop was adjusted by changing the flow area coverage uniformly inside the collimation chamber in the
bottom half of the conditioning chambers. The dimensions of the wood strips and the different area coverage in (%) used are given in table 2.3.

![Image of Hot-wire anemometer and measurement location](image)

**Figure 2.2** Hot-wire anemometer and measurement location

<table>
<thead>
<tr>
<th>Flow Area Covered (%)</th>
<th>Spacing in meters (inches)</th>
</tr>
</thead>
<tbody>
<tr>
<td>None (as is)</td>
<td>NA</td>
</tr>
<tr>
<td>25</td>
<td>0.18 (7.3)</td>
</tr>
<tr>
<td>50</td>
<td>0.09 (3.84)</td>
</tr>
<tr>
<td>75</td>
<td>0.05 (1.95)</td>
</tr>
</tbody>
</table>

**Table 2.3 Spacing of wood strips for area coverage**
Figure 2.3 Area coverage to create various pressure drops in the flow system

Flow measurement method

The flow velocity was measured at the midpoint of the circular exhaust duct of each chamber at downstream of the conditioning chamber (top part) (see figure 2.1). The circular duct with smooth edges at entry and the collimating screens placed prior to the duct enforcing uniformity made it suitable for establishing measurement points in the duct. The measurement was carried using a hot-wire anemometer. Four measurement points were chosen to calculate the average flow velocity. The radial distances of the measurement points from the wall were as shown in table 2.4.
Table 2.4 Velocity measurement points in the circular exhaust duct

<table>
<thead>
<tr>
<th>Measurement position</th>
<th>Radial distance from wall in meters (inches)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C</td>
<td>0.216 (8.5)</td>
</tr>
<tr>
<td>R1</td>
<td>0.161 (6.375)</td>
</tr>
<tr>
<td>R2</td>
<td>0.107 (4.25)</td>
</tr>
<tr>
<td>R3</td>
<td>0.053 (2.125)</td>
</tr>
</tbody>
</table>

Establishing the measurement points deviated from the prescribed measurement method by ASHRAE (ASHRAE, 2009) due to following reasons:

1. Limitation associated with the length of the anemometer to points along the diameter. As a result, symmetric distribution of velocity was assumed.
2. Limitation regarding measurement position near walls – The position near walls obtained by ASHRAE method was found to be too small and hence was within the length of the probe holder. Also, the length of the ceramic filament of the anemometer was found to be comparable with the length obtained by ASHRAE guidelines which makes it hard in discerning the exact position of the probe.

In order to overcome the above limitations, the following steps were followed to obtain (calculate) the average velocity in the duct.

1. Velocities at various locations in a cross section of the exhaust duct (C, R1, R2, and R3) were measured as mentioned above.
2. Symmetrical distribution of velocity was assumed.
3. The velocities measured at different radial distances (C, R1, R2, and R3) were plotted against the radial distances to obtain flow profile curve.
4. An equation for describing velocity profile in the whole cross section area was obtained based upon the measured profile curve using curve fitting algorithm executed in MATLAB (Appendix –III)
5. Radial distances recommended by ASHRAE were used to determine velocities at those radial points using the above derived equation.
6. The velocities were averaged to get the average flow velocity.
7. Average velocity was then multiplied with cross-sectional area to obtain flow rate.

2.2.3.2 Phase 2 testing

Phase 2 testing was performed to measure the velocity profile in the core chamber. Air speed alone does not provide any insight about the spatial distribution of velocities. An air velocity measurement includes the direction of air flow along with speed. This enables a more comprehensive understanding of the air flow inside the core chamber.

Five 3-D ultrasonic anemometers were used to measure the velocity in the chamber at various locations and heights to understand the velocity gradients and lateral distribution. The instruments were setup inside the chamber and the parameters measured include:

- Velocity – U direction (direction perpendicular to flow)
- Velocity- V direction (direction along the flow)
- Velocity- W direction (vertical direction)
- Temperature (Sonic temperature)

A 4-channel data logger was used to record the velocity outputs from the anemometer. The frequency of the logger was set such that data was recorded every to 10 seconds i.e. 6 times per minute. The anemometers were powered using a portable rechargeable battery. The default settings of the anemometer were changed to suit the experimental design, which included serial outputs in the form of u, v, w, T (sonic temperature); where +u- wind from the east; +v- wind from the north; +w- wind blowing upwards (updraft).

Core chamber air velocity measurement method

A customized measurement method was adopted to conduct the phase-2 testing with a view to address the following issues:

1. Velocity profile resolution improves as the number of measurement points is
increased. This is also true with measurements at varying heights.

2. The simultaneous measurements of velocity at different locations require the anemometers to be setup and running simultaneously. Considering the dimensions of the chamber, placing multiple anemometers simultaneous may disturb the natural flow which in turn leads to observing unrealistic velocity values.

3. Placing multiple anemometers in different locations and heights with least disturbance can be achieved by sequentially performing tests with different spatial locations and same operating configurations. But using only one sensor at a time leads to inefficient utilization of available time and resources for performing the test.

Hence, a measurement approach which adequately captures the air flow characteristics satisfactorily and at the same time considers the time and resource requirements was designed.

The whole test consisted of velocity measurements at 5 different locations and 3 different heights at each location. Height 1 (0.20 m) was assumed to represent the height of a bird and hence, was termed as bird height. The positions were labeled according to their positions with respect to an observer facing the chamber in the direction of the flow (inlet). The summary of the measurement configuration is provided in table 2.5.

Table 2.5 Summary of 3-D velocity measurement configuration

<table>
<thead>
<tr>
<th>Position</th>
<th>Description</th>
<th>Height</th>
</tr>
</thead>
<tbody>
<tr>
<td>LF-Left Front</td>
<td>0.61m (2ft) from the left wall; 0.61m (2ft) from the inlet</td>
<td>H1- 0.20 m (8 inches) above the bedding (Bird Height)</td>
</tr>
<tr>
<td>RF-Right Front</td>
<td>0.61m (2ft) from the right wall; 0.61m (2ft) from the inlet</td>
<td>H2- 0.63 m (25 inches) above bedding</td>
</tr>
<tr>
<td>LB-Left Back</td>
<td>0.61m (2ft) from the left wall; 0.61m (2ft) from the outlet</td>
<td>H3- 1.21 m (4 feet) above the bedding</td>
</tr>
<tr>
<td>RB-Right Back</td>
<td>0.61m (2ft) from the right wall; 0.61m (2ft) from the outlet</td>
<td></td>
</tr>
<tr>
<td>C-Center</td>
<td>1.21m (4ft) from the inlet; 1.21m (4ft) from the left or right wall</td>
<td></td>
</tr>
</tbody>
</table>
The measurement design was as described below:

1. Two sonic anemometers were setup during the first run for a certain operating configuration. The two anemometers were placed apart diagonally in the chamber which realized the maximum separating distance which leads to minimum disturbance as shown in figure 2.4a.

2. The above instrument setup was maintained at 3 different heights (table 2.5) in the subsequent runs.

3. The locations of the anemometers were changed and placed on the other diagonal and step was repeated for this setup (figure 2.4b).

4. Finally, velocity at the center of the chamber (figure 2.4c) was measured using a single anemometer and the test was repeated for varying heights.

Figure 2.5 illustrates sensor setup for measurements at the bird height.

Figure 2.4 Velocity monitoring configuration and annotation; LB-Left Back, RF- Right Front, RB- Right Back, LF- Left Front, C-Center
2.2.3.3 Air flow balancing and damper flow rate

The chamber design is a recirculation system where the amount of air recirculated is controlled by the damper. The damper opening is in turn controlled by the linear movement of the rod attached to the damper opening in a hinged door like mechanism.

The need for circulating fresh air inside the chamber is fulfilled by the damper opening. Hence, the amount of fresh air intake was estimated by measuring the air flow at the damper outlet.

A narrow steel duct with smooth edges was designed conforming to ASHRAE guidelines (ASHRAE, 2009) and fabricated exclusively to measure the discharged air flow rate by the damper (figure 2.1) such that the amount of fresh air intake will be calculated accordingly.
The height and width of the extended damper duct had the same dimensions as of the damper outlet which is 0.54 m by 0.38 m (21.5 in by 15 in.). The duct was 3.65 m (12 feet) long which allowed the flow to fully develop and tend to become uniform. Figure 2.6 shows the duct and the measurement points.

![Figure 2.6 Damper flow testing- duct and measuring points](image)

**Measurement method**

After obtaining the air velocity measurements in a cross section of the extended rectangular duct, the air flow rate was calculated following ASHRAE guidelines. In total, 25 measurement points in a cross section area were used to take air velocity measurements using a hand held hot wire anemometer. The range of the anemometer was set to 15 m/s (3000 FPM) and the time constant was set to 7 seconds in order to average out the turbulence. The damper flow testing was conducted for 3 RPMs (200, 600, 1200) and 3 damper opening (10, 20, 50) combinations.
2.3 Data analysis

2.3.1. Phase 1 testing – Comparison of the air flow rates among six chambers

ANOVA was used to study the variances in the flow rates among chambers. Assuming null hypothesis that all chambers had the same flow rate, a multiple comparison method, ‘Tukey’s Honest Significant Difference’ method (α=0.05) was implemented to obtain pairwise comparison of the flow rates in different chambers. R package was used for the statistical analysis.

2.3.2. Phase 2 testing – the core chamber velocity analysis

The velocity component “u” following the flow direction and perpendicular to the plane of inlet was considered for analysis purposes as the other velocity components were found to be very small compared to the velocity ‘u’. The velocity at bird height is more relevant to understand the thermal comfort of the bird and hence, the bird height (0.20 m) velocity was analyzed in depth. The positions inside the core chamber were compared to assess the difference among chambers. ANOVA was performed to assess significant difference among chambers. Height 2 (0.63 m) and Height 3 (1.21 m) were analyzed with a view to understand the flow dynamics from the measured values. Tecplot 360 (Version 2013 R1, Bellevue, WA, USA), a commercial visualization package, was used to visualize the velocity at different heights. Different profile views were obtained to visualize the velocity (u) in the core chamber.

2.3.3 Damper flow rate

ANOVA was used to compare damper flow rates among chambers. Assuming null hypothesis that all chambers had the same damper flow rate, a multiple comparison method, ‘Tukey’s HSD’ method was implemented to obtain pairwise comparison of the damper flow rates in different chambers.
2.4 Results and discussion

2.4.1 Phase 1 testing

Chamber system flow rates versus pressure drop

Table 2.6 summarizes the flow rates and corresponding static pressure across system for all chambers at fan shaft speed of 400 and 600 RPM with 30% damper opening. The average chamber to chamber variation was 5.06% at an RPM of 600. The highest static pressure was 2.59 in. of water realized in chamber 1 and chamber 3 at an RPM of 1200 at maximum area coverage with a damper opening of 10%. The static pressure increased substantially with decrease in the damper openings for the reason that the system is a recirculation system. In the initial testing, it was discovered that the flow rates and the corresponding static pressure drops in chamber 5 and chamber 6 differed from the other chambers substantially. This was due to the fact that both the chambers had air leakage in the system flow path. Corrective action was taken immediately after the leakage problem was identified. Nevertheless, the flow rates in these chambers were 3.5% lower on an average than other chambers. A similar flow rate could be achieved by increasing the RPM such that all chambers had the same flow rate.

Table 2.6 Chamber system flow rates versus pressure drop at 30% damper opening; Q in CFM, ΔP in inches of water

<table>
<thead>
<tr>
<th>Screen coverage (%)</th>
<th>Chamber 1</th>
<th>Chamber 2</th>
<th>Chamber 3</th>
<th>Chamber 4</th>
<th>Chamber 5</th>
<th>Chamber 6</th>
</tr>
</thead>
<tbody>
<tr>
<td>ΔP</td>
<td>0</td>
<td>0.21</td>
<td>0.22</td>
<td>0.25</td>
<td>0.22</td>
<td>0.25</td>
</tr>
<tr>
<td>Q</td>
<td>2093</td>
<td>2170</td>
<td>2164</td>
<td>2115</td>
<td>2164</td>
<td>2115</td>
</tr>
<tr>
<td>ΔP</td>
<td>0.16</td>
<td>0.2</td>
<td>0.21</td>
<td>0.22</td>
<td>0.2</td>
<td>0.22</td>
</tr>
<tr>
<td>Q</td>
<td>2028</td>
<td>2126</td>
<td>2112</td>
<td>2000</td>
<td>2002</td>
<td>1950</td>
</tr>
<tr>
<td>ΔP</td>
<td>0.23</td>
<td>0.24</td>
<td>0.26</td>
<td>0.25</td>
<td>0.22</td>
<td>0.22</td>
</tr>
<tr>
<td>Q</td>
<td>2213</td>
<td>2147</td>
<td>2155</td>
<td>2099</td>
<td>2002</td>
<td>1950</td>
</tr>
<tr>
<td>ΔP</td>
<td>0.2</td>
<td>0.21</td>
<td>0.22</td>
<td>0.25</td>
<td>0.22</td>
<td>0.22</td>
</tr>
<tr>
<td>Q</td>
<td>2191</td>
<td>2150</td>
<td>2002</td>
<td>2173</td>
<td>1950</td>
<td>1996</td>
</tr>
<tr>
<td>ΔP</td>
<td>0.18</td>
<td>0.19</td>
<td>0.22</td>
<td>0.22</td>
<td>0.22</td>
<td>0.22</td>
</tr>
<tr>
<td>Q</td>
<td>2037</td>
<td>2086</td>
<td>2008</td>
<td>1950</td>
<td>1996</td>
<td></td>
</tr>
<tr>
<td>ΔP</td>
<td>0.19</td>
<td>0.22</td>
<td>0.21</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Q</td>
<td>2060</td>
<td>2067</td>
<td>2045</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
### Table 2.6 Continued.

<table>
<thead>
<tr>
<th>Screen coverage (%)</th>
<th>Chamber 1</th>
<th>Chamber 2</th>
<th>Chamber 3</th>
<th>Chamber 4</th>
<th>Chamber 5</th>
<th>Chamber 6</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$\Delta P$</td>
<td>$Q$</td>
<td>$\Delta P$</td>
<td>$Q$</td>
<td>$\Delta P$</td>
<td>$Q$</td>
</tr>
<tr>
<td>0</td>
<td>0.53</td>
<td>3214</td>
<td>0.41</td>
<td>3251</td>
<td>0.51</td>
<td>3542</td>
</tr>
<tr>
<td>25</td>
<td>0.55</td>
<td>3361</td>
<td>0.46</td>
<td>3215</td>
<td>0.53</td>
<td>3409</td>
</tr>
<tr>
<td>50</td>
<td>0.54</td>
<td>3352</td>
<td>0.47</td>
<td>3219</td>
<td>0.53</td>
<td>3338</td>
</tr>
<tr>
<td>75</td>
<td>0.56</td>
<td>3204</td>
<td>0.5</td>
<td>3073</td>
<td>0.55</td>
<td>3299</td>
</tr>
</tbody>
</table>

Flow rates and the corresponding static pressure at other RPMs (200, 800, 1000, and 1200) and damper openings (10%, 30%, 50%, 70%) are provided in Appendix I. tables A1-1-4.

Analysis of variance (ANOVA) showed no significant difference in the mean flow rate among different chambers. The F-statistic was found to be 0.213 ($F_{\text{Critical}}=2.293$) and the corresponding $p$-value was 0.956 and thereby, failing to reject the null hypothesis of no significant difference in means. Although, the mean flow rates in among chambers did not show any statistical difference chamber 5 and chamber 6 had lower flow rates and need RPM adjustment. Table 2.7 summarizes the results of Tukey’s HSD test.
Table 2.7 Tukey’s HSD method (alpha=0.05): Pairwise comparison of flow rates in different chambers

<table>
<thead>
<tr>
<th>Chamber comparison</th>
<th>diff</th>
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<th>upr</th>
<th>p adj</th>
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<td>141.54669</td>
<td>-1462.482</td>
<td>1745.576</td>
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</table>
Flow consistency test

In order to quantify the difference in flow rates among the chambers a flow consistency test (leakage test) was carried out. The test was carried out for each chamber at lower RPMs (200, 400, 600) for a very short period of time (about 5 minutes at each RPM) such that the seals were not over pressurized. The damper was completely shut and the blower was run at different RPMs. The velocity and static pressure were measured at the exhaust duct similar to phase 1 testing. Table 2.8 summarizes the findings of the test.

Table 2.8 Flow rate difference in % at 600 RPM

<table>
<thead>
<tr>
<th>Chamber 1</th>
<th>Chamber 2</th>
<th>Chamber 3</th>
<th>Chamber 4</th>
<th>Chamber 5*</th>
<th>Chamber 6</th>
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</thead>
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</tr>
<tr>
<td>Chamber 3</td>
<td>3.68</td>
<td>6.80</td>
<td>0.00</td>
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<td>13.58</td>
</tr>
<tr>
<td>Chamber 4</td>
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<td>0.00</td>
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</tr>
<tr>
<td>Chamber 5</td>
<td>-11.45</td>
<td>-7.83</td>
<td>-15.71</td>
<td>-11.16</td>
<td>0.00</td>
</tr>
<tr>
<td>Chamber 6</td>
<td>-12.10</td>
<td>-8.46</td>
<td>-16.39</td>
<td>-11.81</td>
<td>-0.58</td>
</tr>
</tbody>
</table>

* The testing results before the leakage in Chamber 5 was identified and corrective action was taken. The differences of the flow rates between chambers 3 and 5 were down to 13.4% after the leakage problem in Chamber 5 was fixed.

Considering the fact that blowers in all chambers were identical and were run at the same RPM the chambers 1 and 3 had the least relative air leakage whereas chambers 5 and 6 had the maximum leakage. At 600 RPM the flow rate in chambers 5 and 6 was found to be 10-15% lower than that of chambers 1 and 3. However, this test does not reflect the actual flow conditions as the damper was kept close during the test. The test provided a basis for adjusting the RPM of the blowers in chambers to attain a flow rate of similar magnitudes.
2.4.2 Phase-2 testing

**Velocity at the bird height (0.20 m)**

Figures 2.6-2.8 show some examples of the air velocity at bird height (0.2m) inside the core chamber at different positions obtained at different RPMs. All the chambers were found to have similar velocities variations. Chamber 5 and 6 had lower velocities as a result of lower overall flow rates. This is most probably due to some minor air leakage along the flow path especially in areas where the chamber components are joined and pressed against each other. The figures (2.6-2.8) indicated that the flow was slightly none uniform as positions at the same distance from the inlet had different velocities. This may be partially due to the placement of flow inlet duct entering the conditioning chamber in the bottom part of the chamber more towards the left of the center line of the chamber. However, the difference was non-significant when the accuracy of the measurement was taken into consideration. The positions near the inlet had velocities ranging from 0.8 m/s to 2 m/s at 600 to 1200 RPMs, respectively.

The positions in the rear end of the chamber had velocities significantly ($p=0.002$ at 600 RPM 30 Hz motor frequency) lower than the velocities at positions near the inlet. This is certainly true because of the fact that at bird height the feeders modify the flow in the flow direction rendering significantly lower velocities at positions behind the feeders. Certain discrepancies in velocity measurements were observed due to the reason that the feeders were placed on the bedding material which is fairly uneven and the feeders were not firmly fixed but manually placed during the measurements. It was found from the computational simulations discussed in subsequent chapter that even a slight change in orientation and displacement of the feeders alter the flow and hence, this flow variation is reflected at the measurement positions.
Figure 2.6a Velocity vs. RPM at 30% damper opening for chamber 1

Figure 2.6b Velocity vs. RPM at 70% damper opening for chamber 1
The flow in chamber 1 was found to be consistent compared to the flow in other chambers. Figure 2.6a and figure 2.6b show the velocity variation at different RPMs for 30% and 70% damper opening respectively. A similar trend existed at different damper openings with lower velocities at larger damper opening. The ‘LF’ position had higher velocity though it was at the same distance the ‘RF’ positions. The difference between the velocities at ‘LF’ and ‘RF’ positions increased with increase in the RPM. This indicates the flow was slightly directional but not erratic. Similarly, positions ‘LB’ and ‘RB’ had a similar variation owing to higher velocities in the left section of the core chamber. Chamber 2 (shown in Appendix I) behaved similar to chamber 1 with lower velocities but not substantially different from that of chamber 1.

![Graph showing velocity vs. RPM at 30% damper opening for chamber 4](image)

**Figure 2.7a Velocity vs. RPM at 30% damper opening for chamber 4**
Similar velocity variations were observed in chamber 4 (Figure 2.7). The flow was more uniform which is reflected by the smaller difference in velocity between positions at same distance from the inlet viz. ‘LF’ and ‘RF’. However, the positions at the rear end of the chamber had significant difference in their velocities and the difference increased with increase in RPM. ‘LB’ position had velocities higher than the ‘RB’ position which was at the same distance from the inlet. Both rear positions had lower velocities compared to other positions owing to the presence of the feeders in the line of flow. Chamber 3 and 4 behaved similarly in case of velocity at bird height (Appendix I). Chamber 3 had slightly higher velocities compared to chamber 4 probably due to lesser air leak in the flow system which is quite apparent assuming all chambers are identical.
Figure 2.8a Velocity vs. RPM at 30% damper opening for chamber 6

Figure 2.8b Velocity vs. RPM at 70% damper opening for chamber 6
Chamber 5 and chamber 6 exhibited similar velocity variations at all the positions inside the core chamber. Both chambers had lower overall flow rates owing to some air leak in the flow system. This was reflected in the pressure drop measurements in phase 1 tests. However, the flow was more uniform compared to other chambers with the positions near the inlet and the center having the same velocity at a particular RPM. Consequently, low velocities were measured at positions away from the inlet as shown in figure 2.8a and figure 2.8b (chamber 6).

The velocity vs. RPM plots for other chambers and at other damper opening settings are provided in Appendix I (Figures A1-1 to 15).

**Velocity comparison among chambers**

The velocities at various positions inside the core chamber were similar among all chambers except chamber 5 and chamber 6 which had lower velocities as shown in the figure 2.9. No statistically significant difference existed in the velocity at different positions considered individually. The deviation in velocity at the center of the core chamber (position ‘C’) in case of chamber 3 (figure 2.9b) may be due to the uneven surface of the bedding material which certainly has a significant impact on the air speed at bird height near the floor.
Figure 2.9a Velocity vs. RPM at 30% damper opening at position ‘LF’

Figure 2.9b Velocity vs. RPM at 30% damper opening at position ‘C’
Velocities at height 2 and height 3

Height 2 (0.63 m) and Height 3 (1.21 m) provided a means to analyze the flow patterns using field measurements. The vertical stratification of the velocity gradient was evident from the velocities measured at all 3 heights at 5 different positions in the core chamber. Velocities at Height 3 included negative values especially at positions near to the inlet. This is due to formation of vortices due to sudden expansion of duct area into the core chamber. This resulted in very low mean velocity values at height 3. Consequently, the velocities away from the inlet (rear end of the chamber and the center position ‘C’) had velocities greater than that at positions near the inlet. Height 2 had higher velocity on an average across the whole chamber for the fact that the flow had less obstructions and height 1 (0.2 m) and height 3 (1.21 m) had lower velocities due to surface roughness and vortices respectively.
However, in many observations near the inlet the velocity at bird height (0.2 m) was higher than at height 2 (0.63 m). This was probably due to the fact that the upper layer of air moved faster in upward direction than the lower layers causing lesser horizontal momentum at height 2. Hence, the observations do not agree with the assumption of uniform velocity across the inlet. The ‘z’ velocity component at positions near the inlet was as high as 0.19 m/s (at 1200RPM) which is relatively significant when compared to the velocity in ‘x’ direction (direction of flow).

All the 3 heights were used as an input to visualize the velocity variation using the visualization tool of Tecplot. Figures 2.10 – 2.12 shows some examples of the 3-D contour plots for different chambers under given RPM and damper opening settings. Each figure has 3 different profile views displayed in 3 different windows. The top window shows a general front view. The bottom windows illustrate the side view and top view, respectively. In general, +x direction spans the width, +y direction spans the length and +z direction represents the altitude. The square boxes in the top window represent the measured locations.
Figure 2.10 Velocity profiles in core chamber of chamber 1 at 600 RPM (30 Hz) and 30% damper opening.
The above figure clearly shows the variation of the velocity in the vertical direction. At height 3 the velocity was very low at positions near the inlet. This is due to the jet effect of the flow at the inlet. The flow variation across the chamber can be seen from the side profiles in the above figure. The velocity was greater at the inlet and slowly fades with increase in distance. The effect of feeders is evident from the side view shown in the figure. The velocity variation in the side view shows low velocity in the rear end of the chamber.

Figure 2.11 Velocity profiles in core chamber of chamber 2 at 600 RPM (30 Hz) and 30% damper opening
Chamber 2 had a more uniform flow compared to chamber 1. The velocity contour shows a more uniform distribution in the top view shown in the figure 2.11 compared to that of chamber 1 (figure 2.10). However, the average velocity was relatively low compared to chamber 1 and chamber 3 (figure 2.7 and figure 2.9).

Figure 2.12 Velocity profiles in core chamber of chamber 3 at 600 RPM (30 Hz) and 30% damper opening
Similar to chamber 1, chamber 3 has higher velocity in the left section of the core chamber at bird height as depicted by the top view of the above velocity contour.

2.4.3 Damper flow—Fresh air intake

Table 2.9 lists the measured damper flow rates for different chambers at different operating configurations. It was observed that greater variation in damper flow rates occurred at higher RPMs and larger damper openings. This is certain because with higher air speeds the turbulence increases and the uncertainty in the measured velocity increases consequently. Figure 2.13 shows the variation in damper flow rates among chambers. However, Analysis of variance (ANOVA) showed no significant difference in the mean damper flow among different chambers. The F-statistic was found to be 0.005 and the corresponding p-value was found to be close to 1 and thereby, failing to reject the null hypothesis of no significant difference in means.

Table 2.9 Summary of airflow rates through dampers (CFM) in different chambers

<table>
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<td>1200</td>
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<td>1879</td>
<td>2045</td>
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</table>
Table 2.10 shows the pairwise comparison obtained using Tukey’s HSD method with an alpha value of 0.05. The $p$-values in all the cases were found to be equal to one indicating no significant difference in damper flow rates.
Table 2.10 Tukey’s HSD method (alpha=0.05): Pairwise comparison using of damper flow rates in different chambers

<table>
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<th>Chamber Comparison</th>
<th>diff</th>
<th>lwr</th>
<th>upr</th>
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2.5 Conclusion and further studies

Flow testing of the poultry engineering chambers was carried out. The flow rate in the overall system in each chamber was assessed. Also, the velocity profiles in the animal occupied region was measured with a view to visualize the velocity variation across the core chamber and establish a field dataset to validate the results obtained using CFD which is further discussed in detail in chapter 3.

The observations of the flow testing can be summarized as follows:

1. Phase-1 testing was aimed at analyzing the flow rates and the corresponding pressure drops across the system as a consequence of the system geometry and components. The test was carried out at 6 different RPMs and 4 damper opening combinations in each of the 6 chambers. The average difference in flow rates among chambers was 5.06% at 600 RPM. Chamber 1 and chamber 3 had higher flow rates. Chambers 5 and 6 had lower flow rates even after fixing the leakage. Both the chambers 5-6 had approximately the same flow rates and hence, an RPM correction could be applied to obtain uniform flow rates in all chambers. A flow consistency test was performed to quantify the absolute flow rate difference between chambers 5-6 and other chambers. This difference can be used to apply an RPM correction at any operating RPM.

2. Phase-2 test aimed at visualizing the variation in velocity inside the core chamber where the test species are housed. The test results mainly served two purposes viz. to develop velocity contours such that the variation of velocity variation across the core chamber could be visualized and the CFD results could be validated. The air flow was found to be higher at bird height (0.20 m) at positions near the inlet probably due to the geometry of the turnaround structure just before the inlet. Positions away from the inlet (‘LB’, ‘RB’) had significantly lower velocities due to the presence of feeders in the line of flow. However, the velocity at height 2 (0.63 m) at positions away from the inlet had velocities comparable to those near the inlets due to lesser flow disturbance. Height 3 (1.21 m) had higher velocities at positions away from the inlet due to the jet effect and expansion of the flow jet at the inlet.
Chamber 1 and Chamber 3 showed higher velocity in the left section of the core chambers partially due to the geometrical placement of the inlet duct entering the conditioning chamber in the lower segment of the chamber system. However, the deviation in flow between left and right sections was prominent only at higher blower RPMs. Other chambers had fairly more uniform flow across the core chambers.

3. The damper flow rate measurements provided a means of assessing the blower performance (speed, slippage etc at a particular RPM) as the flow is only dependent on the damper opening and not on the geometry of the whole system. As a result, the blowers in all the chambers were found to perform consistently. There was no significant deviation in the damper flow rates among the chambers indicating similar damper structure and blower performance. The difference between the damper flow rates increased with increase in RPM. Higher velocities result in greater turbulence and hence less accurate flow measurements.

4. The overall structure of the Poultry Chamber with few corrective measures is suitable for conducting poultry research and air quality studies. RPM corrections applied to chambers with lower flow rates or air leakage would result in similar flow conditions and hence similar indoor environment. With such a system in place, a controlled environment can be established to conduct reliable research studies.

5. The performance of the chamber complex can be further improved by automating the damper opening using linear actuators which will eliminate the errors in adjusting the damper manually. Also, increasing the number of collimating screens before the core chamber inlet can improve uniformity in flow. Despite pressure drop and dust accumulation, increasing the number of screens improve the flow structure in the system.
CHAPTER 3 PERFORMANCE EVALUATION OF THE POULTRY ENGINEERING CHAMBERS: CFD MODELING

3.1 Introduction

3.1.1 Application of Computational Fluid Dynamics (CFD) in agriculture

Over the years, the development of modern computational techniques like Computational Fluid Dynamics (CFD) coupled with increased computing power as a result of the advancement in hardware technologies have made these methods more versatility, accurate and easy to use (Chen and Srebric, 2002). The availability of commercial CFD packages has made it possible to apply these methods to almost any field of fluid dynamics study. Numerical methods pave way to test and validate ideas and designs before being implemented or fabricated. This gives a tremendous advantage over experimental designs and testing making it highly economical and a great aid for decision making. Currently, simulating industrial processes in order to assure process feasibility and process safety has become necessary even at small scales. The ability of these computational methods like CFD to virtually simulate any property or parameter has made it widely applicable in the areas including aerospace and materials engineering, bio-chemistry, plasma physics, biosystems and agriculture, energy efficiency in buildings and Heating, Ventilation and Air Conditioning (HVAC). Computational Fluid Dynamics (CFD) is being used extensively in intensive agro-based production systems like green-houses, livestock production, meat processing and animal housing units. In a recent comprehensive review, Chen (2009) found that detailed CFD-based studies represented about 70% of the modeling investigations of room ventilation in the past 7 years. CFD has been used successfully in modeling air flows in building (Chen and Srebric, 2002).

Irrespective of the type of building or indoor environment, CFD can be conveniently used for simulating the microclimate. In agricultural industry, production depends on numerous factors. Not only factors like climate and weather, which are external, influence the production but also the local conditions of the surroundings play a very important role in
quality and quantity of the production, especially when the production is indoors. Many studies have showed that the ability to maintain desired environmental conditions is highly dependent on the design and performance of ventilation conditions (Norton et al., 2007). This is invariably true because the ventilation which represents the air exchange is the driving factor for the temperature, humidity, and air quality and contaminant concentration in the local surroundings. In ventilated structures the exchange processes are mainly dominated by convective heat and mass transfers (Roy et al., 2002). The air flow patterns form the basis for the link between the external environment and the indoor climate and thus a complete understanding of the air flow is necessary to visualize the conditions required for the specific application (Norton et al., 2007). It is very important to understand the air distribution for purposes ranging from identifying hot spots and heat distribution to spread of airborne diseases and energy conservation. All these contribute towards establishing sustainable and safe production system.

Numerous studies have stated the importance of uniform air distribution and the importance of air exchange rate (Spoodler et al., 2000; Pawar et al., 2007; Bustamante et al., 2013). Kavanagh (2003) reported that the knowledge of principles governing the distribution of the indoor climatic variables is necessary when designing production systems or optimizing their performance. In case of crop production, the prevailing environment in the greenhouse needs to be in the temperature range suitable to normal plant growth in a particular season. According to Boulard and Wang (2002) quantitative understanding of the micro climate can help growers optimize crop growth by reducing over-consumption of water and the loss of nutrients to environment. Bot (2001) concluded that the distribution of microclimate variables inside the greenhouse not only causes non-uniform production but also generates problems with pests and diseases. In animal houses, control over the air flow is needed to maintain the optimum temperature and humidity remove gaseous pollutants and maintain the percentage of fresh air.

Greater control leads to a healthier and more productive environment for animals as well as for the people working in animal houses (Spoodler et al, 2000).
The requirement of uniformity in temperature and other variables of the microenvironment is more serious in case of livestock housing systems for the reason that animals themselves contribute substantially to the heat inside the housing systems and also, younger animals are more sensitive to the environment around them. Thermoregulation is an important mechanism by which animals are able to balance heat production with heat loss while maintaining a constant body temperature when interacting with external environment (Hahn, 1999). Hence, energy used in maintaining a thermal balance with local environment determines the amount of energy that is made available for growth (Norton et al., 2010). Lago et al (2006) found that in naturally ventilated houses with low control over thermal environment leads to increase in respiratory diseases in young animals.

As compared to natural ventilation, mechanical ventilation proves to be more effective in maintaining the thermal comfort of livestock animals (Bustamante et al., 2013). The extensive use of mechanical ventilation in pig houses, poultry barns, and cattle houses has helped increase the production rate by minimizing mortality rate caused by heat stress. In case of poultry industry, in the U.S., the production income was over $40 billion (USDA, 2014). According to the statistics by USDA, the demand for broiler meat is increasing substantially year by year. This has resulted in making large investments in technology to enhance the broiler production. Consequently, more number of poultry houses are being established and existing houses are pressurized to produce more leading to more broilers per house. However, this affects the environment for the broilers especially in summer seasons in hot and humid parts of the country. Numerous studies have been conducted to study the responses of the broilers to different kinds of environment and environmental variables (Lott et al., 1998; May et al., 2000; Simmons et al., 2003; Yahav et al., 2001). Huge investments have been made to establish sensor systems to study the microenvironment inside poultry houses (Bustamante et al., 2012).

Blanes-Vidal et al., (2008) and Norton et al., (2007, 2010) among others have clearly stated the limitations of experimental methods and have vouched for using numerical methods like CFD as a reliable means of studying in depth the prevalent microenvironment.
However, it is extremely important and necessary to validate the models used with experimental data before considering it as a tool to make decisions. Blanes-Vidal et al., (2008) used a multi-sensor system to quantify the temperature, pressure and air velocity in broiler houses and used these experimental results to validate CFD models. Many other studies have been carried out to elucidate the use of CFD in modeling ventilation systems, air flow distribution, temperature variation, distribution of pollutants like NH₃ (Pawar et al., 2010; Bjerg et al., 2013).

As explained in Chapter 1 and 2 there are some serious limitations of conducting field studies. The alternative commonly adopted to address problems with the microenvironment pertaining to poultry is by conducting studies on poultry in laboratories or research chambers. As discussed in chapter 2 dedicated research chambers have been used to study closely the physiological responses of the broilers to microenvironment in various scenarios. However, as stated in earlier chapters the results obtained using most of the existing research chambers may be less reliable due to inherent limitations of the existing research chambers and hence, the poultry engineering chamber at NCSU was designed and built to foster animal welfare and air quality research.

3.1.2 Rationale for using CFD

This study attempts to incorporate the advantages of both field studies using research chambers and CFD and strives to eliminate the limitations of these methods. As briefly discussed in chapter 1, the study evaluates a dedicated set of chambers using experimental testing as well as CFD modeling. The application of CFD to the region of interest, which is the animal occupied region not only helps comprehend the animal behavior to microenvironment but also takes into account other minute details like flow near the bedding, flow at bird height, effect of flocking etc. similar to a commercial farm. Using CFD eliminates the use of a large number of sensors and hence makes it more economically feasible and avoids unnecessary obstruction to the flow or bird movement. The mesh refinement and advanced discretization techniques associated with the CFD algorithm provides a means to improve the resolution. Through this approach of using dedicated test
chambers coupled with CFD modeling, it is possible to attain and study conditions closest to the actual conditions and hence, study what the animals actually feel.

This approach provides a superior way to analyze the physiological responses of the poultry to the microenvironment compared to mere field studies and lab scale studies.

This study addresses the following:
1. Flow dynamics and velocity trajectories in the animal occupied space to assess the effectiveness of the overall chamber at different flow rates.
2. Velocity profiles at different altitudes and longitudinal transition of flow which provides a clear distribution of velocities around animal occupied zone
3. Effect of animals on the flow dynamics including effects on velocity distribution, heat production and overall thermal environment.

The CFD simulations developed in this study can be used to improve the thermal environment or can be used in isolation to test different operating configurations and its effects on the flow.

### 3.1.3 Theory of CFD

The basic idea behind all CFD techniques is the resolution of a set of partial differential equations where the equations correspond to mass or continuity and momentum or Navier-Stoke’s law and energy (Patankar, 1980). For an incompressible fluid under isothermal conditions the equations of fluid motion are given as Eq (1), Eq(2) and Eq(3) (FloEFD, 2013)

\[
\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u_i)}{\partial x_i} = 0 \tag{1}
\]

\[
\frac{\partial (\rho u_i)}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_j} = - \frac{\partial p}{\partial x_i} + \frac{\partial (\tau_{ij})}{\partial x_j} + \rho g_i + F_i \tag{2}
\]

\[
\frac{\partial (\rho c_T)}{\partial t} + \frac{\partial (\rho u_j c_T)}{\partial x_j} = - \frac{\partial (k \frac{\partial T}{\partial x_i})}{\partial x_i} + S_T \tag{3}
\]

where \( \rho \): fluid density (kg m\(^{-3}\)); \( t \): time (s); \( x, x_i, x_j \): length components (m); \( u_i, u_j \): velocity component (m s\(^{-1}\)); \( p \): pressure (Pa); \( \tau_{ij} \): stress tensor (Pa); \( g_i \): gravitational acceleration
Turbulent flows are predicted using turbulence models in which the instantaneous governing equations are averaged over a time period greater than of the turbulent motion. There are a number turbulent models used in the industry as well as in research depending on the required accuracy, computational time and geometry. Standard K-epsilon, renormalization group k-epsilon (RNG), Reynolds stress closure model (RSM). Reynolds stress closure models have proved to be more beneficial in modeling flows in confined rooms (Schalin and Nielson, 2004). Advanced techniques like Large Eddy Simulation (LES) where large turbulent eddies are considered highly anisotropic on both mean velocity gradients and geometry of the flow domain (Norton et al., 2007). LES methods require large computational power and time. Reynolds averaged Navier-Stoke’e Equations (RANS) adopts time averaging mechanism to determine the effect of turbulent on the mean flow field. However, the standard k-epsilon model described by Launder and Spalding, (1974) is the most commonly used turbulence model in modeling agricultural buildings (Barzanas et al., 2004; Norton et al., 2008). In this study, the standard k-epsilon model was used and the rationale behind using this model was that it has been extensively used in modelling poultry farms with mechanical and natural ventilation (Blanes et al, 2008, Lee et al., 2007; Pawar et al., 2007). The reason behind the model being widely used is its convergence behavior and reasonable accuracy and also, studies carried out using more complex models do not differ substantially (Gebremedhin and Wu, 2003). The equations of turbulence transport for standard k-epsilon model are given by Eq (4) and Eq (5)

\[
\frac{\partial \rho k}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i k) = \frac{\partial}{\partial x_i} \left[ \left( \frac{\mu + \mu_k}{\sigma_k} \frac{\partial k}{\partial x_i} \right) \right] + S_k
\]

\[
\frac{\partial \rho \varepsilon}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i \varepsilon) = \frac{\partial}{\partial x_i} \left[ \left( \frac{\mu + \mu_k}{\sigma_k} \frac{\partial \varepsilon}{\partial x_i} \right) \right] + S_\varepsilon
\]
Where \( k \): turbulent kinetic energy (m\(^2\) s\(^{-2}\)); \( \mu \): fluid viscosity (m\(^2\) s); \( \mu_t \): turbulent viscosity (m\(^2\) s); \( \sigma_k \): turbulent Prandtl number for \( k \); \( S_k \): the generation of kinetic energy; \( \varepsilon \): turbulent dissipation rate (m\(^2\) s\(^{-3}\))

### 3.2 Methodology

#### 3.2.1 Flow domain

The core chamber of the specially designed chamber systems was considered as the flow domain. As each chamber was identical only chamber 1 was used for simulating purposes owing to more consistent flow rates through the chamber. The geometry of the flow domain was developed using a commercial package, Creo Parametric (version 2.0, PTC Inc., Needham, MA, USA). The accessories or components such as the water line and feeders inside the core chamber were also considered in the geometry. Figure 3.1 shows the 3-D drawing of the core chamber.

A three-dimensional simulation of the indoor airflow was carried out by the commercial code FloEFD for Creo (Version 13, Mentor Graphics, Wilsonville, Oregon, USA). The inlet and the outlet were modeled as rectangular openings as shown in figure 3.1. Specific properties such as roughness and heat transfer coefficients were assigned to bedding material, drinkers and feeders (table 3.2). The core chamber was considered alone when running the simulations. The rationale behind simulating the flow in core chamber alone is that the core chamber is the region where broilers are housed and hence the intent of the study is to characterize the animal surroundings. Also, this approach does not require modeling of air leakages either by defining porosity or by correcting the air flowrate which in turn helps to simulate closely the actual scenario in the core chamber.
As the commercial CFD package, FloEFD for Creo (Version 13, Mentor Graphics, Wilsonville, Oregon, USA) is very user friendly and has many features applicable to different fields of study. FloEFD has been used earlier in various research studies (Remsburg et al., 2010; Kawamorita et al., 2012; Wang et al., 2009). It has a set of predefined database for different materials and commonly used flow components like fans. The engineering database has the properties of commonly used materials as default input files and also, the package allows the user to add custom materials and specify properties and save it as a module for future use.
3.2.2 Meshing

FloEFD has adaptive meshing capabilities. The design developed in Creo Parametric was directly imported into the FloEFD interface and the mesh was developed. The application used structured rectangular meshes. The application also uses partial meshes when required, based on the geometry. The procedure for meshing involves splitting each of the basic mesh cells intersecting with the solid/fluid interface uniformly into 8 child cells. Then, each of the child cells intersecting with the interface is in turn split into 8 cells of next level, and so on, until the user specified cell size is attained (FloEFD, 2013). The package provides 8 levels (1-8) of mesh refinement, although, level 3 is basically recommended for any specific engineering application. Level 5 provided a good accuracy and accounted for a reasonable computational time and hence was adopted to perform the rest of the analysis.

3.2.3 Numerical simulation

Simulations were carried out under steady-state conditions. Analysis was carried out assuming the flow to be steady. The application uses several parameters as constraints to reach convergence. The ‘maximum travels’ criteria determine the calculation period required for a flow disturbance to travel across the entire flow domain. This, by default, was set to a value of 4. Number of iterations could be specified and for this study the number of iterations were set to automatic and hence, the numbers depended on the other convergence criteria. Usually, the numbers of iterations were around 90-110 in case of steady-state flow. Convergence goals could be set before solving the flow problem to specify a particular goal as a criterion for convergence. Average static pressure at inlet and velocity at outlet were specified as surface goals. The advantage of specifying one or more goals was that the application allows monitoring the goal parameters during the simulation and considers the goal parameters for convergence. This generally aids faster convergence rates and complete convergence of the variables of importance. Further, the study simulated three different cases in order to provide a comprehensive picture of the effectiveness of the chamber for animal welfare and air quality studies.
In each case, the flow was simulated for a set of operating configurations which are most commonly used and hence, tends to be representative of the real world scenario. The cases with their respective set of simulations are listed in Table 3.1.

Table 3.1 Simulation of different cases

<table>
<thead>
<tr>
<th>Case</th>
<th>Specifications</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>General flow</td>
<td>Represents the general flow dynamics as a consequence of the chamber geometry</td>
</tr>
<tr>
<td></td>
<td>dynamics</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Velocity</td>
<td>Specific boundary conditions measured at inlet and outlet</td>
</tr>
<tr>
<td></td>
<td>contours</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Simulated Bird</td>
<td>Specific boundary conditions measured at inlet and outlet; heat flux from the</td>
</tr>
<tr>
<td></td>
<td></td>
<td>bird; surface characteristics of the bird</td>
</tr>
</tbody>
</table>

3.2.4 Boundary conditions

Basically, two sets of boundary conditions were used in the study. The first set consisted of boundary conditions which were assumed to be appropriate (or used commonly in literature) like the wall roughness, roughness of the bedding material, heat conduction in solids (drinkers and feeders), air properties including inlet temperature and pressure and density. An average humidity of 50% was considered. The type 1 boundary conditions are listed in Table 3.2.
<table>
<thead>
<tr>
<th>Computational Settings</th>
</tr>
</thead>
<tbody>
<tr>
<td>3D double precision</td>
</tr>
<tr>
<td>Steady-State</td>
</tr>
<tr>
<td>Turbulence model: Standard k-epsilon</td>
</tr>
<tr>
<td>Spatial approximations: Cell-centered finite volume (FV); Spatial discretization: Implicit second order Upwind based on modified QUICK approximations and the TVD method</td>
</tr>
<tr>
<td>Pressure-velocity coupling : SIMPLE algorithm with operator splitting technique</td>
</tr>
<tr>
<td>Numerical algorithm: Pressure-correction</td>
</tr>
<tr>
<td>Numerical Solution Technique: Preconditioned generalized conjugate gradient method</td>
</tr>
<tr>
<td>Numerical preconditioning: Incomplete LU factorization</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Constant Type 1 Boundary Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air properties: Density: 1.225 kg m^{-3}; C\textsubscript{p}: 1006.43 J kg\textsuperscript{-1} K\textsuperscript{-1}; Thermal Conductivity:0.0242 Wm\textsuperscript{-1}K\textsuperscript{-1}; Viscosity:1.789 E-5 kg m\textsuperscript{-1}s\textsuperscript{-1}</td>
</tr>
<tr>
<td>Component Properties - Feeder : Steel , Thermal conductivity - 50.2 W/m K</td>
</tr>
<tr>
<td>Wall boundary conditions: Adiabatic walls with No-slip condition; Bedding: Roughness- 0.03 m</td>
</tr>
<tr>
<td>Atmospheric pressure : 101325 Pa. ; Gravitational Constant : 9.81 ms\textsuperscript{-1}</td>
</tr>
</tbody>
</table>
The second group of boundary conditions was comprised of the inlet and outlet parameters, which were the ones directly, involved in the fluid flow calculations. These boundary conditions were considered after taking into account the practical difficulties such as time required to measure the parameters and the accuracy of those measurements.

Two sets of inlet and outlet boundary conditions were considered. The first set being ‘volume inlet’ and ‘static pressure outlet’ conditions. The inlet flow volume was considered to be equal to the measured flow rate at the exhaust duct of the chamber system. Therefore, this boundary condition assumes uniform velocity in the direction of flow across the inlet. The second set of boundary conditions were ‘velocity inlet’ and ‘static pressure outlet’ conditions. This set of boundary condition was a function of the measured velocity components across the inlet. For this boundary condition the velocity at the inlet in all three directions were measured at multiple points (44 measuring points). The velocities were measured using a sonic anemometer placed just in front of the inlets. Velocities were measured at intervals of 0.2 m (8 in.) across the inlet and at 4 different heights (0.15 m, 0.38 m, 0.48 m, 0.76 m). However, the value of the boundary conditions varied based on different cases and their simulations as mentioned in Table 3.1. The summary of inlet and outlet boundary conditions for different cases is listed in Table 3.3.
Table 3.3 Boundary conditions for different cases

<table>
<thead>
<tr>
<th>Case(^{[a][b]})</th>
<th>Boundary Conditions</th>
<th>System Operating Parameters</th>
<th>Description</th>
</tr>
</thead>
</table>
| 1                | Inlet- Volume flow rate  
Outlet- Static pressure | 600 RPM (30 Hz)  
30% Damper Opening | Volume flow rate measured at exhaust duct; Static pressure measured at outlet |
| 2                | Inlet- Volume flow rate  
Outlet- Static pressure  
BC-1 | 600 RPM (30 Hz),  
1200 RPM (60 Hz)  
30% Damper Opening | Volume flow rate measured at exhaust duct; Static pressure measured at outlet |
| 3                | Inlet- Volume flow rate  
Outlet- Static pressure  
BC-2 | 600 RPM (30 Hz),  
1200 RPM (60 Hz)  
30% Damper Opening | Velocity measured at inlet (44 monitoring points); Static pressure measured at outlet |

\(^{[a]}\)Case 1 deals with assessing the general flow dynamics as a consequence of the chamber geometry and hence, only BC-1 boundary conditions were considered; case 2 and case 3 considered BC-1 and BC-2 boundary conditions.

\(^{[b]}\) Refer table 3.2 for specific description of each case

Case 3 aimed at visualizing qualitatively the effect of birds on the flow regime and the effect of flocking. In case 3, the birds were modeled as solid spheres with specific heat flux and surface characteristics. Simulations were carried out with eleven birds modeled to analyze the effect of location and distribution of air velocity.
Another simulation was carried out with thirty birds representing the general numbers of birds during testing in the chamber. Although the chamber was designed with capacity of housing about 60 birds, the simulations considered less numbers of birds as the main goal was to understand the qualitative effects of flocking and the presence of birds themselves as a factor affecting the microclimate. Each of the birds was modeled as a solid sphere of 0.21 m (8.5 inches) diameter such that the surface area is 0.15 m$^2$. The surface roughness was assumed to be 3000 micrometer. No detailed literature was found regarding the surface roughness of broilers at different age. Heat production was calculated using the empirical equation (Hiensohn and Cimbala, 2003) with an assumed bird mass of 2kg. Finally, heat flux (W/m$^2$) was calculated and input in the model. Case 3 tends to give a qualitative understanding of the effect of birds. Hence, the simulated birds roughly represented a physical bird. No data was available to validate the model. Therefore, it only provided a means to visualize how flocking affects the surrounding microclimate.

3.2.5 CFD validation

Validation is utmost important when dealing with computational models to make sure that the model results conform to the measured values (Chen and Srebric, 2002). A multi-way ANOVA was carried out to compare the computational results with the measured values. The data set included values at 5 different positions and 3 for each position. Two different RPMs of the blower rotary speeds were simulated to compare the results. An error analysis in terms of percentage of bulk velocity $E_b$ was carried out to determine the variation in computational results at each data point using Eq (6)

$$ E_b = \frac{V_{cfd} - V_{measured}}{V_b} $$

(6)

Where $V_{cfd}$ is the value is predicted by CFD; $V_{measured}$ is the measured value; $V_b$ is the bulk jet velocity (initial average inlet velocity). Further, normalized means square error as presented by Saraz et al. (2010) was used to examine the validity of the model.
An NMSE (eq. 7 and eq. 8) value less than 0.25 indicate that the model is in good agreement (Saraz et al., 2010).

\[
NMSE = \frac{(c_p - c_o)^2}{c_{pm} c_{om}} \quad (7)
\]

\[
(c_p - c_o)^2 = \sum \frac{(c_{pi}-c_{oi})^2}{n} \quad (8)
\]

3.3 Results and discussion

3.3.1 Field measurements

The mean and standard deviation of air velocity at different positions and heights in the core chamber at 600 and 1200 RPMs are listed in Tables 3.4 & 3.5. The maximum air velocity was measured at 0.63 m. This is highly likely because the plane almost coincided with the velocity centerline of the inlet velocity jet. The mean velocities at 0.65 m were 0.89 ms\(^{-1}\) and 1.92 ms\(^{-1}\) at 600 and 1200 RPM, respectively. Regarding the velocity at spatial locations, the velocity at positions LB and RB were relatively low. This is due the presence of feeders, which obstruct the flow. At bird height the velocity was greatest at locations close to inlet (LF and RF) and relatively high in the center of the core chamber. The positions closer to the outlet had 30-60% lower air velocity compared to positions near the inlet due to the presence of feeders.

In general, the velocity measured at inlet (44 monitoring points) indicated the presence of velocity components in the plane of inlet (y-axis) and also in the downward direction (z-axis). This indicated a flow less uniform than expected. These measured velocities were later used in case 2 CFD simulations.
Table 3.4 Measured air velocity at different positions and heights

<table>
<thead>
<tr>
<th>Position</th>
<th>Air velocity (m/s)</th>
<th>600 RPM</th>
<th>1200 RPM</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.20 m (Bird height)</td>
<td>0.63 m</td>
<td>0.20 m (Bird height)</td>
</tr>
<tr>
<td>LB</td>
<td>0.736</td>
<td>0.798</td>
<td>1.537</td>
</tr>
<tr>
<td>LF</td>
<td>0.991</td>
<td>0.872</td>
<td>2.060</td>
</tr>
<tr>
<td>RB</td>
<td>0.568</td>
<td>0.960</td>
<td>1.280</td>
</tr>
<tr>
<td>RF</td>
<td>0.849</td>
<td>0.974</td>
<td>1.888</td>
</tr>
<tr>
<td>C</td>
<td>0.828</td>
<td>0.868</td>
<td>1.768</td>
</tr>
</tbody>
</table>

0.794±0.156 (mean±SD)
0.894±0.073 (mean±SD)
1.706±0.305 (mean±SD)
1.924±0.093 (mean±SD)

[a] Refer to figure 2.2 for description of sampling locations

Table 3.5 Measured air velocity at inlet for BC-2 boundary conditions

<table>
<thead>
<tr>
<th>RPM</th>
<th>Velocity at inlet (m/s)[b]</th>
<th>Differential pressure at outlet (pa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>600</td>
<td>( V_x = 0.98 ), ( V_y = -0.05 ), ( V_z = -0.06 )</td>
<td>4.98</td>
</tr>
<tr>
<td>1200</td>
<td>( V_x = 1.97 ), ( V_y = -0.07 ), ( V_z = -0.09 )</td>
<td>9.71</td>
</tr>
</tbody>
</table>

[b] x-Direction perpendicular to inlet; y- in the plane of inlet (\( ^{-} \) for east to west); z= plane perpendicular to the floor (\( ^{+} \) for updraft)

3.3.2 Simulated results

Case 1 represented the general flow characteristics with average inlet velocity perpendicular to the plane of flow. The inlet velocity was consequence of the volume rate inlet boundary condition. Figures 3.2 – 3.5 show the cut plots at different planes corresponding to the experimental points and flow trajectories of the velocity field. The air entered through the inlet followed by a backward step reducing the velocity near the floor. The flow field resembled an inlet jet without significant vortices in the bird zone. The velocity decreased as it traveled through the chamber. At bird height, the feeder affected the flow field, significantly reducing the velocity by almost 50% near the outlet.
Higher velocities were observed in the center of the chamber and near the chamber walls. Due to flow expansion large vortices were observed above the inlet height (height 3). The steady state assumption and the imposed turbulence (k-epsilon model) does not represent the transient flow dynamics which in turn results in deviation from measured values at regions where vortices resolve into smaller eddies. The limitation of non-transient flows was has been reported by Norton et al., (2006). The orientation of the chamber drawing was such that the air flow was indicated in + z direction (inlet to outlet) in all the simulations.

Figure 3.2 Velocity profile at bird height (0.20 m)
Figure 3.3 Velocity profile at 0.63 m

Figure 3.4 Longitudinal cross section of velocity profile (Side profile)
In case 2 the boundary condition BC-1 was found to be less accurate. This may be due to air leakage between the core chamber and the exhaust duct. Hence, BC-2 was considered to be more representative of the real conditions and therefore, was used in further analysis.

3.3.3 Measured vs. simulated results

The differences between the CFD simulated air velocity and measured velocity is shown in Table 3.6. Velocities at height were subjected to eddy formation resulting in fluctuating positive and negative values. However, velocities at height 1 and height 2 were used to validate the CFD model as the grid resolution failed to capture the formation of vortices above the inlet height. The highest absolute difference between CFD simulated air velocity and measured value was 0.23ms$^{-1}$ and observed at bird height. This was due to the presence of feeder which significantly affected the velocity flow field. Also, in the field
measurements the feeders were not rigidly fixed and hence, even a slight displacement or orientation may have a significant effect on the actual flow. On an average CFD simulation overestimated the velocities compared to measured values. However, multiway-ANOVA results indicated that there was no significant difference between the simulated and measured results. Table 3.6 summarizes the results from ANOVA.

Table 3.6 Summary of ANOVA for simulated and measured air velocity

<table>
<thead>
<tr>
<th>Source</th>
<th>DF</th>
<th>Sum of Squares</th>
<th>Mean Square</th>
<th>F Value</th>
<th>Pr &gt; F</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model vs. Experimental</td>
<td>1</td>
<td>0.003</td>
<td>0.003</td>
<td>0.092</td>
<td>0.5415</td>
</tr>
<tr>
<td>Height</td>
<td>1[^a]</td>
<td>0.404</td>
<td>0.404</td>
<td>11.607</td>
<td>0.00163</td>
</tr>
<tr>
<td>Wind Velocity (RPM)</td>
<td>1</td>
<td>8.922</td>
<td>8.922</td>
<td>256.32</td>
<td>&lt;2E-16</td>
</tr>
<tr>
<td>Error</td>
<td>36</td>
<td>1.254</td>
<td>0.035</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

[^a] only two heights (H1 & H2) were considered, therefore, df=1

There was no statistical evidence of any difference between the simulated and the measured values ($p=0.5415$). However, there was significant evidence for differences in both measured and simulated values with respect to height and blower RPM.

3.3.4 CFD validation

The validation of CFD simulated results was carried out in terms of the distribution of air velocities at different positions and heights. The error between the simulated and measured values as a percentage of the initial velocity (mean inlet velocity) $E_b$ was assessed. 18 out of 20 points had $E_b$ value less than or equal to 10% in which 10 of them had values less than or equal to 5%. The “LB” position at 600 RPM had the highest $E_b$ of 13.59%. This is certainly due to the effect of feeders which are placed in the flow field. Similarly, “RB”
positions had greater $E_b$ owing to the presence of feeders in the line of flow. The effect of feeders was most significant for points at the bird height. However, the NMSE value was relatively very low with a value of 0.007. The $E_b$ values, expressed as percentage, are summarized in Table 3.7

Table 3.7 Difference between measured and simulated air velocities (ms$^{-1}$) at each sampling point

<table>
<thead>
<tr>
<th>Position$^{[a]}$</th>
<th>Air velocities (m/s)</th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.20 m (Bird height)</td>
<td>0.63 m</td>
<td>0.20 m (Bird height)</td>
<td>0.63 m</td>
</tr>
<tr>
<td>LB</td>
<td>0.088</td>
<td>8.8</td>
<td>-0.134</td>
<td>11.76</td>
</tr>
<tr>
<td>LF</td>
<td>0.055</td>
<td>5.5</td>
<td>-0.090</td>
<td>8.49</td>
</tr>
<tr>
<td>RB</td>
<td>-0.033</td>
<td>-3.3</td>
<td>-0.034</td>
<td>3.25</td>
</tr>
<tr>
<td>RF</td>
<td>-0.085</td>
<td>-8.5</td>
<td>-0.025</td>
<td>2.99</td>
</tr>
<tr>
<td>C</td>
<td>-0.061</td>
<td>-6.1</td>
<td>-0.085</td>
<td>-8.59</td>
</tr>
</tbody>
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[a]-Refer experimental design for position description

3.3.5 Simulation with bird models

Simulation with bird models provided a qualitative understanding of the effect of birds on the microclimate. It also helped visualize the effect of flocking. The heat generated by the birds contributed significantly to the fluid temperature. The average surface temperature of the bird models was 300.38 K at 600 RPM (30 Hz) with eleven simulated birds. Air velocity affected the surface temperature of the bird models significantly. Consequently, simulated birds placed in positions with higher velocity had lower average temperature. Although, the chamber floor area is much smaller compared to a commercial farm, temperature differences still exists, however it may not be very significant. Figures 3.6-3.8 show the different velocity profiles with eleven bird models in the core chamber at 600 RPM (30 Hz).
Velocity contours at 1200 RPM (60Hz) at air temperature of 303K with 30 birds are provided in Appendix II (figures A-2-1 to 5).

Figure 3.6 Velocity distribution at bird height (0.20 m) (with 11 birds)--Top view (Air flow in z-direction)

The birds in the center received more air flow but more birds placed close to one another results in lower velocities around the bird. This effect can be very important in large commercial farms where the stocking density is very high. An individual bird may not experience the same average flow rate. However, this can be tackled by increasing the
average air flow rate by installing more number of fans or by increasing fan speed.

Figure 3.7 Longitudinal cross section of velocity distribution (600 RPM) –Side view (Air flow in z-direction)

Figure 3.7 shows the velocity distribution in the plane close to center of the chamber slicing across the model bird. Again, the effect of close proximity of birds was observed. Figure 3.8 shows the flow trajectories of the velocity adjusted to the inlet height. The flow above the inlet height resulted in formation of vortices. The velocity increases at the center of the outlet due to the sudden contraction of the chamber at outlet.
An important aspect of modeling birds inside the chamber was to obtain the temperature variation in birds at various locations inside the core chamber. The fluid models showed significant difference in the bird model surface temperature at different locations inside the core chamber. The surface temperature of the bird models was largely affected by the feeders and flocking. The locations in the center of the chamber near the drinkers had higher velocities.
Generally, birds tend to remain in the center due to higher velocities and the presence of drinker. However, more numbers of birds in the center may result in lower air flow over the group of birds. Figure 3.9 shows the temperatures of the solid bird models at 600 RPM (30Hz) blower speed with air temperature of 293.23 K (20 °C).

![Figure 3.9 Surface temperatures of simulated bird models (600 RPM) with 11 birds –Top view (Air flow in z-direction)](image-url)

The bird model close to the feeders had higher temperatures compared to other locations. The mean surface temperature of the bird model behind the feeder was 2.4 °C greater than the average surface temperature of the bird models. The bird model placed near
the inlet had the least average surface temperature, 2 °C less than the mean surface temperature. The higher the inlet air temperature the greater was the surface temperature difference. At air temperature of 303K (about 30 °C) the difference in the mean surface temperature and the highest average surface temperature was 2.9 °C and was observed for the bird model behind the feeder. Figure 3.10 shows a cut plot of bird model surface temperature at bird height (0.20 m).

![Figure 3.10 Cut plot of bird surface temperature at bird height (600 RPM) with 11 birds – Top view (Air flow in z-direction)](image)

Figure 3.11- 3.14 shows the velocity contours for 30 bird models simulated inside the chamber. More number of birds did not affect the fluid temperature significantly except in the regions of higher bird density.
Figure 3.13 shows the flow trajectory in the whole chamber representing steady flow. Clear formation of vortices was observed above the inlet height. The figure also indicates the limitation of using meshing with lower resolution. Very small grid spacing would be needed to analyze eddies formation. However, the vortex formation was a consequence of chamber geometry and had no effect on the bird occupied region which is the main area of interest in this study.

Figure 3.11 Velocity distribution at bird height with 30 birds –Top view (Air flow in z-direction)
Figure 3.12 Cut plot of bird surface temperature at bird height (600 RPM) with 30 birds – Top view (Air flow in z-direction)
Figure 3.13 Flow trajectories of velocity with 30 birds (600 RPM)–Isometric view (Air flow in z-direction)

Figure 3.14 Cut plot of fluid temperature at bird height with 30 birds (600 RPM)–top view (Air flow in z-direction)
3.4 Conclusion and further studies

A performance analysis was carried out on a dedicated poultry engineering chamber designed for air quality and animal welfare studies. The flow in the animal occupied region was analyzed using CFD. The CFD simulated results were validated using accurate field measurements. The air velocities at 5 different positions and 2 different heights (Chapter 2) were used to check the agreement between the measured and the simulated results. Statistical analysis (ANOVA) revealed that there was no significant difference between the modeled and the measured air velocities. Two blower’s rotary speeds (RPMs) were considered in the study. The average air velocity at bird height at 600 RPM obtained using measured values was $0.794\pm0.156$ m/s and the average air velocity obtained using corresponding CFD model was $0.809\pm0.169$ m/s. The average air velocity at bird height at 1200 RPM obtained using measured values was $1.706\pm0.305$ m/s and the average air velocity obtained using corresponding CFD model was $1.642\pm0.395$ m/s. Greater variances were observed at bird height due to the presence of feeders in the line of flow. Height 3 had widely varying velocities due to eddy formation just above the inlet height. The mean velocity at height 3 at 600 RPM was $0.152\pm0.227$ m/s. The NMSE of the measured and simulated values was 0.007. An NMSE less than 0.25 is considered to be a good indicators of concordance. However, the boundary conditions play an important role in predicting accurate results.

The CFD models with simulated birds indicated higher bird surface temperature in locations with low flow rates and the difference was significant. Higher air temperatures may result in greater surface temperatures although the RPM can be adjusted to retain the temperature within the thermoregulation range. Grouping or flocking can have a significant impact on the average air flow over the group. However, the poultry tends to move around especially when the lights are on and during feeding. Higher temperatures drive birds to separate from one another unless there is not enough room for bird movement. Further, bird models which represent the body characteristics more closely and heat dissipation through layered skin would provide a deeper understanding of the heat interaction between the bird and the surroundings.
The application of CFD along with thorough validation with experimental results indicated the utility and the fair uniformity of air velocity in the poultry chamber rendering the chamber useful for conducting animal welfare and air quality studies. CFD simulations can be effectively used to predict the air velocity and hence the temperature inside the core chamber at different ventilation rates with error well within the standard limits. ASHRAE recommends an error percentage of less than 20 % for complex indoor flows (ASHRAE, 2009). According to Harall and Boon (1997) and Blanes-Vidal et al. (2008) the results obtained in this study can be considered reliable. However, the bird simulations are only an indicative of the real world conditions as experimental data was not available to validate the CFD results. Nonetheless, the simulations give a qualitative understanding of the indoor environment. Further, the model results coupled with specific mathematical filters may be employed to automate the whole system with a feedback loop for establishing a real time controlled environment.
REFERENCES


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APPENDICES
APPENDIX I Chamber flow rates & velocity plots

Table A-1-1 Flow rates and pressure drops at 10% damper (ΔP in inches of water)

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Table A-1-2 Flow rates and pressure drops at 30% damper (ΔP in inches of water)

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Table A-1-3 Flow rates and pressure drops at 50% damper (ΔP in inches of water)

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### Table A-1-4 Flow rates and pressure drops at 70% damper (ΔP in inches of water)

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<th>Coverage %</th>
<th>RPM</th>
<th>Chamber 1 Flow rate (CFM)</th>
<th>ΔP</th>
<th>Chamber 2 Flow rate (CFM)</th>
<th>ΔP</th>
<th>Chamber 3 Flow rate (CFM)</th>
<th>ΔP</th>
<th>Chamber 4 Flow rate (CFM)</th>
<th>ΔP</th>
<th>Chamber 5 Flow rate (CFM)</th>
<th>ΔP</th>
<th>Chamber 6 Flow rate (CFM)</th>
<th>ΔP</th>
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### Table A-1-5 Blower Specification

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<th>Model</th>
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<tr>
<td>Wheel dia.</td>
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<tr>
<td>Shaft dia.</td>
<td>0.03 m (1.4375 In.)</td>
</tr>
<tr>
<td>Inlet dia.</td>
<td>0.73 m (29 In.)</td>
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### Table A-1-6 Air Delivery at 1470 RPM (fan curve)

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<th>SP (in. water)</th>
<th>Air delivery (CFM)</th>
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Air velocity in core chamber at bird height

Figure A1-1 Velocity vs. RPM at 10% damper opening for chamber 1

Figure A1-2 Velocity vs. RPM at 50% damper opening for chamber 1
Figure A1-3 Velocity vs. RPM at 10% damper opening for chamber 2

Figure A1-4 Velocity vs. RPM at 30% damper opening for chamber 2
Figure A1-5 Velocity vs. RPM at 50% damper opening for chamber 2

Figure A1-6 Velocity vs. RPM at 70% damper opening for chamber 2
Figure A1-7 Velocity vs. RPM at 10% damper opening for chamber 3

Figure A1-8 Velocity vs. RPM at 30% damper opening for chamber 3
Figure A1-9 Velocity vs. RPM at 50% damper opening for chamber 3

Figure A1-10 Velocity vs. RPM at 70% damper opening for chamber 3
Figure A1-11 Velocity vs. RPM at 10% damper opening for chamber 4

Figure A1-12 Velocity vs. RPM at 50% damper opening for chamber 4
Figure A1-13 Velocity vs. RPM at 10% damper opening for chamber 5

Figure A1-14 Velocity vs. RPM at 30% damper opening for chamber 5
Figure A1-15 Velocity vs. RPM at 50% damper opening for chamber 5

Figure A1-16 Velocity vs. RPM at 70% damper opening for chamber 5
Figure A1-17 Velocity vs. RPM at 10% damper opening for chamber 6

Figure A1-18 Velocity vs. RPM at 50% damper opening for chamber 6
Chamber to chamber air velocity comparison at bird height

Figure A1-19 Velocity vs. RPM at 10% damper opening at position ‘LB’

Figure A1-20 Velocity vs. RPM at 30% damper opening at position ‘LB’
Figure A1-21 Velocity vs. RPM at 50% damper opening at position ‘LB’

Figure A1-22 Velocity vs. RPM at 70% damper opening at position ‘LB’
Figure A1-23 Velocity vs. RPM at 10% damper opening at position ‘RF’

Figure A1-24 Velocity vs. RPM at 50% damper opening at position ‘RF’
Figure A1-25 Velocity vs. RPM at 70% damper opening at position ‘RF’

Figure A1-26 Velocity vs. RPM at 10% damper opening at position ‘RB’
Figure A1-27 Velocity vs. RPM at 30% damper opening at position ‘RB’

Figure A1-28 Velocity vs. RPM at 50% damper opening at position ‘RB’
Figure A1-29 Velocity vs. RPM at 70% damper opening at position ‘RB’
APPENDIX-II

CFD Simulations at inlet air temperature of 303.23K (30°C)

Figure A2-1 Velocity distribution at bird height with 30 birds 1200 RPM –Top view (Air flow in z-direction)
Figure A2-2 Surface temperatures of simulated bird models–Top view (Air flow in z-direction)
Figure A2-3 Cut plot of bird surface temperature at bird height with 30 birds – Top view (Air flow in z-direction)
Figure A2-4 Flow trajectories of velocity with 30 birds (600 RPM)–Isometric view (Air flow in z-direction)
Figure A2.5 Cut plot of fluid temperature at bird height with 30 birds - Top view (Air flow in z-direction)
1. Matlab code for calculating average air velocity in exhaust duct – Phase-1 testing

Matlab code-

%Input velocity readings-output curve fitting equations
clear all; close all; clc;
fvel=input('Enter matrix:');
traverse_point=[14.875 12.75 10.625 8.5 6.375 4.25 2.125];
rows=size(fvel,1);
t_1=[0.544^2;0.544;1];
t_2=[2.295^2;2.295;1];
t_3=[5.457^2;5.457;1];
t_4=[11.543^2;11.543;1];
t_5=[14.705^2;14.705;1];
t_6=[16.456^2;16.456;1];
t=horzcat(t_1,t_2,t_3,t_4,t_5,t_6);
for k=1:rows
  rowise=fvel(k,:);
  set_eq(k,:)=polyfit(traverse_point,rowise,2);
  for i=1:6
    t_7=t(:,i);
    vel(i)=set_eq(k,:)*t_7;
  end
  Avg_vel(k)=mean(vel);
end

2. R code for comparing within chamber air velocities and plotting using ggplot2

#Plot and compare left front position of all chambers
c1=read.csv("C1-RF.csv",header=TRUE)
c2=read.csv("C2-RF.csv",header=TRUE)
c3=read.csv("C3-RF.csv",header=TRUE)
c4=read.csv("C4-RF.csv",header=TRUE)
c5=read.csv("C5-RF.csv",header=TRUE)
c6=read.csv("C6-RF.csv",header=TRUE)
left_front=cbind(c1[,1],c2[,1],c3[,1],c4[,1],c5[,1],c6[,1])
left_front=as.data.frame(left_front)
colnames(left_front)=c("Chamber 1","Chamber 2","Chamber 3","Chamber 4","Chamber 5",
                     "Chamber 6")
left_front$RPM=c5[,2]
left_front$RPM=rep(c(400,600,800,1000,1200,2000),4)
open=c(10,30,50,70)
damper=rep(open,each=6)
left_front$damper=damper
plot_chamber=function(opening){
  left_front_RPM=subset(left_front,left_front[,8]==opening)
  left_front_RPM=left_front_RPM[,1:7]
  left_front_RPM=melt(left_front_RPM,id="RPM")
colnames(left_front_RPM)=c("RPM","Chamber","Velocity")
title=paste("Left-Back Position- RPM Vs Velocity\nfor Different Chambers @",opening,"%","opening",sep="")
ggplot(left_front_RPM,aes(x=RPM,y=Velocity,colour=Chamber))+geom_line(aes(line=Chamber),size=0.9)+
  ylab("Velocity(m/s)")+ xlab("RPM")+
  theme(plot.title=element_text(lineheight=0.8,face="bold"),
        panel.background=element_rect(fill="grey80"))
}

3. **R code for comparing chamber to chamber air velocities and plotting using ggplot2**

    LB=read.csv("C6-LB.csv",header=TRUE)
    LF=read.csv("C6-LF.csv",header=TRUE)
RB=read.csv("C6-RB.csv",header=TRUE)
RF=read.csv("C6-RF.csv",header=TRUE)
C=read.csv("C6-C.csv",header=TRUE)
vel=cbind(LB[,1],LF[,1],RB[,1],RF[,1],C[,1])
vel=as.data.frame(vel)
colnames(vel)=c("LB","LF","RB","RF","C")
vel$RPM=rep(c(400,600,800,1000,1200,200),4)
damper=rep(open,each=6)
open=c(10,30,50,70)
vel$Damper=damper
library(ggplot2)
require(reshape2)
plot_chamber=function(opening){
vel_RPM=subset(vel,vel[,7]==opening)
vel_RPM=vel_RPM[,1:6]
vel_RPM=melt(vel_RPM,id="RPM")
colnames(vel_RPM)=c("RPM","Position","Velocity")
title=paste("Chamber 3- RPM Vs Velocity\n for Different Positions @ ", opening,"\%","opening",sep="")
ggplot(vel_RPM,aes(x=RPM,y=Velocity,colour=Position))+geom_line(aes(linetype= Position),size=0.9)+
  xlab("RPM")+ylab("Velocity (m/s)")+
  theme(plot.title=element_text(lineheight=0.8,face="bold"),
        panel.background=element_rect(fill="grey80"))
}
plot_chamber(10); plot_chamber(30); plot_chamber(50); plot_chamber(70)