5.1 Introduction

Experiments are the only means by which cause and effect can be established. It allows for precise control of variables with which real performance characteristics of any machine can truly be ascertained. The experimental method must consist of well organized standard procedures and measures which allow it to be easily repeated and form a sound base for validation of theoretical results.

It was clear from literature review that the number of blades [14, 31, 44] and slip factor [22, 76] has remarkable influence on fan performance.

Eck Bruno [14], W. J. Kearton [25], states that the optimum number of blades of a radial impeller can only be truly ascertained by experiments. Hence one of the objectives of present work was to establish influence of number of blades on fan performance by carrying out experiments under varying speed and varying flow conditions.

For the proper design of centrifugal machines, it is essential to estimate the slip factor correctly. Several co-relations as well as empirical equations are used in literature for estimating the slip factor [9, 14, 75, 76, 78, 79, 83, 87]. Hence, experiments are also planned to evaluate values of slip factor and to find out variation of slip factor along the blade profile at different locations of volute casing, with two different impeller geometries.

It is revealed from literature review that centrifugal fan design suggested by different researchers differ widely. Therefore the performance of fans fabricated as per individual and explicit design methodologies suggested by Church, Osborne and retrieved from fundamental principles of fluid flow having minimum assumptions, should be critically evaluated. Based on experimental results obtained, it is observed that there exists a wide performance difference amongst fans under study. It concludes that each design method, as an individual, is not performing as marked. It has also revealed that there is a need to develop unified design methodology.
In later stage, forward and backward curved radial tipped centrifugal fans are fabricated as per proposed unified design methodology and their performance is recorded by conducting experiments. Their results have clearly shown that fans based on unified design are good enough to achieve desired performance. Major performance parameters achieved are on higher side of design point.

Performance of centrifugal fan is highly affected by hydraulic, volumetric and power losses [9, 28, 34]. Successful design methodology must be accompanied by correct estimation of such losses which can give desired and designed performance. The centrifugal fan, because of their relatively longer flow passages and greater turning of flow, suffer higher losses compared to axial type [28]. Scope of experimental work was extended towards experimental measurements of hydraulic, volumetric and power losses occurring in flow passage of radial tipped centrifugal fan and henceforth to improve suggested unified design methodology.

To achieve desired goals a systematical development of fans, test set up and experimental matrix was planned in phase wise manner as follows:

- **Phase - I:** Experimental Optimization of Finite Number of Blades under Varying Speed Conditions
  - This phase is further sub divided in to two stages.
    - **Stage 1:** Influence of Suction Pressure on Performance of the Fan
    - **Stage 2:** Optimization of Finite Number of Blades

- **Phase - II:** To evaluate the value of slip factor at Varying Number of Blades and Speed Conditions as well as to check the effect of blade profile on Slip Factor

- **Phase - III:** Comparative Assessment of Explicit Design Methodologies.

- **Phase - IV:** Performance Evaluation of Unified Design Methodology and Comparative Performance Evaluation of Forward and Backward Curved Radial Tipped Centrifugal Fan.

- **Phase - V:** Assessment of Theoretical and Experimental Losses.

### 5.2 Phase Wise Development of Fans, Test Set-Up and Test Procedure

Each phase of study specifically requires a systematic development of fans, test set up and test procedure relevant to set goals.
Accordingly for each phase of study, the relevant fabrication of fans, its test set up and test procedure has been systematically devised and presented in subsequent sections.

5.2.1 Phase - I: Experimental optimization of finite number of blades under varying speed conditions

The general design procedure is adopted as per the methodology outlined by Eck Bruno [14]. It was streamlined by Bela Mishra [30]. Profile of the blade is designed as per the methodology suggested by Austin Church [26]. The numbers of blades are estimated as per the empirical formulas suggested by Eck Bruno, Pfleiaderer and Stepanoff [9]. After calculating the optimum numbers of blades by empirical formulas, they are verified experimentally in the steps of 8, 12, 16 and 24.

Transparent forward curved radial tipped centrifugal fan was fabricated from 3 mm and 4 mm acrylic sheets. Acrylic is a commercial name of poly methyl methacrylate (PMMA) thermoplastic material. PMMA is in solid phase at room temperature and it is fully transparent. It helps in better flow visualization and having less coefficient of friction (which results in a thin boundary layer). The transition temperature \( T_g \) of commercial grades of PMMA range from 85 to 165°C. Required shape and profile from acrylic sheet is fabricated by cutting, heating and compression molding methods. Heating, clamping and cooling are carried out such that stress concentration and distortion can be avoided after cooling. To get exact geometrical profile of parts, patterns/replicas are made from sheet metal or wood. Strong adhesives and fasteners are used to join individual parts together to prepare leak proof assembly. Finally, parts are machined to get required dimensions. Impeller is dynamically balanced. Blades are fastened between front and back shroud plate in such a way that number of blades can be varied in the steps of 8, 12, 16 and 24 without affecting balancing. Impeller, suction duct and volute casing are also made from acrylic material so that flow visualization is possible when coloured dust is mixed in flow streams to see velocity vectors and flow stream lines. Acrylic material is brittle at room temperature and possesses less impact strength and hence the fabrication work of this material is challenging.

First stage of experiments was designed to study the influence of suction pressure on performance of centrifugal fan. Second stage experiments were conducted
to optimize finite number of blades under varying speed and suction pressure. Plate 5.1 shows transparent fan assembly fabricated for both stages of phase I experiments.

Plate 5.1 Transparent Fan Assembly for Phase I Experiments

The stage 1 of phase 1 is designed to study the influence of suction pressure on performance of centrifugal fan. The design point parameters were 1150 Pa static stage pressure rise at 0.417 m³/s volume flow rate and speed of impeller 2800 rpm. The number of blades is varied in four steps of 8, 12, 16 and 24. The suction pressure is varied with the help of orifice plates of different diameter. Suction pressure variation is carried out by using six orifice plates of diameter 80, 90, 110, 120, 130, 150 mm and full opening of suction duct. Here, suction pressure variation lies in the range of 167 N/m² to 1364 N/m². These experiments have revealed that the orifice of 110 mm diameter offers the design point stage pressure rise at optimum static efficiency.

The stage 2 of phase 1 is designed to find optimum number of blades at constant suction pressure and under varying speed conditions. Number of blade is varied in four steps of 8, 12, 16 and 24. During each set up of experiment, the performance parameters are measured. To ascertain the optimisation process, the fan performance have been evaluated under off design condition keeping the orifice of diameter 110 mm constant and varying the speed of fan through variac. The results received during both stages of this phase of experiments are discussed in detail in subsequent chapters. Following formulas are used to calculate the test parameters. Sample calculations for phase 1 experiments are given in Annexure C.
• **Input power**  \( = \sqrt{3} \times \text{voltage} \times \text{current} \times \cos\phi, \text{in} \ W \)

  Where \( \phi = \text{Power factor (which is assumed as 0.75)} \)

• **Shaft power**  \( = \text{input power} \times \text{efficiency} \)

  (Efficiency of an induction motor is considered 80%)

• **Discharge**  \( = C_d \times A_2 \times \sqrt{2 \times g \times h_m \times ((\rho_w/\rho) - 1)} \)

  \( = 3.193 \times (\text{inlet diameter})^2 \times \sqrt{P_v} \)

Where \( P_v = \text{pressure head at vena-contracta of an orifice in m of H}_2\text{O}. \)

\( C_d = 0.62 \)

• Stagnation pressure \( P_o = \text{Static pressure} + \text{velocity pressure} = P + (\rho v^2)/2 \)

• Air power = differential pressure head \( \times \) discharge.

• Hydraulic efficiency \( \eta_{hy} = \text{air power} / \text{shaft power} \).

Plate 5.2 shows experimental set up used for phase I experiments.

**Plate 5.2 Experimental Setup for Phase I for Optimization of Finite Number of Blades**

### 5.2.2 Phase - II: Experimental investigations on slip factor at varying number of blades and speed conditions

Another important factor influencing the designing of centrifugal fan is slip loss. Slip loss is defined as the ratio of actual and ideal values of the whirl velocity components at exit of impeller. It has significant effect on fan performance. It is essential to estimate the slip factor correctly while designing centrifugal fans. During literature survey, it is studied that several co-relations and empirical relations are used
for estimating the slip factor. Analytical results of these relations differ widely for same input data. Most of all concludes that the value of slip factor is constant and it is dependent of impeller geometry only [31, 91]. However, fewer historical evidences and experience has challenged this statement and have pointed that above statement is partially correct and the slip factor not only depends on the geometry of the impeller but also on the specific speed and flow rate [73]. Moreover, the value of slip factor does not remain constant at any location at the exit from impeller blades.

The objective of this phase is to find experimental value of slip factor and effect of number of blades on slip factor. In addition, work is carried out to check the effect of volute location and blade width on slip factor. A specially designed, fabricated and calibrated three hole pressure probe is used to sense optimum magnitude of actual static pressure and total pressures at impeller exit. Pressure probe is mounted on the sliding fixture which is having angular movement mechanism. So, the angular position of probe can be varied to sense maximum velocity direction at optimum flow angle.

As per the basic requirement for the experimentation, the three hole pressure probe is made from very small diameter surgical steel tubes (0.6 and 0.8mm ID, 1 and 1.5mm OD). Therefore variation and deviation in the flow velocity due to its insertion within the flow region can be considered negligible. This three hole probe is very sensitive to measure small amount of pressure at any test location. It is also able to sense and determine the exact direction of flow. Probe is light in weight but highly rigid and strong for sustaining the high flow pressures within the fan assembly.

Measurements of flow velocity at volute casing and at the centre of exit area were made with Pitot tube and three hole pressure probe. This exercise was repeated for all set of given number of blades. Static, total and dynamic pressure readings are recorded. Velocity is calculated for known volute casing exit area of 0.010725 m². Table 5.1 shows velocity and discharge volume flow measurements done with Pitot tube and three hole pressure probe. The difference in measurement leads to establish coefficient of velocity measurement for three hole pressure probe. Thus, three hole pressure probe is calibrated with standard pitot tube.
Table 5.1 Three Hole Pressure Probe Calibration Readings

<table>
<thead>
<tr>
<th>Number of Blades</th>
<th>Pitot Tube Readings</th>
<th>Pressure Probe Readings</th>
<th>Coefficient of Velocity CV</th>
<th>Average CV</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Velocity in m/s</td>
<td>Discharge</td>
<td>Velocity in m/s</td>
<td>Discharge</td>
</tr>
<tr>
<td>8</td>
<td>20.34</td>
<td>0.218</td>
<td>19.24</td>
<td>0.206</td>
</tr>
<tr>
<td>12</td>
<td>20.12</td>
<td>0.216</td>
<td>18.31</td>
<td>0.196</td>
</tr>
<tr>
<td>16</td>
<td>19.12</td>
<td>0.205</td>
<td>17.83</td>
<td>0.191</td>
</tr>
<tr>
<td>24</td>
<td>18.90</td>
<td>0.203</td>
<td>17.21</td>
<td>0.185</td>
</tr>
</tbody>
</table>

The experiments were carried out on centrifugal fan assembly used for phase I with minor modifications. To calculate local slip factors one must have measured data of actual velocity at that location. Measurements are made to find actual flow velocity, static and total pressure at impeller exit and exit flow angles at various locations of volute casing. Tapings are made in volute casing at 0°, 30°, 60°, 90°, 120°, 150°, 180° and 210°. These tapings are shown by A,B,C...H test locations to take pressure reading. The criteria that have to be taken into consideration while selecting the test-locations are as under:

- It must cover the maximum circumferential region of the impeller. Here eight main test locations are selected (from A to H) as shown in Plate 5.3.

Plate 5.3 Probe Insertion Test Locations

- It must cover the total width of the impeller or blade in order to study the effect of blade width on velocity profile as well as slip factor. Along the width, seven sub-locations are selected for each main test locations.
The gap between the probe and the circumference of the impeller must be as minimum possible.

Table 5.2 shows test locations with notation to each sub location. Here test locations are 8 while sub test locations are 56 in number.

Table 5.2 Notations of Probe Insertion Test Locations with Sub-Locations

<table>
<thead>
<tr>
<th>Distance along the blade width (mm)</th>
<th>0</th>
<th>15</th>
<th>30</th>
<th>37</th>
<th>45</th>
<th>60</th>
<th>75</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1</td>
<td>2</td>
<td>3</td>
<td>4</td>
<td>5</td>
<td>6</td>
<td>7</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Main locations along the circumference of the impeller</th>
<th>Sub location points along the blade width</th>
</tr>
</thead>
<tbody>
<tr>
<td>Degree.</td>
<td>Main Location points</td>
</tr>
<tr>
<td>0</td>
<td>A, A, A, A, A, A, A</td>
</tr>
<tr>
<td>30</td>
<td>B, B, B, B, B, B, B</td>
</tr>
<tr>
<td>90</td>
<td>D, D, D, D, D, D, D</td>
</tr>
<tr>
<td>120</td>
<td>E, E, E, E, E, E, E</td>
</tr>
<tr>
<td>150</td>
<td>F, F, F, F, F, F, F</td>
</tr>
</tbody>
</table>

Probe can traverse axially along blade width to take seven different sub location readings as per insertion location. Plate 5.4 and 5.5 shows three hole probe along with measurement of static pressure at that location. Plate 5.6 shows probe gun assembly used to change angular position of side tubes of probe.
Plate 5.7 shows experimental setup with measurement instruments with axial probe sliding and traversing arrangements.

Plate 5.7 Experimental Test Setup for Slip Factor Measurements for Phase II

Theoretical velocity diagram for radial tipped blade exit is prepared with the help of theoretical values of tangential velocity component \( U_2 \), radial velocity component \( V_{r2} \) and flow angle \( \theta_2 \). Actual velocity diagram is prepared with the help of
measured components of absolute velocity $V_2'$ at measured flow angle $\theta_2'$ at impeller exit.

The actual absolute velocity $V_2'$ is measured with the help of probe while the actual whirl angle is measured with the help of angular movement mechanism provided on the sliding fixture. Theoretical component of tangential velocity $U_2$ is also used to construct actual velocity diagram. Actual component of radial velocity $V_{r2}'$ and actual blade angle $\beta_2'$ are received from actual velocity diagram. Construction of actual velocity diagram is shown in Figure 5.1.

![Figure 5.1 Impeller Exit Theoretical and Actual Velocity Triangles](image)

From the above velocity diagram, whirl component of absolute velocity,

$$V_{w2}' = V_2' \cos \theta_2'$$

Slip factor is defined as the ratio of actual to theoretical whirl components of absolute velocities.

$$\text{Slip factor } \mu = \frac{V_{w2}'}{V_{w2}}$$

Thus, experimentally, slip factor is calculated at all sub-locations for 8, 12, 16 and 24 number of blades. Slip factors of each test location is calculated by averaging slip factor values of their sub-locations.

5.2.3 Phase - III: Comparative assessment of explicit design methodologies

It is revealed from literature review that centrifugal fan design suggested by different researchers differ widely. This phase of experimental work is planned for experimental assessment of fan performance, fabricated as per individual and explicit
design methodologies suggested by Church, Osborne and retrieved from fundamental principles of fluid flow having minimum assumptions as discussed in chapter 3.

For this phase, Impeller, casing and test airway ducts are fabricated from 2 mm thick mild steel sheets. The forward curved radial tipped centrifugal fans are fabricated as per the dimensions received after final iteration based on all three, i.e. fundamental, Church and Osborne design methodologies. Fabrication work is carried out by using 14 gauge (2.03 mm) mild steel sheets. Impellers are double shrouded with 16 numbers of blades as optimized by experiments of phase I. Inlet and discharge airway ducts are fabricated as per IS 4894:1987 test standards. All impellers are made static and dynamically balanced on balancing machine. Special templates are made to get desired blade and casing profile. All joints are welded and proper care is taken to avoid distortion. Volume flow variation dampers are used to change mass flow rates across the fan. Arrangement is also made to lock the damper at various flow positions for constant mass flow rate. Plate 5.8 to 5.11 shows impellers, casing and ducts fabricated for phase III.

Plate 5.8 Impeller and Volute Casing as per Fundamental Design
Chapter – 5: Development of Fans, Test Set-Up and Experimental Matrix

Plate 5.9 Impeller and Volute Casing as per Church Design

Plate 5.10 Impellers and Volute Casing as per Osborne Design
To get correct evaluation and performance of the fans, it is necessary to have a very precise experimental set-up and instrumentation by which essential measurements can be achieved with desired accuracy and reliability. These tests are conducted according to standard test code IS: 4894-1987, Indian Standard Specification for Centrifugal Fans (First Revision), Reaffirmed in 1994 [17]. Precise and calibrated measuring and sensing instruments are used to measure flow, pressure, velocity, power and rotational speed. Proper fixtures, supports and stands are used for sturdy and steady setup. Basic experimental set up is similar for all phases and in accordance with IS standard.

Three phase induction motor is used for impeller rotation. Variac is used to change speed of rotation in the steps of 500, 1000, 1500, 2000, 2500 and 2800 rpm. Ammeter, wattmeter and voltmeter are used for electrical measurements.

Observations are recorded for actual stage pressure head developed across the fan, average air discharge, shaft power, airpower developed, static and stagnation efficiencies for each centrifugal fan under study.

Based on experimental results obtained and evaluation made, it is recognized that there exists a wide performance difference amongst fans under study. No fans are performing as marked.
5.2.4 Phase - IV: Unified design methodology and comparative performance evaluation of forward and backward curved radial tipped centrifugal fan

Performance evaluation of experiments carried out under phase III has revealed that all three explicit design methodologies are not performing as marked. It is observed that there exists a wide performance difference amongst fans under study. It has indicated a need to develop unified design methodology to get designed and desired fan performance. Hence, successful outcomes of fundamental, Church and Osborne designs are incorporated and unified design methodology for radial tipped centrifugal fan is developed.

This phase of research work is planned for performance evaluation of forward and backward curved impeller fans fabricated as per unified design methodology. All the fabrication work is carried out from mild steel sheets. Scroll casing is designed and fabricated as per four point method as described in chapter 3. The special templates and fixtures are made for achieving desired blade profile and casing to avoid distortion after welding. Impellers are dynamically balanced after fabrication. Tests are conducted according to test code IS: 4894-1987. Plate 5.12 shows experimental test setup used for phase IV.

Plate 5.12 Experimental Test Setup Used for Unified Design Methodology

Plate 5.13 to 5.17 shows fabrication details of blade profile, impeller in clamped position, forward and backward curved radial tipped impellers, inlet and outlet ducts and scroll casing, respectively.
Plate 5.13 Unified Design Blade Profile

Plate 5.14 Unified Design FCRT Impeller Fabrications

Plate 5.15 Unified Design FCRT and BCRT Impellers
5.2.5 Phase - V: Assessment of theoretical and experimental losses

Overall efficiency of any machine depends on the various losses occurring in the machine at different stages [9, 28 and 34]. There are various types of losses occurring when the fluid passes from inlet duct to outlet duct of a turbo machine. The major losses are classified into three categories as hydraulic or pressure losses, mechanical or power losses and volume flow or leakage losses. Hydraulic losses reduce the available pressure head developed by the impeller thereby reducing the Hydraulic efficiency. Mechanical losses are mainly due to disc Friction and friction between rotating shaft and the journal bearing. Leakage losses reduce the quantity of fluid delivered per unit time and hence reduce the volumetric efficiency.

Performance of centrifugal fan is highly affected by hydraulic, volumetric and power losses [9, 28 and 34]. Successful design methodology must be accompanied by
correct estimation of such losses which can give desired and designed performance. The centrifugal fan, because of their relatively longer flow passages and greater turning of flow, suffer higher losses compared to axial type. Literature review on these losses has exposed that loss estimation method proposed by different researchers differ widely.

Scope of experimental work under phase V was to make experimental measurements of hydraulic, volumetric and power losses occurring in flow passage of radial tipped centrifugal fan and henceforth to improve suggested unified design methodology.

A separate fan assembly based on unified design methodology is fabricated for 0.159 m$^3$/s designed flow rate with 456 Pa static stage pressure rise intended for loss estimation and verification of slip factor. It is worth to mention over here that many researchers have observed dependency of slip factor on impeller size and its geometry volume flow, number of vanes and speed [9, 14, 75, 76, 79, 83 and 87]. Accordingly, the measurements of slip factor in a different geometrical size of impeller were undertaken here in.

Impeller is fabricated from mild steel while volute casing is fabricated from transparent acrylic material for probe insertion and flow visualization. Plate 5.18 and Plate 5.19 shows impeller and transparent casing fabricated for phase V experiments.

Plate 5.18  Radial Tipped Impeller
Velocity and pressure developed across the impeller and across the centrifugal fan stage are measured with a developed and calibrated five hole pressure probe as per G. L. Morrison [104] analysis technique.

Static, total and dynamic pressure readings are recorded. Velocity is calculated for known volte casing exit area of 0.010286 m². Readings are taken at the exit of the centrifugal fan at a fully open position without damping. Table 5.3 shows velocity and discharge volume flow measurements done with Pitot tube and five hole pressure probe. The difference in measurement leads to establish coefficient of velocity measurement for five hole pressure probe. Thus, five-hole pressure probe is calibrated with the help of a standard pitot tube. Calibration of probe is essential to ensure the validity and accuracy of measurements taken by it.

**Table 5.3 Calibration of Five Hole Pressure Probe at 2800 rpm for Forward Curved Impeller Centrifugal Fan**

<table>
<thead>
<tr>
<th>Instrument</th>
<th>Total pressure reading at outlet duct</th>
<th>Static pressure</th>
<th>Dynamic pressure</th>
<th>Velocity of fluid at outlet</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10° Inclined manometer mm of water</td>
<td>Vertical manometer mm of water</td>
<td>N/m²</td>
<td>N/m²</td>
</tr>
<tr>
<td>Pressure</td>
<td>202</td>
<td>35.07</td>
<td>340.00</td>
<td>211.7</td>
</tr>
<tr>
<td>Standard</td>
<td>205</td>
<td>35.59</td>
<td>340.00</td>
<td>212</td>
</tr>
<tr>
<td>pitot tube</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Since the readings obtained by five-hole pressure probe and standard Pitot tube are nearly equal, the value of velocity coefficient is one.

Plate 5.20 shows five hole pressure probe with its holding and traversing mechanism. Plate 5.21 and 5.22 shows probe insertion locations and experimental set up, respectively.
5.3 Experimental Setup and Measurements

Fan testing includes measurements of airflow, pressure, power, and efficiency. Fan performance curves can be developed from these measured data. All the fans fabricated as per individual and unified design methodologies, are tested experimentally to get their optimum performance. These tests are conducted according to standard test code IS: 4894-1987, Indian Standard Specification for Centrifugal Fans (First
Revision), Reaffirmed in 1994 [17]. Precise and calibrated measuring and sensing instruments are used to measure flow, pressure, velocity, power and rotational speed. Proper fixtures, supports and stands are used for sturdy and steady setup. Basic experimental set up is similar for all phases and in accordance with IS standard. Its line diagram is shown in Figure 5.2.

**Figure 5.2  Line Diagram of Experimental Setup as per IS 4894:1987**

The various measurements and their instrumentation for experimental investigations include:

- **Pressure Measurement**

  For suction static pressure measurements in airway duct three section AA, BB and CC are selected as per IS-Standard. At each section four co planer side tapings of 4.5 mm diameter are made as shown in Figure 5.2. Aluminium nipples of 30 mm length are fitted at these side tapings. Tubes made of Poly-carbon material connect the nipples of four side tapings and a square test box at each section. Square test box is used to minimize effect of turbulence on static pressure measurements as well as to get average pressure reading at a given section. Test box is connected to U-tube manometer as shown in Figure 5.2 to measure average static pressure at section AA, BB and CC, in terms of head of water column.

  Similar way, static pressure distribution is measured at 90º, 120º, 180º, 240º, 300º and 360º tapings in volute casing as shown in line diagram. Atmospheric pressure is verified with the help of laboratory aneroid barometer.
• **Speed Measurement**
  The number of revolutions of the motor shaft is measured by contact type digital tachometer.

• **Discharge Measurement**
  The velocity is measured at five different co-planer locations at outlet with the help of digital anemometer. From these readings, average discharge velocity \( V_d \) and thus discharge is calculated. This is also cross checked by measurement of static and total pressure as described in IS 4894:1987.

• **Electrical Measurement**
  The digital voltmeter and ammeter are used for voltage measurement across the phase and current drawn, respectively. The power consumption is also measured by two Wattmeter placed in series. Electrical circuit diagram is shown in Figure 5.3.

![Figure 5.3 Electrical Circuit Line Diagram](image)

Table 5.4 shows list of various instruments used for each phase of experimental work.
Table 5.4 Instrumentation Details

<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Parameter</th>
<th>Locations</th>
<th>Instruments used</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Static pressure</td>
<td>Inlet duct, outlet duct, Impeller inlet, impeller outlet, volute outlet, clearance gap between impeller inlet and casing</td>
<td>Static pressure tapings and pressure probe</td>
</tr>
<tr>
<td>2</td>
<td>Total pressure</td>
<td>Inlet duct, outlet duct, and impeller outlet</td>
<td>Five hole pressure probe</td>
</tr>
<tr>
<td>3</td>
<td>Speed of rotation</td>
<td>Drive shaft</td>
<td>Digital tachometer</td>
</tr>
<tr>
<td>4</td>
<td>Current</td>
<td>Electric Motor</td>
<td>Ammeter</td>
</tr>
<tr>
<td>5</td>
<td>Voltage</td>
<td>Electric Motor</td>
<td>Voltmeter</td>
</tr>
<tr>
<td>6</td>
<td>Power</td>
<td>Electric Motor</td>
<td>Wattmeter</td>
</tr>
</tbody>
</table>

5.4 Instrument Specifications

Table 5.5 shows list of instruments used for phase I to phase V experimental measurements with their make, model number, least count, range and accuracy. Various instruments are used for measurements of pressure, velocity and speed of rotation. Electrical measurements of current, voltage and power are made to calculate input and fan shaft power.

Table 5.5 Specifications of the Instruments used for Experiments

<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Instruments</th>
<th>Model No.</th>
<th>Make</th>
<th>Range</th>
<th>Least Count</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Digital tachometer</td>
<td>DT-2235B</td>
<td>Lutron</td>
<td>2.5 to 19,999 rpm</td>
<td>0.1 rpm</td>
<td>±0.05%</td>
</tr>
<tr>
<td>2</td>
<td>Digital anemometer</td>
<td>AM-4202</td>
<td>Lutron</td>
<td>0.4-30 m/s</td>
<td>0.1</td>
<td>±1.0%</td>
</tr>
<tr>
<td>3</td>
<td>Digital Multimeter</td>
<td>603</td>
<td>Meco</td>
<td>0-750 V</td>
<td>1 V</td>
<td>±1.5%</td>
</tr>
<tr>
<td>4</td>
<td>Digital Multimeter</td>
<td>603</td>
<td>Meco</td>
<td>0-20 A</td>
<td>0.01 A</td>
<td>±2.0%</td>
</tr>
<tr>
<td>5</td>
<td>Analog Wattmeter</td>
<td>-</td>
<td>AE</td>
<td>0-1500 W</td>
<td>5 W</td>
<td>±1.5%</td>
</tr>
<tr>
<td>6</td>
<td>AC Variac</td>
<td>-</td>
<td>AE</td>
<td>0-415 V</td>
<td>4 V</td>
<td></td>
</tr>
</tbody>
</table>
5.5 Uncertainty Analysis

An uncertainty analysis is carried out according to Kline and Mclintock method as compiled by Hollman J.P. [141] and given below in Table 5.6.

**Table 5.6 Uncertainties in Measurements**

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Uncertainty%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input Power</td>
<td>±2.1</td>
</tr>
<tr>
<td>Stage Pressure Rise, Pa</td>
<td>±0.1</td>
</tr>
<tr>
<td>Volume Flow, m³/s</td>
<td>±1.5</td>
</tr>
<tr>
<td>Static Efficiency, %</td>
<td>±2.0</td>
</tr>
</tbody>
</table>

It is observed that all uncertainties in measurements are well within ± 5.0% of design point values.