Experimental Measurements of the Mean Flow Field in Wide-Angled Diffusers: A Data Bank Contribution

Karanja Kibicho and Anthony Sayers

***Abstract*—Due to adverse pressure gradient along the diverging walls of wide-angled diffusers, the attached flow separates from one wall and remains attached permanently to the other wall in a process called stalling. Stalled diffusers render the whole fluid flow system, in which they are part of, very inefficient. There is then an engineering need to try to understand the whole process of diffuser stall if any meaningful attempts to improve on diffuser efficiency are to be made. In this regard, this paper provides a data bank contribution for the mean flow-field in wide-angled diffusers where the complete velocity and static pressure fields, and pressure recovery data for diffusers in the fully stalled flow regime are experimentally measured. The measurements were carried out at Reynolds numbers between 1.07×10^5 and 2.14×10^5 based on inlet hydraulic diameter and centreline velocity for diffusers whose divergence angles were between 30° and 50°. Variation of Reynolds number did not significantly affect the velocity and static pressure profiles. The wall static pressure recovery was found to be more sensitive to changes in the Reynolds number. By increasing the velocity from 10 m/s to 20 m/s, the wall static pressure recovery increased by 8.31%. However, as the divergence angle was increased, a similar increase in the Reynolds number resulted in a higher percentage increase in pressure recovery. Experimental results showed that regardless of the wall to which the flow was attached, both the velocity and pressure fields were replicated with discrepancies below 2%.

***Keywords*—Two-dimensional, wide-angled, diffuser, stall, separated flows, subsonic flows, diffuser flow regimes

I. INTRODUCTION

IN many engineering applications, diffusers are used to convert kinetic energy into pressure energy. The importance of the diffuser as a single, useful, fluid-mechanical element in wind tunnels and turbo-machinery has been widely known. Understanding of diffuser flows, therefore, is of paramount importance to the design of fluid-flow systems. In the last few decades, a lot of experimental and computational research have been devoted to this subject. Unfortunately, even turbulent flows in two dimensional diffusers are extremely complicated and our understanding of the details of energy transfer and dissipative losses inside a diffuser is still incomplete.

In order to meet some design constraints such as the overall size of a fluid-flow system, wide-angled diffusers with severe flow separation and poor efficiency, are often tolerated. The flow separation is caused mainly by the adverse pressure gradient along the walls of the diffuser, causing a back-flow, in a process known as 'stalling'. The back-flow behavior of stalled diffusers is uniquely related to the diffuser geometry as described in the flow regime chart of Fox and Kline [1].

The convention commonly used for two dimensional diffuser flows is given by Fig.1.

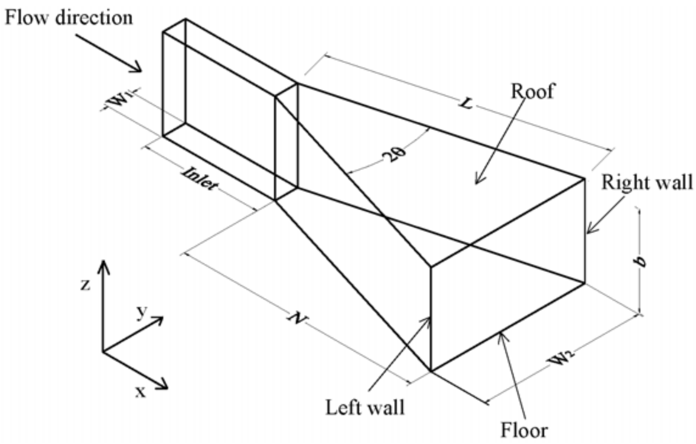

Fig. 1. Two-dimensional diffuser geometry and the frame of reference

The relationship between diffuser geometry and performance is mainly covered by Reneau et al.[2], Kline et al.[3], and Sagi and Johnston [4], among others. They concluded that the optimum effectiveness is achieved when the total divergence angle of the diffuser is approximately 7° and the ratio $\frac{N}{W_1}$ not exceeding 25. They further reported that at high divergence angles, the area ratio became less significant as a variable for determining pressure recovery. Even though their work did not address the flow-structure within the diffusers, and instead covered only the pressure recovery between the inlet and the outlet of the diffusers, the work of Reneau et al.[2] is widely used in the design and mapping of two-dimensional diffusers.

The stalling phenomena in diffusers can be related to the rate of boundary layer growth in the inlet duct and within the diffuser. McMillan and Johnston [5], Norbury [6], and Johnston and Powars [7] have argued that a uniform velocity profile carries the minimum momentum flux possible at a given flow rate, so transformation of a distorted profile to a more uniform one, even in a constant area duct, can result in an increase in static pressure (a process that occurs in diffuser tail pipes). However, uniform flow at the inlet of the diffuser

Karanja Kibicho is with the Department of Mechanical Engineering, Jomo Kenyatta University of Agriculture and Technology, Nairobi, Kenya. email: kkibicho@eng.jkuat.ac.ke
Anthony Sayers is with the Department of Mechanical Engineering, University of Cape Town, South Africa, email: Anthony.Sayers@uct.ac.za

is not always achieved in practice, since various upstream flow conditions like obstructions due to blades, struts, etc. may shed wakes. In any case, it is known that the introduction of the wide-angled diffuser causes severe non-uniformity in the upstream inlet channel. Based on this understanding therefore, the inlet core-flow velocity profile distortions can cause a significant change in the pressure-recovery performance and, by implication, the flow regimes.

If the basic mechanisms that control diffuser performance can be understood, then, this will lead to the design of systems which employ techniques that utilize these mechanisms in improving the efficiency of fluid machines. Once the separation of the flow from the walls in adverse pressure gradients is prevented by any method, then, that method can be applied to design an efficient diffuser with a large divergence angle and a shorter length. Hoffman [8] has argued that if such a method can enhance the transfer of free stream turbulent energy to the diffuser walls, then this transfer of energy will decrease the distortion of the velocity profiles within the diffuser and delay the onset of separation.

II. Experimental apparatus and instrumentation

To minimize the influence of diffuser geometry at the inlet and achieve a fully developed inlet flow, the aspect ratio, $\frac{b}{W_1}$, at the inlet and the normalized inlet duct length, $\frac{N}{W_1}$, were chosen to be 4, and 10, respectively.

The experimental apparatus is shown in Fig.2. Air was delivered by a radial flow fan into a 1.9 m long, rectangular-circular transition duct, which was connected to a 600 mm inner diameter, 3.7 m long circular duct. A 1 m long circular-rectangular transition duct transferred the air to a 400 mm x 100 mm x 1 m long straight inlet duct before its entry into the diffuser proper. After flowing through the diffuser the air was discharged back into the atmosphere.

The diffuser was of plane wall with the roof and floor walls running parallel to each other. The overall error in parallelism for the roof and floor was ± 0.25 mm. Further, the distances between the left and right walls at both entry to and exit from the diffuser were measured and the error in the divergence angle based on these measurements was found to be $\pm 0.24°$. All the plates for the straight inlet section and diffuser were machined from 10 mm thick transparent thermoplastic resin (Perspex) and all sections of the duct were bolted through flanges.

Before the diffuser was bolted to the straight inlet duct, measurements showed that the flow at the exit of this duct was symmetrical about the x axis in both the y and z directions. However, once the diffuser is connected to the straight duct, the presence of stall in the diffuser distorts the velocity profile at the exit of the straight duct [9], [10]. It was desired to measure the reference flow parameters at a location where the flow was symmetrical about the x-axis in both the y and z directions, and hence an arbitrary upstream location in the straight duct, $\frac{x}{W_1} = -2.35$, where these flow conditions were met, was chosen as a suitable flow reference location.

At the diffuser entry, the edges of the entry duct side plate and diffuser entry side plate were machined at appropriate angles to form the required divergence angle on assembly. These side plates were joined using chloroform, and the sharp edges at the entry corners smoothened very carefully by use of a fine file and sand paper.

To straighten the flow, wire mesh screens of 0.9 mm wire thickness, 3.3 mm square eye and 6 holes per square centimeter were installed at the rectangular-circular transition piece exit, circular duct-transition piece interface and transition piece-entry duct interface.

At the reference point, eight static pressure holes of 1.8 mm diameter, were drilled, two on each side of the duct. Due to the manufacturing difficulties of drilling the 1.8 mm holes at the exact diffuser entry, a similar set of eight holes were drilled at a location 2 mm upstream of the diffuser entry. This is the point at which the entry static pressure, P_1, was measured. The static pressures at the reference location and at the diffuser entry were averaged by connecting the eight tubes, to a common ring tube through 1.5 mm flexible vinyl tubing, at each of the two locations. The output of the common ring tube at the reference point was used as the reference pressure, P_{ref}, in all the experiments, while the output of the ring tube at the diffuser entry was the inlet static pressure, P_1. The static pressure P_1 was used as the reference inlet pressure against which pressure recovery values at points along the sidewall, $P_x - P_1$, were evaluated. A port for the insertion of a Pitot tube from the side was provided at the flow reference point for measurement of the total pressure, and hence the average entry axial velocity, U_1 was established.

On each side plate of the inlet duct-diffuser assembly, 60 static pressure holes, and hence a total of 120 static pressure holes for both sides, were drilled. Attention was paid to the locations near the entry of the diffuser where very high pressure gradients due to separation and a sudden change of flow geometry were expected. In this region, the static pressure holes were spaced at 5 mm intervals. This spacing increased progressively to 10, 20 and 50 mm in the downstream direction.

The roof of the diffuser was constructed from six, 150 mm wide strips and two strips of widths 15 mm and 85 mm installed as the first and last strip at the entry and exit of the diffuser respectively. All the strips were made from 10 mm Perspex and were reinforced with 25 mm aluminium square tubes. A 150 mm wide Perspex probing strip was used to move a 3-tube yaw meter in the y direction for measuring the velocity profiles across the diffuser test section. This strip was 2 m long, 10 mm thick and was also reinforced with a 25 mm aluminium square tube. All the 150 mm wide roof strips were removed in turn and replaced with the probing strip. Velocity profiles were then measured at 150 mm intervals in the x direction along the axial length of the diffuser.

All the pressures were measured using a 0-625 Pa full range reluctance differential pressure transducer whose voltage signals were conditioned and calibrated at manufacture to give a linearized 0-5 Vdc output at an accuracy of $\pm 1.5\%$ full scale. The low-pressure port of the transducer was connected to the averaged static pressure tapping at the flow reference point,

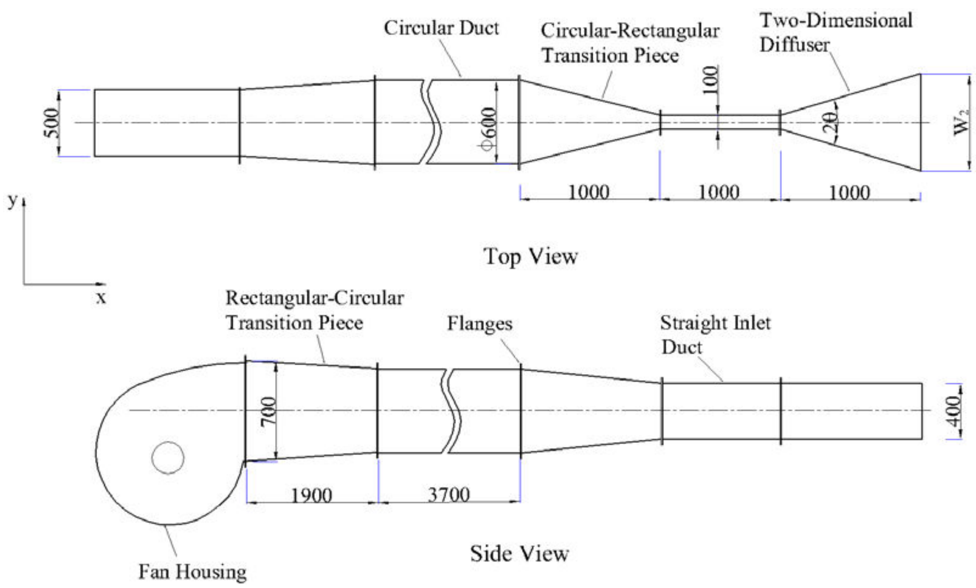

Fig. 2. Experimental apparatus

while the high-pressure port was connected to the output of a Scanivalve pressure scanner. Data was sampled at a frequency of 100 Hz.

A Scanivalve pressure scanner was used to scan the static pressures on the diffuser side walls, and the total pressure at the reference point. The coefficients of pressure recovery C_p and the overall pressure recovery coefficient C_{pr}, were evaluated according to the definition given in Eqn.1

$$C_p = \frac{P_x - P_1}{\frac{1}{2}\rho U_1^2} \tag{1}$$

Probing was done using a 3-tube yaw meter calibrated according to the established chart method of method of Yajnik and Gupta [11], Gundogdu and Carpinlioglu [12] and Rhagava et al.[13]. Results obtained by use of the generated calibration charts were tested at several locations within the diffuser test section against the null-reading technique. The average error in measuring the flow angle was determined as $0.81°$ while the rms error in velocity measurement was 1.37%.

The voltage signals obtained from the reluctance transducer and the hot wire anemometer were digitized through a plug-in 32-bit PCI data acquisition board (DAQ). For each diffuser, the influence of Reynolds number on the flow-field, the influence of change of the wall to which the flow becomes attached after stalling, and the two-dimensionality of the flow were investigated.

III. Results and Discussion

A. Tunnel qualification

1) Inlet channel flow: Uniformity of inlet flow affects the diffuser performance [9]. In this respect, the inlet axial velocity profiles were measured, first without the diffuser and then repeated with the diffuser connected to the inlet duct.

A measure of the non-uniformity of the flow, λ, was evaluated as the overall discrepancy of axial velocities u_i, measured at corresponding points to the right and left of the x-axis. A summary of the non-uniformities for all the diffusers with flow at different Reynolds numbers is given in Table I.

TABLE I
NON-UNIFORMITY, λ (%), OF INLET FLOW WITH DIFFUSER

$Re \times 10^5$	**Re=1.08**	**Re=1.63**	**Re=2.18**
30°	4.17	4.41	4.92
40°	4.21	4.42	4.77
50°	4.23	4.55	5.03

This non-uniformity of the flow caused the corresponding static pressures on the opposite side walls at the inlet to be different. This situation presented a problem in the interpretation of reference inlet conditions both for the velocity and static pressure data, since each flow condition produced different profiles. Consequently, it was found necessary to reference the inlet flow conditions to a location where the flow was reasonably uniform. Such a point was found to be at an arbitrarily chosen upstream location of $\frac{x}{W_1} = -2.35$. At this inlet reference location, the axial velocity profiles were again measured, with and without the diffuser. The highest non-uniformity of the axial velocity profiles measured in both the y and z directions at the reference diffuser location was 0.23% for the $50°$ diffuser.

It still remains unclear why stalling in the two-dimensional fully stalled regime in diffusers occur on a particular wall. Great care was taken to manufacture highly symmetrical diffusers. Tests to rule out the possibility of a bias towards the flow attaching to a particular wall whenever the fan was switched on were carried out. In this regard, for all diffusers and before any measurements were taken, a start-stop check was done whereby the fan was started, the wall to which the flow was attached noted, the fan switched off and started again and the process repeated. These tests were done at an inlet duct velocity of 15 m/s. An intermittency parameter γ_s, was then defined as a ratio of the number of times the flow remained attached to a given wall to the total number of start-stop cycles. γ_s was measured to have a maximum value of 0.60 in favor of the left hand wall for the 50° diffuser.

It was however possible to force the flow to attach to the other wall by partially blocking the flow at the inlet with a piece of Perspex, and directing it to the desired wall where the data acquisition system was positioned. Once 'switched' to the other side, the flow remained attached to that wall.

2) Reproducibility and replication: Flows in wide angled diffusers are inherently unsteady. In order to reduce the effect of the unsteadiness in the experimental results, the pressure transducer differential voltages, were averaged over long durations. The optimal averaging duration was achieved by setting it at a given value and then measuring the velocities and static pressures at a few selected data points while holding the inlet velocity constant. The averaging period was then varied and the measurements at the same data points repeated. Thus, the optimal averaging duration corresponded to the period beyond which the velocity and static pressure readings ceased changing with change in averaging period. This optimal averaging time was kept constant for the rest of the measurements and it was only then that the repeatability tests to determine the overall experimental uncertainties were performed. Due to the large number of data points obtained especially when measuring the velocity profiles, it was found adequate to take readings at each point three times and then average their discrepancies for all the data points. In any case, the statement of overall uncertainties is all that is required. The results for the reproducibility tests are summarized in Table II. Even though the reproducibility seems to worsen as both Reynolds number and divergence angles increased, the maximum discrepancy at 1.88% indicate fairly reproducible experiments of these physically complex flows.

As has already been observed, the wall to which the flow attached was quite random. However, since the data acquisition system was placed on the left side of the diffuser, the flow was always manually 'switched' to the left wall in instances when it attached to the right wall. With this in mind, it was important to confirm that both the pressure and velocity fields were independent of the wall to which flow was attached. To verify this requirement, the flow field was measured with the flow firstly attached to the left wall and secondly with the flow attached to the right wall. The discrepancies of the velocities at corresponding points when the flow was attached to either left or right wall could be established.

The static pressures were measured on the wall to which the flow was attached (unstalled wall). The discrepancies of pressure recovery data for corresponding static pressure holes on the two walls could then be established and are summarized in Table II.

The static pressure field showed more sensitivity to replication than the velocity profiles. The replication discrepancies for the static pressure field worsened as the flow velocity and divergence angle were increased. However, since the replication discrepancies are of the same order of magnitude as the reproducibility, they can be viewed to be a result of the flow unsteadiness and the small imperfections in the diffuser symmetry. The replication discrepancy of the velocity profiles was below 0.6% in all cases.

TABLE II
REPRODUCIBILITY AND REPLICATION OF DIFFUSER FLOWS

		Reproducibility discrepancy,%			Replication discrepancy,%		
		Re$\times 10^5$			Re$\times 10^5$		
	2θ	1.08	1.63	2.18	1.08	1.63	2.18
C_{pr}	30°	1.42	1.57	1.63	1.77	1.85	1.97
	40°	1.53	1.61	1.70	1.91	1.97	2.01
	50°	1.66	1.72	1.88	1.93	1.99	2.13
$\frac{u}{U_1}$	30°	0.32	0.36	0.37	0.52	0.55	0.57
	40°	0.34	0.33	0.39	0.53	0.56	0.58
	50°	0.36	0.37	0.34	0.56	0.58	0.59

3) Reynolds number dependence: Although it is reported in the literature that variation of Reynolds number does not influence the pressure recovery data, the indications obtained while performing the reproducibility and replication tests, was that Reynolds number indeed has a significant influence on the flow field. It was therefore decided to perform tests to evaluate the influence of Reynolds number on both the static pressure and velocity fields. The inlet velocity was set at 10 m/s, 15 m/s and 20 m/s corresponding to Reynolds numbers 1.08×10^5, 1.63×10^5, 2.18×10^5, respectively. Preliminary tests showed that at velocities higher than 20 m/s, undesirable vibration of the diffusers was produced.

Results show that by changing the velocity from 10 m/s to 20 m/s, the static pressure recovery increased by 8.31%, 10.15%, and 9.35% for 30°, 40° and 50° diffusers, respectively.A similar increase in Reynolds number for the velocity profiles showed that the increase of the normalized velocities, was 1.37%, 1.57%, and 1.60% for 30°, 40° and 50° diffusers, respectively. Evidently, the static pressure was influenced by the Reynolds number at values that are outside the overall experimental uncertainties of about 2% in this research.

Results from the limited number of flow cases in this study can not be considered adequate to provide a reasonable correlation. Bearing this in mind and in order to focus on the primary objectives of this research, it was decided that from this point onwards, the Reynolds number be held constant at 1.63×10^5 corresponding to an inlet duct velocity of 15 m/s.

4) Two-dimensionality: The primary assumption made in this study is that the flow is two dimensional. In fact, it is only due to this assumption that the three-tube yaw meter was used to measure the velocity vectors. All diffusers studied in this research had the roof and the floor walls running parallel to each other. It was rational to assume that the boundary layer growth rates from the floor and roof walls were the same and merged at the mid-plane. Therefore, measuring the flow in the mid-plane was adequately representative of a two-dimensional flow.

Traditionally, proof of two-dimensionality of a flow is carried out by measuring axial velocity profiles at two different planes, one below the mid-plane, and another above the mid-plane. Thus, for a given location within the diffuser, axial velocity profiles were measured in three planes, namely; upper, mid and lower planes. The upper and lower planes were both at a 100 mm distance from the roof and floor of the diffuser respectively. For all diffusers in this study, the two-dimensionality of the flow was verified using this procedure, with the velocity profiles being measured at three locations downstream of the diffuser inlet located at $\frac{x}{W_1} = 0.9$, 3.9 and 8.4. With the mid-plane held as the datum, the overall deviations from this plane for the lower and the upper planes produced a maximum deviation of 0.3% meaning that the two-dimensionality was well within the uncertainties of the experimental data in this research.

Flow visualization using woolen tufts was performed in order to observe the steadiness and two-dimensionality of the flow. The woolen tufts were attached to the side walls of the diffuser at several locations. Apart from the tufts that were next to the roof and floor, and which displayed slight fluctuations of movement, all the other tufts faced the downstream direction steadily and ran almost parallel to each other.

IV. Data bank contribution

In the interest of clarity while discussing the experimental results, only representative cases of extensive experimental data have been presented. However, the same rigorous checks and experimental procedures, as discussed in section III-A, were performed for all flow cases. These results are presented in the appendix as a data bank contribution. The results presented in the data bank include, the axial velocity profiles, $\frac{u}{U_1}$, lateral velocity profiles, $\frac{v}{U_1}$, and the wall static pressure recovery and static pressure fields, C_p.

V. Conclusion

In this study, experimental investigations of separated flows in fully stalled wide-angled diffusers have been carried out. Due to the adverse pressure gradient along the diffuser walls, flow separated from one diverging wall and became attached to the other wall, thus forming a region of steady stall within the diffusers. It was not possible to determine in advance, the wall to which the flow would attach. Tests to determine the wall to which the flow remains attached led to the conclusion that the wall of preference was totally random and was probably caused by a slight upstream disturbance that was impossible to detect. However, it was possible to 'switch' the flow from one wall to the other by introducing an inlet disturbance. It was found that once 'switched' to a wall, the flow remained attached to that wall permanently. Experimental results showed that regardless of the wall to which the flow was attached, both the velocity and pressure flow-fields were replicated with discrepancies below 2%.

Although current literature states that for a given geometry, the Reynolds number has little influence on the static pressure recovery, it was found in this study that by increasing the velocity from 10 m/s to 20 m/s, the static pressure recovery for the 30° diffuser increased by 8.31%. However, as the divergence angle was increased, a similar increase in Reynolds number resulted in a higher percentage of pressure recovery. The limited range of Reynolds numbers investigated in this study could not allow a rational correlation between the Reynolds number and the C_{pr} profiles. This range was limited by the physical constraints imposed by the wind tunnel and fan speed. For instance, a change of velocity from 10 m/s to 80 m/s would not change the Reynolds number by even one order of magnitude.

The experimental uncertainties in this research were approximately 2%. Within these uncertainties a reliable data bank contribution has been provided for unvaned fully stalled wide-angled diffusers. The parameters in the data bank include the wall static pressure recovery data, axial and lateral velocity profiles and static pressure profiles. Velocity profiles that are plotted on the proposed normalized length scale have also been included in the data bank.

References

[1] R. Fox and S. Kline, "Flow regime data and design methods for curved subsonic diffusers," *Transactions of the ASME, Journal of Basic Engineering*, vol. 84, pp. 303–312, 1962.

[2] L. Reneau, J. Johnston, and S. Kline, "Performance and design of straight, two-dimensional diffusers," *Transactions of the ASME, Journal of Basic Engineering*, vol. 89, no. 1, pp. 141–150, 1967.

[3] S. Kline, D. Abbort, and R. Fox, "Optimum design of straight-walled diffusers," *Transactions of the ASME, Journal of Basic Engineering*, vol. 81, pp. 321–331, 1959.

[4] C. Sagi and J. Johnston, "The design and performance of two-dimensional, curved diffusers: Part I - Exposition of method," *Transactions of the ASME, Journal of Basic Engineering*, vol. 89, pp. 715–731, 1967.

[5] O. McMillan and J. Johnston, "Performance of low-aspect ratio diffusers with fully developed turbulent inlet flows - Some experimetal results," *Transactions of the ASME, Journal of Fluids Engineering*, vol. 95, no. 3, pp. 385–392, 1973.

[6] J. Norbury, "Some measurements of boundary-layer growth in a two-dimensional diffuser," *Transactions of the ASME, Journal of Basic Engineering*, vol. 81, pp. 285–296, 1959.

[7] J. Johnston and C. Powars, "Some effects of inlet blockage and aspect ratio on diffuser performance," *Transactions of the ASME, Journal of Basic Engineering*, vol. 91, pp. 551–553, 1969.

[8] J. Hoffmann, "Effects of free-stream turbulence on diffuser perfromance," *Transactions of the ASME, Journal of Fluids Engineering*, vol. 103, no. 3, pp. 385–390, 1981.

[9] S. Wolf and J. Johnston, "Effects of non-uniform inlet velocity profiles on flow regimes and performance in two-dimensional diffusers," *Transactions of the ASME, Journal of Basic Engineering*, vol. 91, no. 3, pp. 462–474, 1969.

[10] K. Kaiser and A. McDonald, "Effect of wake-type non-uniform inlet velocity profiles on first appreciable stall in plane-wall diffusers," *Transactions of the ASME, Journal of Fluids Engineering*, vol. 102, no. 3, pp. 283–289, 1980.

[11] K. Yajnik and R. Gupta, "A new probe for measurement of velocity and flow direction in separated flows," *Journal of Physics, Series E, Science Instrumentation*, vol. 6, no. 82-86, 1973.

[12] M. Gundogdu and M. Carpinlioglu, "A multi-tube pressure probe calibration method for measurements of mean flow parameters in swirling flows," *Flow Measurement and Instrumentation*, vol. 9, pp. 243–248, 1998.

[13] A. Rhagava, K. Kumar, R. Malhotra, and D. Agrawal, "A probe for the measurement of velocity field," *Transactions of the ASME, Journal of Fluids Engineering*, vol. 11, no. 1, pp. 143–146, 1979.

ACKNOWLEDGMENT

The authors would like to thank the University Science, Humanities and Engineering Partnerships in Africa, (USHEPiA) for funding the project. The would further would like to thank the Department of Mechanical Engineering, University of Cape Town for the use of their facilities.

Dr. Eng. Karanja Kibicho is a lecturer of many years in the area of Thermo-Fluids at the Jomo Kenyatta university of Agriculture and Technology. He has authored numerous journal papers and articles. He is currently the Head of Department of Mechanical Engineering in Jomo Kenyatta University. He is a member of the Institution of Engineers of Kenya and registered by the Kenya Engineers Registration board.

Prof. Anthony Sayers is an Associate Professor in the Department of Mechanical Engineering at the University of Cape Town. He has authored numerous journal papers and articles. He is also an author of several text books in Fluid Mechanics and Turbo-Machines. He is a member of the South Africa Institution of Mechanical Engineers among other affiliations to professional bodies.

APPENDIX: DATA BANK CONTRIBUTION

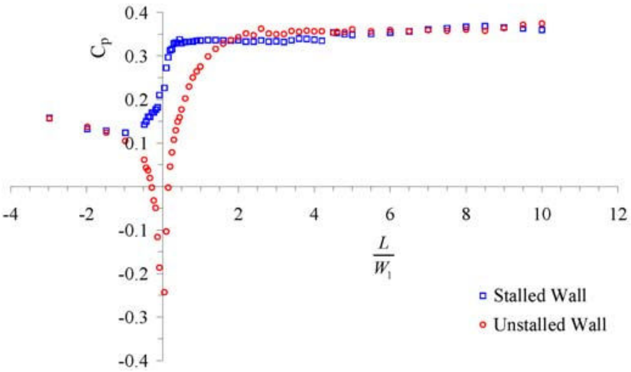

Fig. 3. Coefficient of static pressure for the 30° diffuser

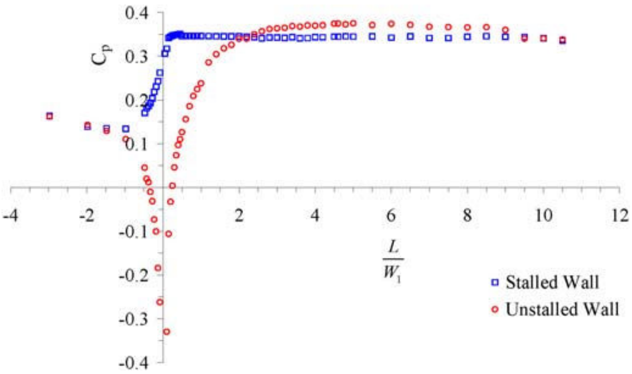

Fig. 4. Coefficient of static pressure for the 40° diffuser

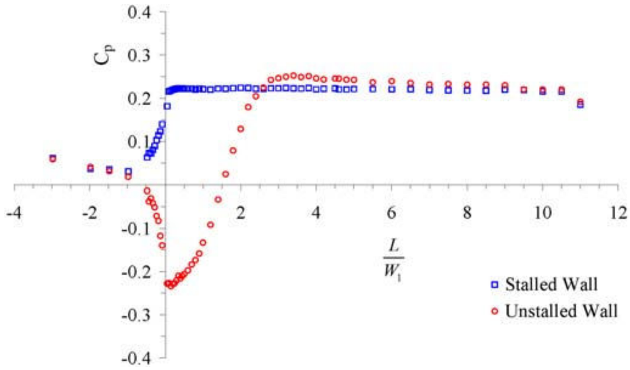

Fig. 5. Coefficient of static pressure for the 50° diffuser

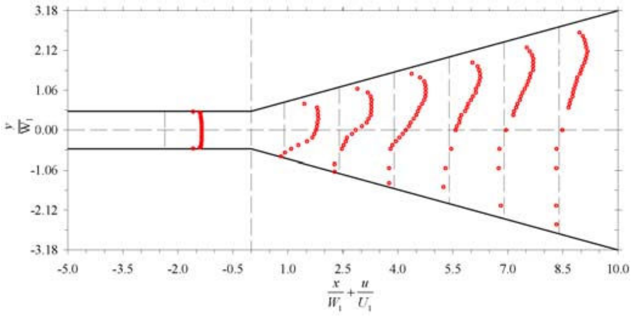

Fig. 6. Axial velocity profiles for the 30° diffuser

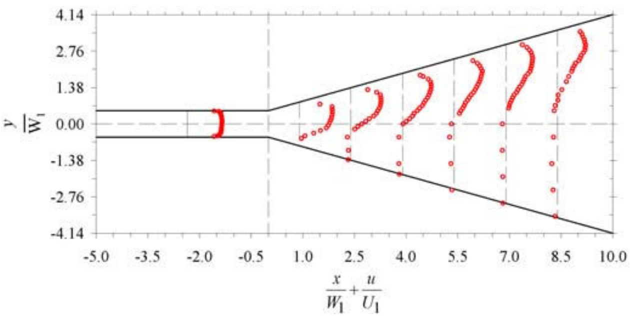

Fig. 7. Axial velocity profiles for the 40° diffuser

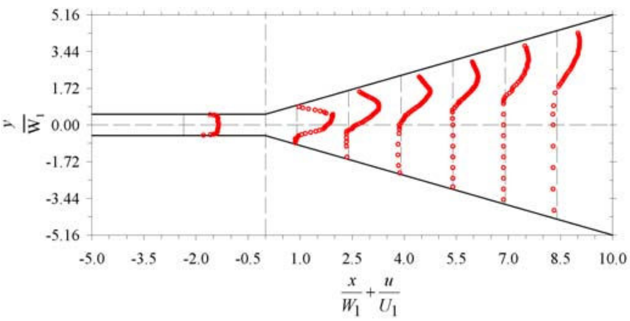

Fig. 8. Axial velocity profiles for the 50° diffuser