Estimation of Pressure Loss Coefficient for Nozzle Diffuser Elements of Valveless Micro Pump used in Electronics Cooling Applications

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***Abstract*— Valveless micro pump is a device used to transmit low flow rate in the absence of moving parts (except the pump diaphragm). Some of the applications of valveless micro pump are chemical analysis systems (ex: fluid injection analysis &electrophoresis systems), micro dosage systems, thermal aspects systems, biochemical industries. The main advantage of valve-less micro pump is that it reduces the mechanical failure and also reduces the clogging & sedimentation problems. Computer mainframes, telecommunication equipment's, super computers & high powered systems will require improved cooling that is not possible from traditional air cooling or direct immersion cooling technologies but this type of cooling is possible, by using valve-less micro pump.**

In This project work flow analysis studies are carried out for conical and planar Nozzle-Diffuser element in valveless micro pump.

The main objective of this work is the, analyse the performance of conical and planar nozzle / diffuser element used in valve-less micro pump by varying the Reynolds number 200,500 and 100.And also varying the cone angle from 5º to 75º at an interval value of 50. This work consisting, CFD (FLUENT) simulation of four different types of diffuser flows: fully developed and thin inlet boundary layer flows through conical and planar diffusers used in valve-less micro pump model by using ANSYS WORKBENCH. More ever the CFD simulations are performed to study the fluid flow inside the designed nozzle / diffuser elements as well as the performance of the designed nozzle / diffuser elements. Then compare the simulated results of pressure & velocity to literature data, improving the accuracy of the implemented model.

Keywords: ANSYS CFD, Fluid flow analysis, Conical and Planar Nozzle/Diffuser element, Divergence angle, Reynolds number.

1.INTRODUCTION

Micro pump is a device used to transmit low flow rate. Research on micro pumps was initiated in 1980 and numerous different pumps have since been developed. They can be manufactured in different materials, but mostly silicon and glass have been used as bulk materials. During the last years plastic has been shown to be a competitive alternative. Different pump principles are conceivable. They can generally be classified into two groups: mechanical and non-mechanical (without moving parts). At least three kinds of mechanical micro pumps have been developed: peristaltic, reciprocating and rotary pumps [8].

1.1 Performance of micro pump depends on following characteristics:

- Flow rate
- Pressure head
- Scaling potential, size etc.

1.2 Applications of Micro Pump:

- Chemical analysis systems (ex: fluid injection analysis & electrophoresis systems)
- micro dosage systems
- thermal aspects systems
- biochemical industries
- ink jet printers
- Electronic cooling systems

1.3 Principle of operation

The operating principle of a valveless micro pump is illustrated in Fig. 1. The particular flow characteristics shown are for small nozzle-diffuser angles. In the expansion mode, as the volume of the pumping chamber increases, more fluid enters the pumping chamber from the element on the right which acts like a diffuser (and hence offers less flow resistance) than the element on the left, which acts like a nozzle.

On the other hand, in the contraction mode, more fluid goes out of the element on the left which now acts as adiffuser, while the element on the right acts as a nozzle. Hence, net fluid transport is achieved in the pumping chamber from right to left.

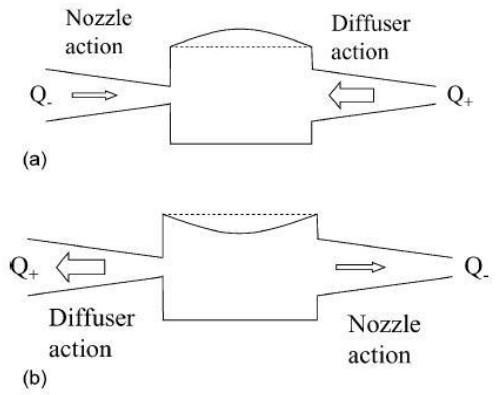


Fig. 1. Flow rectification in a valveless micro pump: (a) expansion mode (increasing volume of the pumping chamber) and (b) contraction mode (decreasing volume of the pumping chamber). The thicker arrows imply higher volume flow rates.

1.4 Nozzle-diffuser elements:

The volume flow rate of a valveless micro pump depends on the rectification efficiency of the pump among other factors (such as amplitude and frequency of operation of the diaphragm). The rectification efficiency, ε, is the ratio of the volume of net fluid pumped to that crossing (entering or leaving) the pump in a given interval of time ($\varepsilon = (Q+ - Q-)/(Q+ - Q-)$, see Fig. 1). The rectification efficiency of nozzle-diffuser micro pumps reported in the literature is very low, generally between 0.01 and 0.2. Since the rectification efficiency of these micro pumps depends on the flow directing ability of the nozzle-diffuser elements, many studies have been directed at better understanding the fluid dynamic behavior and the flow rectification properties of nozzle-diffuser elements [20]. Different shapes of nozzle-diffuser elements have been considered in the literature. They can be broadly classified as spatial and planar. Spatial diffusers can be further divided into conical and pyramidal. These diffusers are schematically shown in Fig. 2.

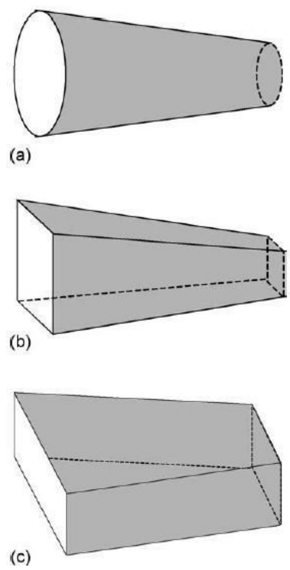


Fig. 2. Schematic of (a) conical, (b) pyramidal, and (c) planar Nozzle-diffuser elements.

Anders Olsson et al. [2] have used CFD software to analyze the performance of nozzle-diffuser elements used in valve-less diffuser pump and nozzle elements used in dynamic micro pumps. The results are compared with measurements and with analytical expressions based on empirical results known from basic fluid mechanics .They have carried out CFD simulation to investigate the working principles of the diffuser element in the valve-less diffuser and nozzle elements used in dynamic micro pumps.

Anders Olsson *et al.* [3] have used commercial Computational Fluid Dynamics program ANSYS/Flotran (version 5.3) to simulate the flow pressure characteristic of several diffuser elements. The simulations are compared with experimental results. It is found that the simulated flow-pressure characteristic agrees well with the measured in the converging-wall direction and for Reynolds number below 300-400 in the diverging-wall direction. For higher Reynolds numbers the pressure loss in the diverging-wall direction is underestimated.

Anders Olsson, [8] he studied first micro machined versions of pumps based on the new valve-less diffuser pump principle and also he did theoretical study for the same.

Nikhil Ramaswamy *et al.* [14] have used a SIMULINK model for the simulation of the valve-less micro pump is developed. In this model the operating parameters namely voltage, diaphragm diameter and thickness are considered for simulation. To optimize the pump performance, three commonly used materials are considered for diaphragm and their performance for different diameters and thickness's is studied. Results obtained through the developed model compare well with earlier results. The volumetric discharge versus pressure difference is used for characterizing the pump performance.

From the review of the literature above, it is clear that fluid flow through nozzle-diffuser elements for micro pumps is not well understood. A number of conflicting results regarding the nature of the flow (laminar or turbulent) and variation of diffuser efficiency with cone angle and diffuser length have been reported.

Clearly, there is a need to better understand the flow behavior through nozzle-diffuser elements at low Reynolds numbers. The present study addresses this need.

2 THEORETICAL ANALYSIS:

2.1 Pressure loss coefficient

The pressure loss coefficient for flows through a gradually contracting nozzle, a gradually expanding diffuser, or a sudden expansion or contraction in an internal flow system is defined as the ratio of pressure drop across the device to the velocity head upstream of the device

$$K = \frac{\Delta p}{\rho v^2/2} \qquad \text{.....2.1}$$

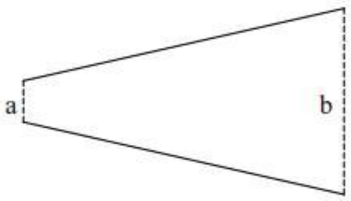


Fig.3 Schematic of a nozzle-diffuser element.

For flow through a gradually expanding diffuser (Fig. 3) or a gradually contracting nozzle, the pressure loss coefficient can be calculated as follows. For flow in the diffuser direction (from cross-section a to b in Fig. 3), the incompressible steady-flow energy equation reduces to

$$p_a + \tfrac{1}{2}\rho v_a^2 = p_b + \tfrac{1}{2}\rho v_b^2 + \Delta p_d \qquad \text{.....2.2}$$

Hence, the pressure loss coefficient can be written as

$$K_d = 1 - \frac{d_a^2}{d_b^2} - C_p$$

Hence for a given diffuser geometry, the pressure loss Introducing the pressure recovery coefficient

$$C_p = \frac{P_2 - P_1}{0.5\rho V^2{}_1}$$

*K*d for spatial diffusers (e.g. conical and pyramidal) can be written as

$$K_d = 1 - \frac{d_a^4}{d_b^4} - C_p \qquad \text{.....2.5}$$

While for planar diffusers,

coefficient can be calculated from the pressure drop and the mean velocity at the neck. Similarly, for flow in the nozzle direction (from cross-section b to a in Fig. 3), the pressure loss coefficient is given by

$$K_n = \frac{\Delta p_n}{\rho v_b^2/2} \qquad \text{.....2.7}$$

2.2 Diffuser efficiency:

The diffuser efficiency of a nozzle-diffuser element is defined as the ratio of the total pressure loss coefficient for flow in the nozzle direction to that for the flow in the diffuser direction

$$\eta = \frac{K_{n,t}}{K_{d,t}} \qquad \text{.....2.8}$$

Hence, $\eta > 1$ will cause a pumping action in the diffuser direction (Fig. 1) in a valveless micro pump, while $\eta < 1$ will lead to pumping action in the nozzle direction. The case where

η **=** 1 corresponds to equal pressure drops in both the nozzle

and the diffuser directions, leading to no flow rectification. In Eq. (2.8), the total pressure loss coefficients for both the diffuser and nozzle directions can be divided into three parts:

(i) losses due to sudden contraction at the entrance, (ii) losses due to gradual contraction or expansion through the length of the nozzle-diffuser, and (iii) losses due to sudden expansion at the exit. The total pressure drop in the diffuser direction can thus be written as

$$\Delta p_{d,t} = \Delta p_{d,en} + \Delta p_d + \Delta p_{d,ex}$$

Therefore, diffuser efficiency can be written as 2.9

$$\eta = \frac{K_{n,t}}{K_{d,t}} = \frac{(K_{n,en} + K_n)(A_a^2/A_b^2) + K_{n,ex}}{K_{d,en} + K_d + K_{d,ex}(A_a^2/A_b^2)} \qquad \text{.....2.10}$$

3. PROPOSED STUDY:

The main objective of the present project work is to examine the fluid Flow behaviour and performance in the conical and planar Nozzle/diffuser element of a valveless micro pump.

The main objective can be subdivided as follows:

- To understand the governing factors and physics which are associated with the nozzle/diffuser element used in piezoelectric valve-less micro pump.
- To analyze the loss co-efficient of conical and planar

.....2.3

Nozzle/diffuser element at Re-200, 500 and 1000 and also varying the diverging angle from 5^o to 75^o.

- To analyze the efficiency of 5^o and 70^o conical Nozzle/diffuser element at Re-200, 500 and 1000.
- To analyze the flow rectification behavior of Nozzle/diffuser element of angle 5^o and 70^o at Re-200, 500 and 1000.
- To compare the values obtained from the numerical simulation with the values obtained from the manual calculation.

By analysing the results, a best design (dimensions of nozzle and diffuser)) is to be select to implement in valveless micro pump for best application.

The present work is useful to understand the nozzle and diffuser performance for varies cone angles and how cone angle effects on pressure and velocity of fluid in the nozzle and diffuser in valve-less micro pump.

4. Methodology:

The project work follows the below mentioned stages to meet the objective:

Stage 1: Collection of Theoretical data's from literature survey.

Stage 2: Manual calculation of velocities, mass flow rate and volume flow rate for low Reynolds numbers.

Stage3: Computational fluid flow analysis of nozzle and diffuser (conical and planar) using ANSYS-14(fluent)

It consists of:

i. Modeling
ii. Meshing
iii. Solving (pre-processing)
iv. Output Result (post-processing)

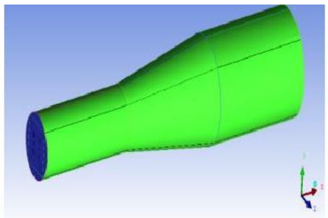
Stage 4: Tabulation of results
Stage 5: Discussion on results
Stage 6: Selection of best design
Stage 7: Conclusion

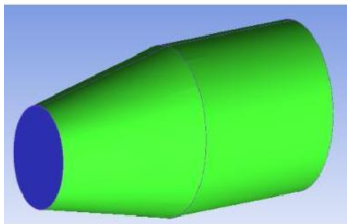
Note: Flow characteristics for low Reynolds number flow in a nozzle (as opposed to a diffuser) are not expected to be significantly different from those at the higher Reynolds numbers hence, nozzle flow is not considered in this work, and the focus is restricted to diffuser flow.

The problem of determining the pressure drop and the average neck velocity in a diffuser is solved numerically using the finite volume method. The commercially available software package Fluent, is used to model and solve the problem. Conical and planar diffusers are considered; the effects of both sharp and smooth entrance conditions are studied. Smooth edges will cause the flow entering the diffuser to be relatively fully developed, while sharp edges will lead to thin boundary layers at the inlet cross-section. Steady-state laminar flow simulations are carried out for Reynolds numbers of 200, 500 and 1000.

The particular geometries modeled for the four cases considered fully developed and thin boundary layer inlet flow for the conical and planar diffusers are shown in Fig. 4.

Case (1)





Case (2)

Fig. 4 Geometries modelled to simulate (1) fully developed, (2) thin inlet boundary layer inlet flow in a conical diffuser.

Meshing is carried out by ANSYS-ICEM CFD.

The flow is considered incompressible. Uniform velocity inlet and uniform pressure outlet boundary conditions are applied at cross-sections 1 and 4, respectively, for all the four cases. In addition, no-slip boundary conditions are imposed at the walls.

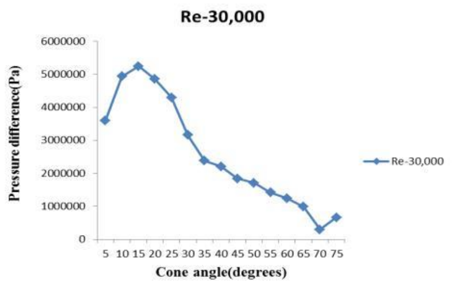
5. Model validation

The numerical model was validated by simulating turbulent flow at high Reynolds numbers in conical diffusers for fully developed and thin inlet boundary layer flows. The two equation κ–ε model in Fluent was used to model the turbulent flow for a Reynolds number of 30,000 the numerical values of the predicted pressure loss coefficients are compared to the experimental values available for turbulent flows in the literature. The comparison is shown in Table1. The experimental data for a thin inlet boundary layer are from [17] and those for a fully developed boundary layer are from [20]. The experimental values are linearly interpolated between data read from charts. While there is reasonable agreement between the predicted and experimental results, the predictions are lower in general, especially at the larger angles. This may be attributed to inadequate handling of separation by the κ–ε model and additional pressure losses in the experimental set-up due to wall friction and roughness effects which are not accounted for in the predictions.

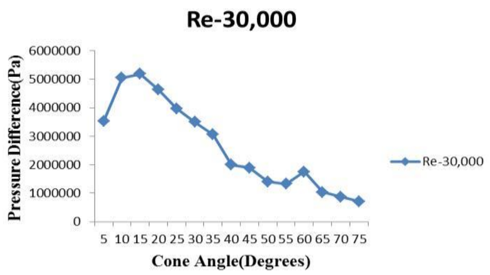
Table1 Model validation Table of Conical diffuser for Re30, 000

Angle(^{O})	Inlet Boundary layer	Predicted Value	Experimental Data from literature survey
5	Fully developed	0.084	0.06
70	Fully developed	0.960	1.20
5	Thin	0.10	0.13
70	Thin	0.885	1.02

6. RESULTS AND DISCUSSION:

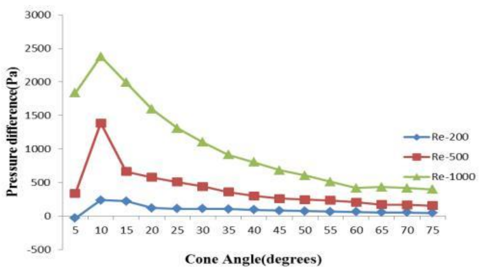


(a)

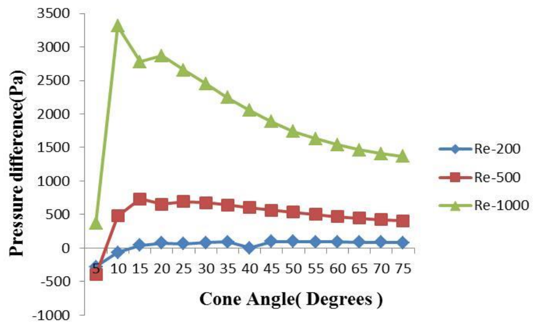


(b)

Fig.6.1 (a) and (b) shows a Pressure difference (p_2-p_1) curve for fully developed inlet and thin inlet boundary layer flows in a conical diffuser at Re-30,000(turbulent flow). It can be observed that, static pressure difference increases from 5^{o} to15^{o}, and decreases for the cone angle more than 15^{o} due to increase in channel area.

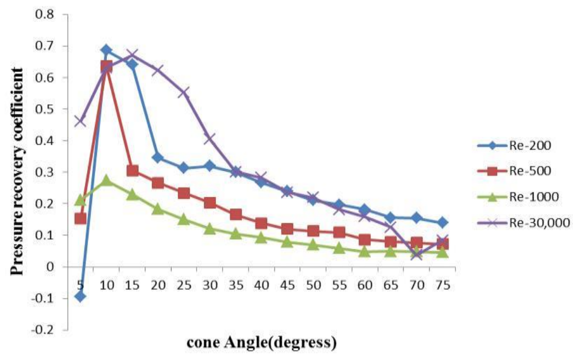


(a)

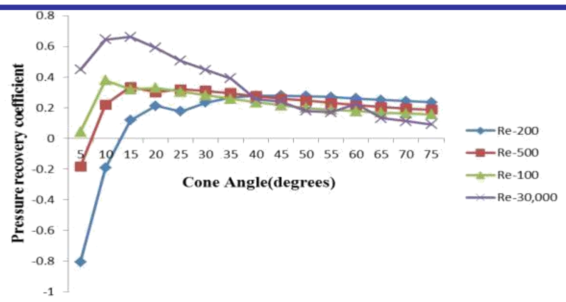


(b)

Fig.6.2 (a) and (b) shows a Pressure difference (p_2-p_1) curve for fully developed inlet and thin inlet boundary layer flows in a conical diffuser at Re-200, 500, 1000 (laminar flow). It can be observed from fig.(a), static pressure increases from 5^{o} to10 ^{o}, and decreases for the cone angle more than 15^{o} due increase in channel area and from fig.(b) it can be observed that static pressure increases from 5^{o} to15^{o}, and decreases for the cone angle more than 20^{o} at Re=500,1000 and for Re=200 static pressure increases from 5^{o} to10^{o}, and decreases for the cone angle more than 15^{o} due to change in flow.



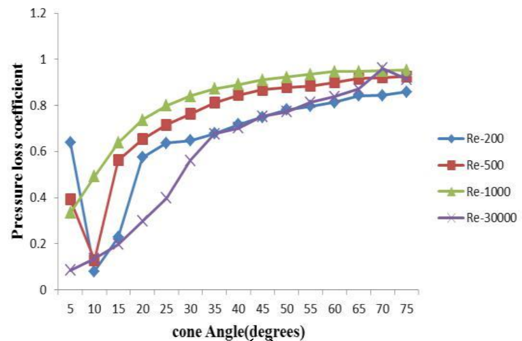
(a)



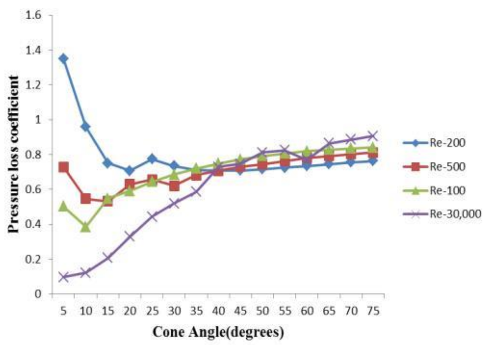
(b)

Fig.6.3 (a) and (b) shows the pressure recovery coefficient curve for fully developed inlet and thin inlet boundary layer flows in a conical diffuser at Re-200, 500, 1000(laminar flow), 30,000(turbulent flow). It can be observed that, for laminar flows pressure recovery coefficient increases from 5^o to10^o, and decreases for the cone angle more than 10^o and for turbulent flow, it can be observed that, pressure recovery coefficient increases from 5^o to15^o, and decreases for the cone angle more than 15^o due to increase in channel area.

And also from Fig.5.3 (a) and (b) high pressure recovery coefficient exit in the fully developed inlet boundary layer flows in a conical diffuser than compare to the thin inlet boundary layer flows in a conical diffuser, due to change in flow.



(a)



(b)

Pressure loss coefficients (K_d) for fully developed and thin inlet boundary layer flows in a conical diffuser are plotted in Fig. 6.4(a) and(b), respectively, as a function of cone full-angle for Reynolds numbers (Re) of 200, 500 and 1000 and for high Reynolds number turbulent flow(Re=30,000).

The pressure loss coefficients at low Reynolds number were calculated using Eq. (1.21) in combination with the pressure drop and the average neck velocity obtained from the numerical simulations.

Moreover, at the low Reynolds numbers considered, K_d for a given diffuser angle varies significantly with Reynolds number, especially at small cone angles. In contrast, K_d for high Reynolds number flows does not vary with Reynolds number.

For fully developed inlet boundary layer flow, K_d is the lowest for Re = 1000 and the highest for Re = 200, when the cone full-angle is less than approximately 10°.

For larger full-angles, the opposite is true, i.e. K_d is lowest for Re =200 and highest for Re = 1000.

For small cone angles, the loss coefficients decrease with increasing Re, while at large cone angles, they increase with increasing Re.

The high loss coefficients for small diffuser angles at low Re are believed to be due to the dominance of viscous forces in these very ordered flows.

As the cone angle increases, flow separation occurs, which is associated with higher loss coefficients (higher than the viscous contributions). Since flow separation is more dominant for higher Re, loss coefficients at the larger cone angles are also greater for larger Re, due to change in area.

The same phenomena are also observed for the thin inlet boundary layer flow, although in this case, viscous losses seem to dominate up to cone full-angles of 5^o–10^o.

For the case of the fully developed inlet boundary layer, back flow at the outlet boundary was first observed for the cone full-angle of 15^o at Re = 500 and 1000, and for the full-angle of 20^o at Re = 200. Similarly, for the thin inlet boundary layer, back flow at the outlet boundary was first observed for the cone full-angle of 20^o at Re = 500 and 1000 and for the full-angle of 25^o at Re = 200.

The conditions under which back flow starts correspond to the cone angle beyond which the higher Reynolds number flow has the greater loss coefficients.

Comparing the numerical values of K_d for the fully developed and thin inlet boundary layer flows, it can be observed that for small cone angles, K_d is smaller for the fully developed boundary layer and vice versa (K_d is smaller for the thin inlet boundary layer flows for large cone angles). Also, this behaviour is peculiar to low Reynolds number flow.

At high Reynolds numbers, K_d for the fully developed inlet boundary layer flow is smaller than that for thin inlet boundary layers for 5^o and larger than that for 70^o angles.

7. CONCLUSION:

The following key conclusions may be drawn from the results of the present work:

1. It is found that the general trends of variation of pressure loss coefficient with diffuser angle for both fully developed and thin inlet boundary layer flows through gradually expanding diffusers are similar to that for high Reynolds number turbulent flow. However, pressure loss coefficients for low Reynolds number laminar flows are a strong function of the flow Reynolds number, especially at small diffuser angles.
2. The variation of pressure loss coefficient with Reynolds number follows opposite trends for small and large diffuser angles. Hence, the Reynolds number of the flow should be considered in the design of micro pumps employing such valves.
3. It is observed that unlike high Reynolds number flows; the pressure loss coefficients for thin inlet boundary layer flows are not always smaller than those for fully developed inlet boundary layer flows.
4. It is observed that high pressure recovery exist in fully developed and thin inlet conical diffuser at 10^o-15^o cone angle diffuser elements gives good pressure recovery in planar diffuser.

8. SCOPE FOR FUTURE WORK

The present work can be continue to analyze the effect of variation of length in nozzle diffuser element on the pressure loss coefficients and flow rectification.

By varying the thickness and cross section of planar diffuser element, further pressure loss coefficient and flow rectification characteristics can be studied.

In present work, study is limited to cone angle upto 75 degrees, further study for more than 75degree can be do.

Studies on Planar nozzle-diffuser elements can be done as they are easier to fabricate using silicon micro fabrication techniques.

The complete micro pump used in electronic cooling applications will also be analyzed for different types (design) of the nozzle-diffuser elements.

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