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INVESTIGATION OF HEAT EXCHANGER INCLINATION IN FORCED-DRAUGHT AIR-COOLED HEAT EXCHANGERS

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ABSTRACT

The purpose of this study is to determine the influence of inclining the heat exchanger relative to the fan in a forced-draught air-cooled heat exchanger. Since inclination increases plenum depth, the effect of inclination is also compared with increasing plenum depth without inclination. The experimental study shows that inclination improves thermal performance by only 0.5%, when compared with a baseline non-inclined case with a shallow plenum. Similarly, increasing plenum depth without inclination has a thermal performance benefit of approximately 1%. The numerical study shows that, as the heat exchanger is inclined, the low velocity core at the centre of the heat exchanger moves to one side.

INTRODUCTION

An air-cooled heat exchanger (ACHE) is a device used to transfer heat energy from a hot fluid to ambient air. It typically consists of an axial fan and heat exchanger, separated by an enclosed region known as a plenum chamber. This paper is concerned with forced-draught ACHEs where, as the name suggests, the fan forces air through the heat exchanger.

In some industrial forced-draught ACHEs, such as those in the generating set industry, the flow must turn through 90° after exiting the heat exchanger. This is illustrated in Figure 1, and is the motivation for investigation of inclination of the heat exchanger.

NOMENCLATURE

d	[m]	Diameter
f	[-]	Fanning friction factor
g	[m/s^2]	Acceleration due to gravity
p	[Pa]	Pressure
Q	[W]	Heat transfer rate
Re	[-]	Reynolds number
T	[K]	Temperature
x	[m]	Depth

Special characters

Δ	[-]	Differential
θ	[°]	Inclination angle
ρ	[kg/m^3]	Density

Subscripts

d	Diameter
Dh	Hydraulic diameter
fc	Fan casing
he	Heat exchanger
$henu$	Heat exchanger non-uniform
heu	Heat exchanger uniform
Lp	Louver pitch
pl	Plenum
st	Standard
sys	System
t	Test

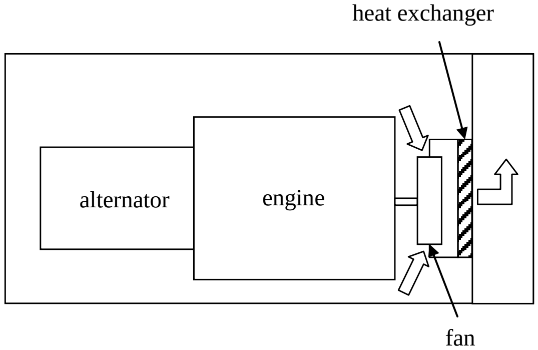

Figure 1 Cooling system in an enclosed generating set

In cooling systems such as in Figure 1, due to packaging constraints, the plenum depths are typically shallower than the critical minimum as found by Meyer and Kröger [1].

Furthermore, the axial fan often operates in the mixed-flow region of the fan characteristic, due to the restrictive nature of the system, as shown in Figure 2. These two factors lead to a reduction in the thermal performance of the system.

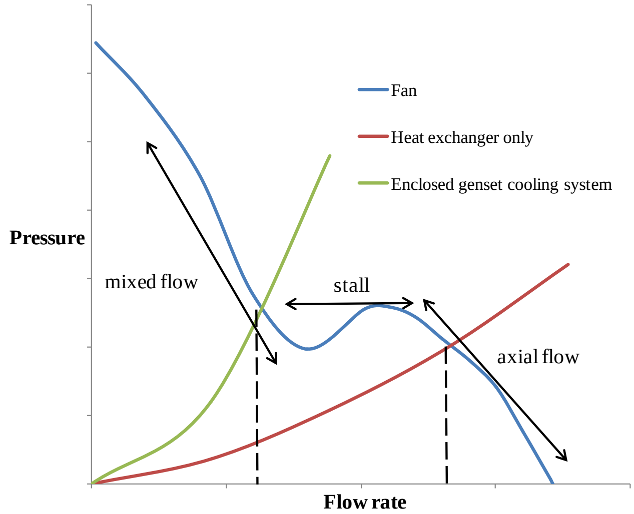

Figure 2 Operating point on fan characteristic

In this paper, the influence of inclination of the heat exchanger relative to the fan is investigated. Inclination affects the thermal performance in that it affects the flow rate through the heat exchanger, for a given fan speed and atmospheric conditions. It also affects the distribution of flow through the heat exchanger, which also impacts thermal performance.

Various studies have shown that maldistribution of flow in ACHEs affects the thermal performance by only a few percent, even with a shallow plenum [1-3]. Most studies of maldistribution of flow are focussed on profiles from ACHEs where the fan operates in the axial flow part of the fan characteristic. Other studies using artificially generated velocity profiles have found larger thermal performance reductions, some by more than 10% [4-6].

The flow rate through the system will change with inclination due to a number of factors. Firstly, the performance of the fan will change due to its interaction with the heat exchanger. This is known as the system or installation effect [7]. Secondly, it has been noted that pressure is recovered in the plenum, as some of the kinetic energy of the air exiting the fan is converted to pressure. This pressure recovery is a function of the plenum chamber depth [1], which will change when the heat exchanger is inclined. More pressure recovery leads to a greater flow rate.

Other factors will affect the flow rate through the system. One such factor is the distribution of flow through the heat exchanger. This will change with inclination, affecting the pressure drop across the heat exchanger, and hence the flow rate through the system. The final effect on flow rate is the influence of inlet losses as the flow enters the heat exchanger. The angle of incidence of the flow entering the heat exchanger will change with inclination. Flow approaching at an angle gives a region of separated flow at the entrance of the airside passages of the heat exchanger. Studies have been conducted with heat exchangers inclined in a duct with uniform inlet flow [8-10]. They have shown that inlet losses increase significantly at inclination angles of 60° or more.

The literature on inclining heat exchangers is predominantly focussed on heat exchangers inclined in a duct, with uniform inlet flow [8-13]. The main findings are that pressure losses increase with inclination, with significant increases occurring at inclination angles greater than 60°. Heat transfer coefficients in most cases are not significantly affected by inclination. To the author's knowledge, there are no studies available in the literature on inclining a single heat exchanger relative to an axial fan.

In this paper, an isothermal experimental study is conducted to investigate the effects of inclination on performance. The study also includes increasing the depth of the plenum without inclination. It is conducted across the fan characteristic, to include mixed-flow operating points. The experimental study is complemented by a numerical study using CFD to gain an understanding of the flow patterns in the plenum and how they change with inclination angle.

EXPERIMENTAL STUDY

An isothermal experimental study was conducted in order to determine the effects of inclination on ACHE performance. The means of inclination is illustrated in Figure 3. The heat exchanger was inclined about its top edge, and the cross sectional area of the heat exchanger for all tests was constant.

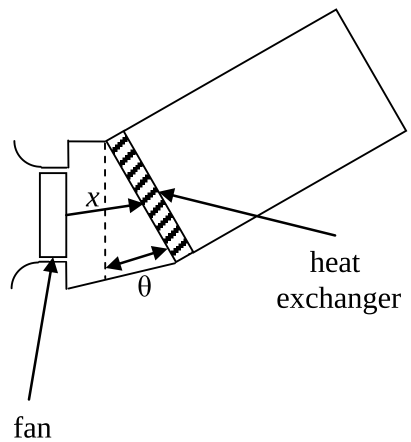

Figure 3 Inclination of the heat exchanger

EXPERIMENTAL EQUIPMENT

A fan test rig was manufactured according to ISO 5801:2008 [14]. This is shown schematically in Figure 4.

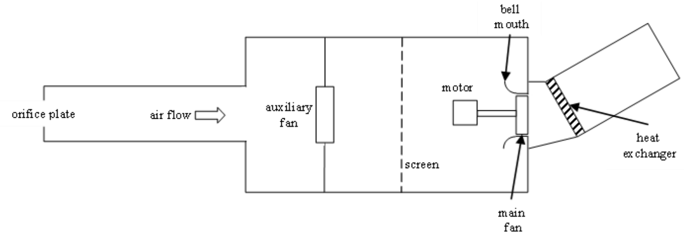

Figure 4 Fan test facility

The mass flow rate through the test rig was measured using an orifice plate in an installation according to ISO 15377 [15]. An auxiliary fan was used to control the flow rate through the system and to boost the performance of the main fan. A screen after the auxiliary fan was used to provide uniform flow into the main fan. An outlet duct was constructed after the heat exchanger, to make numerical modelling of the system simpler. The main fan used was an axial fan with a sickle type blade profile. The heat exchanger investigated was a louvered fin type with a single row of flat tubes.

A data acquisition system was used to collect and process the data from the test rig. This included a Labview program which calculated the flow rate through the system based on the pressure drop across the orifice plate.

EXPERIMENTAL MEASUREMENTS AND PROCEDURE

In the experiments, the flow rate through the system was fixed and the pressure difference between the fan inlet and heat exchanger outlet was measured. The fan characteristic was measured by removing the exit assembly consisting of the shroud, heat exchanger and exit duct, changing the flow rate and measuring the pressure rise across the fan. Measurements were taken at five different flow rates on the fan characteristic, in order to determine the effects of inclination at different operating points. The measurement points and the measured fan characteristic are shown in Figure 5.

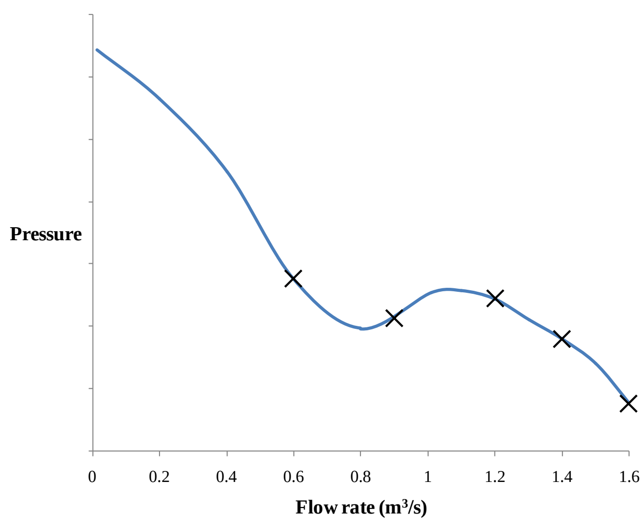

Figure 5 Fan characteristic with measurement points

The heat exchanger characteristic used was based on supplier measurements. The data was checked by removing the main fan and using the auxiliary fan to drive the flow through the heat exchanger. A plenum 0.8 fan diameters deep was used and the pressure loss across the heat exchanger was measured for different flow rates. The two sets of measurements were close enough to have confidence in the supplier data.

Pressures on the test rig were measured using pressure transducers, calibrated against a Betz manometer. The fan shaft speed was measured using a laser optical speed sensor, and the torque with an in-line rotary torque transducer. Room pressure was measured with a precision barometer. Temperature measurements were taken at the orifice inlet and main fan inlet using PT 100 sensors. Humidity was also measured at the fan inlet. The velocities of the air through the heat exchanger were measured at the exit face on a grid of 16 evenly spaced locations using a vane anemometer. All instruments were calibrated.

The procedure for a given flow rate was to take readings every second for ten seconds, and to repeat this to give two sets of readings that could be averaged. Anemometer readings were also taken twice and averaged.

PROCESSING AND PRESENTATION OF EXPERIMENTAL RESULTS

As previously mentioned, the flow rate was fixed and the pressure across the fan and heat exchanger was measured. This pressure will be abbreviated as Δp_{system} or Δp_{sys}. The main components of Δp_{sys} are

$$\Delta p_{sys} = \Delta p_{fan} + \Delta p_{pl} - \Delta p_{he} \tag{1}$$

where Δp_{fan} is the pressure rise across the fan, Δp_{pl} is the pressure rise across the plenum and Δp_{he} is the pressure drop across the heat exchanger.

Δp_{he} is composed of the pressure drop due to uniform flow Δp_{heu} plus an additional drop due to the non-uniformity of the flow through the heat exchanger Δp_{henu}.

Rearranging equation (1) by combining Δp_{sys} and Δp_{heu} gives

$$\Delta p_{sys} + \Delta p_{heu} = \Delta p_{fan} + \Delta p_{pl} - \Delta p_{henu} \tag{2}$$

Plotting Δp_{fan} + Δp_{pl} - Δp_{henu} gives a means of directly comparing the performance of different inclinations or plenum depths, accounting for all the factors that are changing.

The results required some processing before they could be presented in the form of equation (2). The goal was to present all data at standard conditions, in this case being defined as:

ρ_{st} = 1.2 kg/m^3

T_{st} = 293.15 K

The data at test conditions needed to be corrected to standard conditions. For the fan, this meant density correction of the pressure rise across it, using the following relationship

$$\frac{\rho_t}{\Delta p_t} = \frac{\rho_{st}}{\Delta p_{st}} \tag{3}$$

derived from the Euler equation for turbomachinery.

The pressure rise across the fan will not be completely Reynolds number independent, as the Reynolds number of the fan $Re_d \approx 0.9 \times 10^6$ which is below the threshold Reynolds number of 2.5×10^6 as noted by [16]. The relationship between fan pressure rise and Reynolds number is unknown, so it is not possible to correct for it, although the impact of deviation from standard conditions in the tests is small.

For the heat exchanger, the relationship between Fanning friction factor and Reynolds number is given by

$$f \propto \mathrm{Re}_{Lp}^{-0.39} \tag{4}$$

for $1000 < Re_{Dh} < 4000$. Re_{Dh} for this study is at the high end of this range. Equation (4) [17] is for a heat exchanger of a similar type to that used in this study.

Since head loss is directly proportional to Fanning friction factor

$$\Delta p = \rho g h$$
$$\Rightarrow \Delta p \propto \rho g f$$
$$\Rightarrow \Delta p \propto \rho \,\mathrm{Re}^{-0.39}$$

The result is

$$\Delta p \propto \rho^{0.6} \tag{5}$$

Also

$$\Delta p \propto \mu^{0.4} \tag{6}$$

The heat exchanger pressure loss is corrected using equations (5) and (6).

The procedure for correcting the data is to first calculate the plenum recovery at test conditions using equation (2), with all parts of equation (2) being at test conditions. The plenum recovery is assumed to be constant, no matter what the air properties. This plenum recovery is then used again in equation (2) with all parts being at standard air conditions, to work out $\Delta p_{fan} + \Delta p_{pl} - \Delta p_{henu}$.

The pressure drop across the heat exchanger is calculated based on the anemometer measurements taken at the heat exchanger exit. Jetting occurs at the heat exchanger exit, which means that the anemometer reads velocities larger than the true velocity, so the readings need to be scaled to match the mass flow rate as measured with the orifice plate. The corrected readings are then used with the supplier heat exchanger data to determine the pressure drop for each section, which is then averaged for the whole heat exchanger. This gives the total pressure loss across the heat exchanger. Subtracting the loss based on the average velocity gives Δp_{henu}.

EXPERIMENTAL RESULTS

The repeatability of the results was checked by conducting three experiments on three separate days at 30° inclination. For each repeat, the exit assembly was disassembled and reassembled. The results are shown in Figure 6.

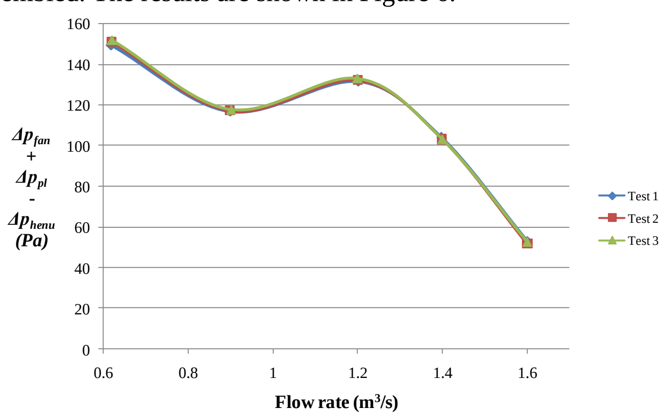

Figure 6 Repeatability of results at 30° inclination

The error in the pressure measurements related to Figure 6 (and all other figures containing experimental pressure measurements) is ± 1.5 Pa, which is small. This includes 1 Pa to account for the resolution of the Betz manometer used for calibration and 0.5 Pa for the accuracy of the pressure transducer. The error in the mass flow rate is also small. It is approximately 1%, resulting from uncertainty in the discharge coefficient of the orifice plate.

The results for inclination of the heat exchanger are shown in Figure 7.

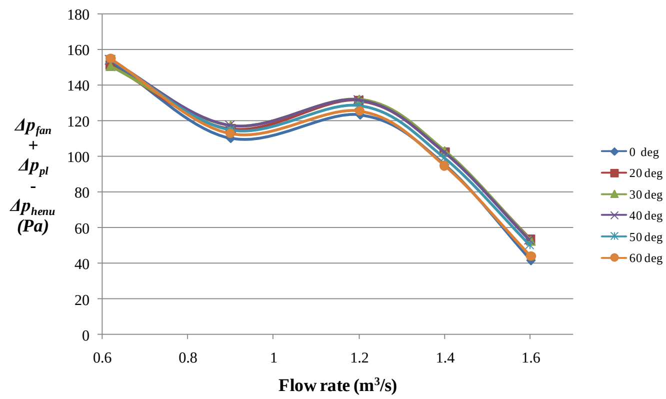

Figure 7 Effect of inclination of heat exchanger

The results for increasing the depth of the plenum without inclination are shown in Figure 8. Note that the plenum depth x_{pl} is non-dimensionalised by dividing by the fan casing diameter d_{fc}.

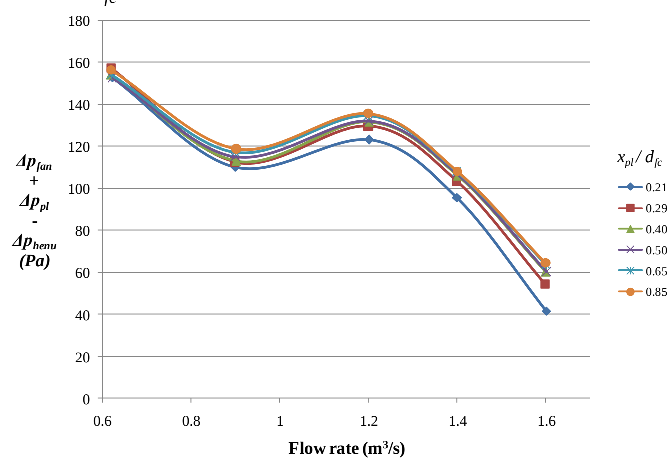

Figure 8 Effect of increased plenum depth without inclination

By examining Figures 7 and 8, a number of conclusions can be drawn. Firstly, the effect of inclination and increasing plenum depth is negligible in the mixed-flow region of the fan characteristic. Secondly, by comparing Figures 7 and 8, the effect of increasing plenum depth is more significant than inclining the heat exchanger. Figure 7 shows that the optimum inclination angle is somewhere around 20° to 40° inclination. Figure 8 shows that the optimum plenum depth is 0.65 fan diameters or greater - the critical minimum plenum depth. The heat exchanger used has a low pressure loss factor, so this critical minimum depth confirms the findings of [1] that critical minimum depth increases with decreasing heat exchanger restriction.

Figure 9 shows the results at 1.4 m^3/s for both inclination and depth only. The y-axis label is the pressure loss across the system. The smallest pressure loss is for 30° inclination, indicating that this is the optimum inclination angle. The smaller the system pressure loss, the better the performance as this means that the auxiliary fan does not have to work as hard to achieve the required flow rate, as the performance of the main fan is better.

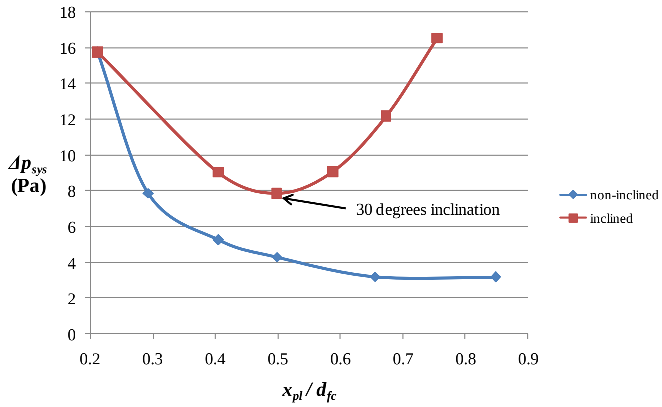

Figure 9 Effects of inclination and depth at 1.4 m^3/s.

The thermal performance of a number of cases of interest was calculated, using supplier data for the heat exchanger thermal performance and the heat exchanger velocity measurements. The thermal performance was determined for each velocity measurement square of the heat exchanger and the results summed over the entire face. The case with no inclination and a plenum depth of 0.21 fan diameters was used as a baseline. The results are shown in Table 1. The results show that the effects of inclination and increased plenum depth on thermal performance are small.

Table 1 Thermal performance improvement due to inclination and increased plenum depth

case	flow rate (m^3/s)	% change in flow	Q (W)	% change in Q
0 degrees	1.359	n/a	33838	n/a
30 degrees	1.382	1.69	34013	0.52
x_{pl} / d_{fc} = 0.65	1.397	2.80	34215	1.12

A final experimental consideration is the significance of inlet losses on the air side of the heat exchanger passages. This was investigated by removing the main fan and measuring the pressure drop across the heat exchanger for 60° inclination and for a perpendicular heat exchanger in a plenum of equivalent depth. The results are shown in Figure 10, and the conclusion is that inlet losses are significant at 60° inclination, for fairly uniform non-swirling flow entering the heat exchanger.

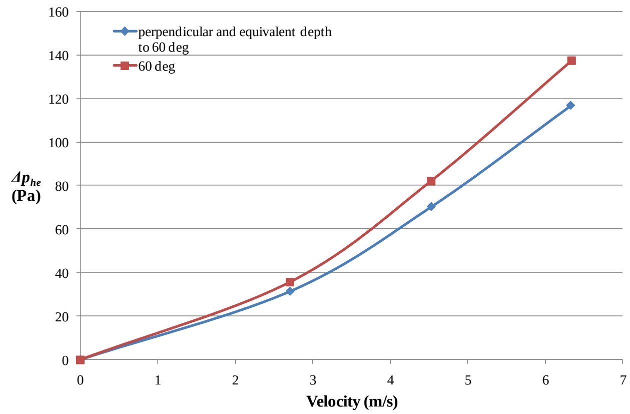

Figure 10 Effects of inlet losses at 60° inclination

NUMERICAL STUDY

A CFD study was conducted using ANSYS CFX to investigate how the flow changes in the plenum with inclination. The model is shown in Figure 11. All simulations were conducted at 1.4 m^3/s.

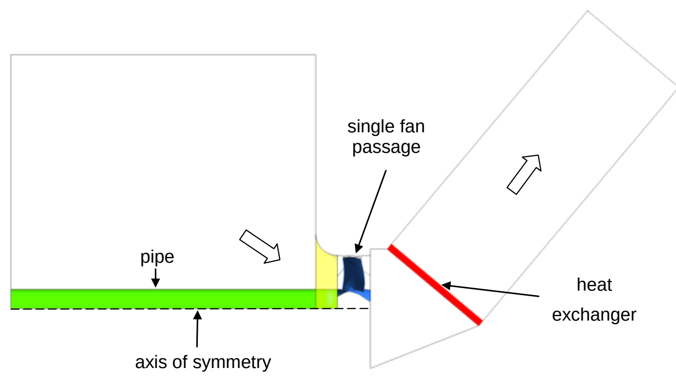

Figure 11 CFD model

A pipe was included with the same diameter as the fan hub, directly in front of the hub, as on the test rig. This was to provide a wall for flow to attach to as it enters the fan. The fan was modelled using a mixing plane multiple reference frame model. This means that the flow is circumferentially averaged at the inlet and outlet of the fan domain. Only one fan passage is modelled, and one quarter of the upstream domain, as the flow is axisymmetric up to the fan exit.

The heat exchanger was modelled as a porous medium, with the flow constrained to move in a direction perpendicular to the inlet face. The flow through the airside passages was not modelled in detail, so inlet losses were not included in the CFD model.

In the simulations the Reynolds-averaged Navier-Stokes equations were solved. The flow was modelled as being steady, viscous, compressible and isothermal. The turbulence model used was k-ω SST. The advection scheme used was formally second-order accurate in space.

The fan was meshed with a structured mesh using ANSYS Turbogrid. Each passage had approximately one million cells.

A structured grid was also used for the outer domain, with approximately two million cells.

Sensitivity studies were conducted for the simulations, to ensure that the results of interest were independent of residual target, outer domain size, turbulence intensity, fan blade cell y^+ and independent of the number of cells in the fan and outer domains.

MODEL VALIDATION

The CFD model was validated by taking pressure measurements along the side walls of the plenum in the experiments. A total of 18 tappings were located along the side and base walls of the plenum, on lines midway along the walls. The model and the experiment compared well. For example, the comparison for the base of the plenum is shown in Figure 12, for 40° inclination.

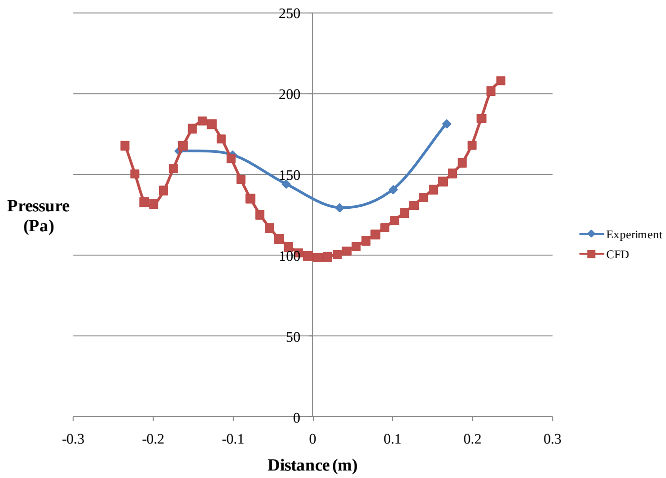

Figure 12 Validation of CFD - pressure midway along plenum base for 40° inclination

Further validation was achieved by comparing the pressure loss across the system with the experiment, for different inclination angles. This is shown in Figure 13.

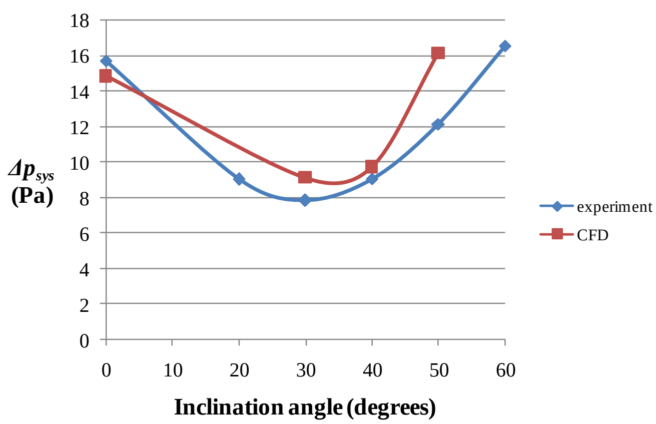

Figure 13 Validation of CFD - system pressure loss at 1.4 m^3/s

In Figure 13 the results compare well between the experiment and the CFD. The CFD model does not account for inlet losses, so as the shape of the two lines are similar, this suggests that inlet losses do not change with angle. This can be explained by the fact that the flow entering the heat exchanger is swirling, no matter what the inclination angle. Figure 10 suggested that inlet losses are significant (not including fan swirl), but this is not apparent from Figure 13 as the two lines are almost on top of each other.

Finally, the velocity distribution measured at the heat exchanger exit was compared with the CFD. This is shown in Figure 14. The patterns are similar, although the range of velocities is higher for the CFD.

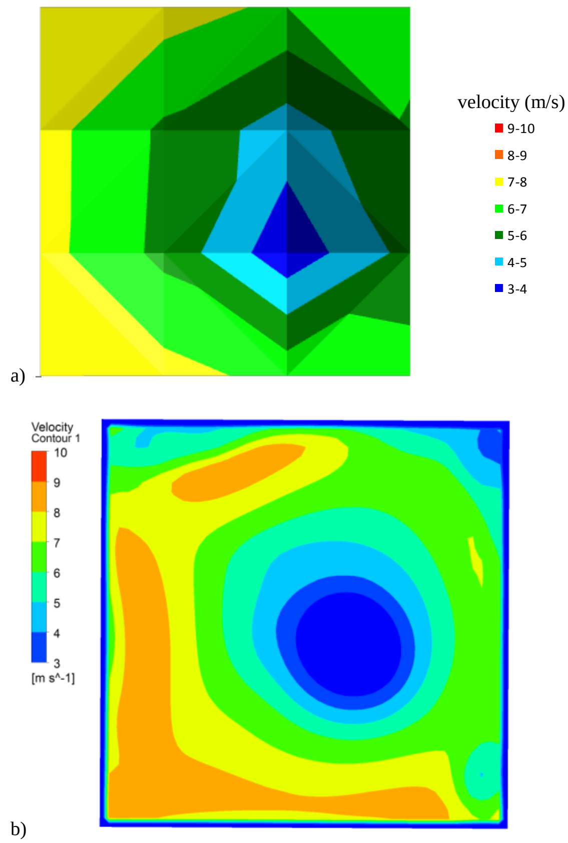

Figure 14 Heat exchanger exit velocity distribution at 30° inclination for a) experiment and b) CFD

CFD RESULTS

Figure 15 is a plot of the area-averaged pressure rise across the plenum from the fan outlet to the heat exchanger inlet. Also plotted is the fan static pressure rise, which is the area-averaged static pressure at fan outlet minus the mass flow averaged total pressure at the outer domain inlet.

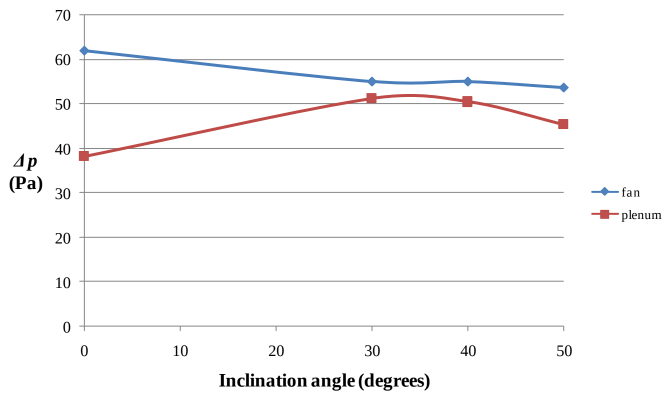

Figure 15 Pressure rise across fan and plenum

From Figure 15 it can be concluded that the closer the heat exchanger is to the fan, the higher the static pressure rise across it. It also shows that pressure recovery in the plenum increases with plenum depth up to approximately 30° to 40° inclination, where it starts to reduce. Plotting the sum of the two gives a pattern that is the mirror of the CFD results in Figure 13.

Figure 16 is a plot of the area-averaged pressure at planes at different sections along the plenum, perpendicular to the main flow direction. The direction *x* is shown in Figure 3. It shows that pressure recovers fairly consistently across the plenum.

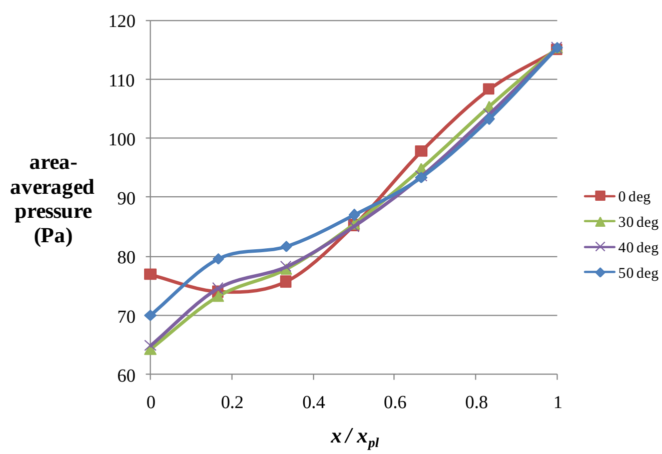

Figure 16 Pressure recovery in the plenum

The final issue to discuss is how the flow through the heat exchanger changes with inclination. Velocity contour plots at the heat exchanger exit are shown for different inclination angles in Figure 17. The results show that the low velocity core at the centre of the heat exchanger moves to one side as the heat exchanger is inclined. This occurs because of the swirl of the flow exiting the fan and its interaction with the corners of the square duct and the angled heat exchanger downstream. Plots at a flow rate in the mixed-flow region of the fan characteristic from experimental results showed a similar effect.

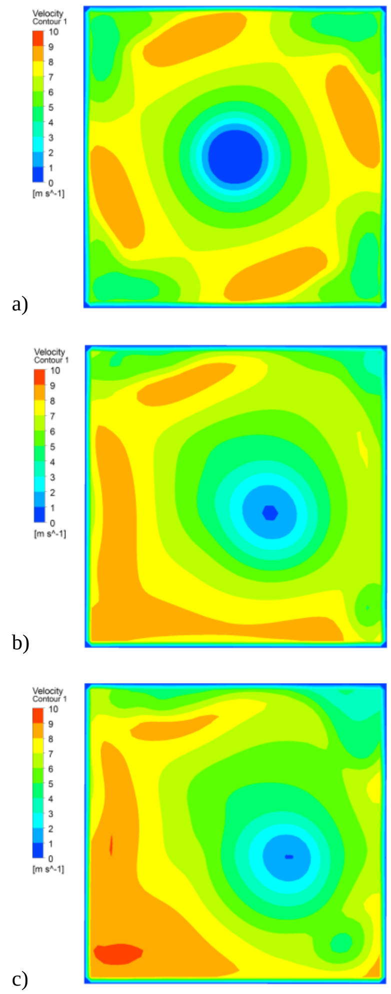

Figure 17 Heat exchanger exit velocity plots for a) 0° inclination, b) 30° inclination and c) 50° inclination

Figure 18 is a plot of streamlines in the plenum of the ACHE at 30° inclination, viewed from the exit of the downstream duct. On the right hand side, the flow bypasses the heat exchanger inlet, due to the swirl of the flow exiting the fan. On the left hand side, the swirl of the flow means that the flow is more aligned with the airside passages. Therefore on that side most flow passes directly through the heat exchanger rather than over its face.

Most turbocharged engine cooling systems have a charge air cooler as well as a jacket water cooler. These are often located side by side. If one of the two coolers has a significantly worse performance than the other, then it may be beneficial with inclined heat exchangers to locate that cooler on the side where the flow is more likely to pass directly through rather than bypass the inlet face. This may enable the cooling system to operate in an environment with a higher ambient temperature.

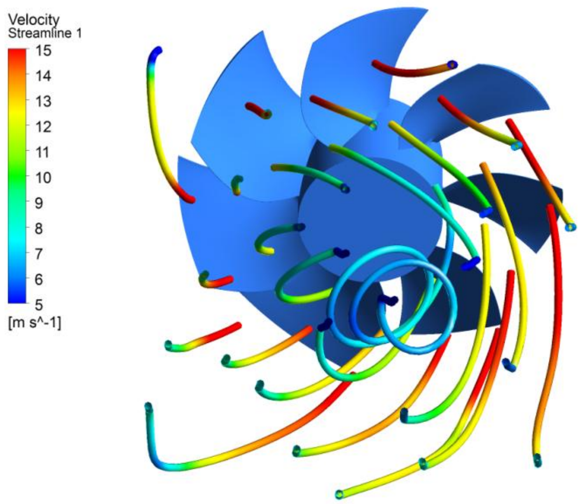

Figure 18 Plenum streamlines at 30 degrees inclination, viewed from the exit of the downstream duct

CONCLUSIONS

The main conclusion of the study is that inclination leads to a small increase in the thermal performance of the ACHE of approximately 0.5% for the optimum inclination angle of 30°, when compared with the baseline case of a shallow plenum. Increasing the depth of the plenum is slightly more effective, giving a performance increase of 1% for a plenum 0.65 fan diameters deep.

Of more significance is what happens to the flow through the heat exchanger as it is inclined. The tendency is for the low velocity core to move to one side as the inclination angle is increased. This could lead to improvements in the performance of the cooling system if the ACHE consisted of multiple side by side heat exchangers. In that case the designer could locate the performance limiting component on the side opposite that of the low velocity core.

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