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DETERMINATION OF HEAT TRANSFER RATE AND PRESSURE DROP PERFORMANCE OF AN INTERCOOLER FOR HEAVY DUTY ENGINES

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Intercoolers have been increasingly used in internal combustion engines with supercharging since 1990s because of their positive effect on engine power and fuel consumption. In this study, a louvered fin and plate intercooler were experimentally investigated for heat capacity ratios between 0.027-0.125 and compactness value of 664.6 m^2/m^3 for heavy duty engines. Heat transfer rate and pressure drop performance of the intercooler were determined with dimensionless indicators such as effectiveness, Nusselt number and friction factor. Temperature difference between fluids were changed between 50-110 ^oC and Reynolds number for cooling air side was changed between Re=1500-5500 in louvered fins with having a hydraulic diameter of 0.0019 m for Prandtl number value of Pr=0.82 of the cooling air. Intercooler effectiveness was found between 94-98% and friction factor was in the range of 1.2-1.6. The intercooler was determined as favorable for heavy duty engines, especially agricultural tractor engines. According to the experimental results, it was understood that designed and manufactured experimental setup can be effectively used for determining thermal and hydrodynamic performances of practically used intercoolers. The obtained results will be helpful for the validation and assessment of numerical methods and designing as well as testing new compact heat exchangers.

Keywords: Effectiveness, Heat transfer, Intercooler, Number of transfer units, Pressure drop.

1. Introduction

Today's state of art technology for internal combustion engines is focused on three important problems; cost, environmental impact and depleting fossil fuels. These problems should be overcome without sacrificing performance and comfort. One of the most accepted solution for aforementioned problems is downsizing of the engine via supercharging [1].

Supercharging increases charge air density and hence more fuel can be burned to attain more power and higher efficiency for the same engine size. In other words, engine sizes can be reduced for a specific power output. However charge air temperatures increase with supercharging which, in return, decreases air density comparing its possible value at ambient temperature. Canliet al. (2010) conducted a theoretical study about intercooling concept by using selected real internal combustion engine properties and a real turbocharging system compressor and turbine map [2]. They recorded that thecomparison of the entrance of the cooling supercharged charge air and ambient air temperature resulted in 53% more power output from the same engine for the case of supercharger.

For charge air cooling, the most applied method is using a heat exchanger between supercharging system and engine. This heat exchanger called as intercooler or aftercooler. When real time operational conditions are taken into consideration, *intercoolers* have some characteristics affecting charge air cooling or intercooling ratio. These characteristics and methods for evaluating them are same with heat exchangers in the operational range of intercoolers. One of the most important characteristics of intercooler is the effectiveness which represents the ability of intercooler to cool down charge air to ambient air temperature. When intercooler effectiveness is 1.0 or 100 percent, charge air temperature is equal to ambient air temperature. Cooling capacity and pressure drop value are other important characteristics.

There are a big variety of scientific studies for calculating and determining heat exchanger characteristics. Nuntaphanet al.(2010) preferred experimental method for determining heat exchanger effectiveness and heat transfer coefficient [3]. Ismail et al. (2010) reviewed the specific literature about wavy fin offset fin configuration in respect of pumping losses and heat transfer enhancement [4]. They presented the most recent numerical and experimental equations from the literature. Results from these equations were compared from experimental results of Kays and London (1998) [5]. They found 30% agreement with experimental results for friction factor and 20% agreement with experimental results for Colburn factor. Li and Wang (2010) also used experimental method for determining hydrodynamic and thermal performance of a brazed louvered fin heat exchanger [6]. Experiments were conducted at 400-1600 Reynolds number interval. Wen and Ho (2009) conducted an experimental study in the range of $600 \leq Re \leq 2000$ for the cold fluid of the finned tube heat exchanger [7]. Effectiveness-number of transfer unit method was used. They indicated that pressure drop, heat transfer amount, friction factor and Colburn factor increased about 10.0-31.9%, 11.8-24.0%, 2.2-27.5% and 0.5-2.7%, respectively when wavy fin preferred instead of flat fin. Peng and Ling (2009) investigated artificial neural networks for determining friction factor and Colburn factor of plate fin compact heat exchangers [8]. They expressed that 1.5% deviation was occurred for Colburn factor and 1% deviation was occurred for friction factor by the utilization of artificial neural network. In that study, back propagation method was preferred. Metin (2008) compared experimental results with ε-NTU and computational fluid dynamics (CFD) results [9]. Corberanet al. (2008) investigated pressure drop in plate fin heat exchangers with an experimental setup in the range of $20 \leq Re \leq 5000$ [10]. Various correlations from literature were compared with experimental results of this study. Jungi et al. (2007) experimentally investigated 11 different wavy fin geometries on flat tubes [11]. Inside flat tubes, hot fluid was flowing at fixed volumetric flow rate at 2.5 m^3/h value. Cooling air Reynolds numbers was changed between 800-6500 according to its volumetric flow rate and fin geometry. Effectiveness-number of transfer units was utilized during determination of heat transfer characteristics. With their derived correlations, they suggested that results can be predicted with 95% accuracy. The heat transfer enhancement of transverse ribs in circular tubes with a length-to-diameter ratio of 87 was experimentally investigated by San and Huang (2006) [12]. The mean heat transfer and friction data were obtained for the air flow started from the entrance. An isothermal surface condition was considered. The rib pitch-to-tube diameter ratio (p/d) was in the range 0.304–5.72; the rib height-to-tube diameter ratio (e/d) was in the range 0.015–0.143; the considered Reynolds number (Re) was in the range 4608–12,936. They reported that the mean Nusselt number (Nu) and friction factor (f) were individually correlated as a function of p/d, e/d and Re.Kays and London, experimentally studied 132 compact heat exchanger surfaces in a special designed experimental setup for determining their heat transfer and pressure drop performances [5]. They present their results with dimensionless numbers changing with Reynolds numbers in a specific Prandtl number range. For determining heat exchanger effectiveness in cross flow compact heat exchangers, a broad contented study was conducted by Sekulicetet al. [13]. They categorized the methods of determining heat exchanger performances as analytical methods, approximate approaches, curve fitting methods, numerical methods, and methods for designing matrix form and cross flow heat exchangers. Pigotti and Shah [14], obtained 18 open formulas due to effectiveness-number of transfer unit method. Patankar and Prakash investigated the effect of plate thickness to heat transfer and pressure drop [15]. Kim and Sohn performed an experimental study on saturated flow boiling heat transfer of R113 in a vertical rectangular channel with offset strip fins [16]. Two-phase pressure gradients and boiling heat transfer coefficients in an electrically heated test section were measured for the quality range of 0–0.6, mass flux range of 17–43 kg/m^2 s and heat flux of 500–3000 W/m^2. Muzychka [17] and Muzychka and Yovanovich [18] performed experiments using transmission oil as the working fluid and derived correlations by an analytical model. The number of fin geometries that they tested, however, was not sufficiently large to establish a general correlation.Canliperformed a broad experimental investigation on different compact heat exchanger configurations for air to compressed-air streams via a special designed experimental setup and determined heat exchanger effectiveness, heat transfer and pressure drop characteristics in his master of Science thesis [1]. There are also other experimental studies for specific heat exchanger configurations by different authors [19, 20, 21]. Canliet. al.(2011) conducted experimental performance analysis of finned in line circular tube bank intercooler configuration at low range thermal capacity ratios [22]. Using similar experimental data and correlations, theoretical comparisons of heat exchanger configurations can be performed. Bilen and Doganet. al. reported two representatives of those studies [23-24]. Li and Wang (2010) and Doganet. al. (2011) also used experimental method for determining hydrodynamic and thermal performance of brazed louvered fin heat exchanger and wavy-staggered fin heat exchanger, respectively [25, 26].

Because the thermal resistance of air is high, the dominant thermal resistance of the intercoolers is sourced from air, which may account for 85% or more of the total thermal resistance. The use of enhanced fin surface is the most effective way to improve the overall performance of the intercooler to meet the demand of high efficiency and low cost. Fins employed on the gas side can increase the heat exchanger surface area and strengthen the flow disturbance. Typically, these enhanced surfaces are developed from corrugated fin to interrupted fin such as slits, louvers, and offset-strip fins. The wavy surface can periodically changes the main-flow direction and causes better flow mix, the slit or louvered-fin can periodically interrupt the main-flow, break and renew the thermal boundary layer [27].

In this work, a louvered fin and plate intercooler characteristics and their effects to intercooling were experimentally investigated. Intercooler configuration is explained in an elaborate manner and results are presented in graphical form for cooling capacity, pressure drop in heat exchanger core and intercooler effectiveness. Details of experimental setup and method used in this study are also provided in the content of present paper.Heat capacity ratios were between 0.027-0.125 and compactness value was664.6 m^2/m^3. Heat transfer rate and pressure drop performance of the intercooler were determined with dimensionless indicators such as effectiveness, Nusselt number and friction factor. Temperature difference between fluids were changed between 30-90^{o}C and Reynolds number for cooling air side was changed between Re=1200-5500 in louvered fins with a hydraulic diameter of 0.0019 m for Prandtl number value of Pr=0.82 of the cooling air.

2. Experimental Setup and Procedure

Intercoolers which use ambient air as coolant are the focus of the present study. Hence, cooling air channel of the experimental setup was designed accordingly. One can replace a heat exchanger with another easily and therefore a variety of heat exchangers can be tested in the experimental setup. Details are given below.

2.1. Experimental setup

To provide a better understanding, three dimensional solid drawings of the experimental setup is given in Fig.1.

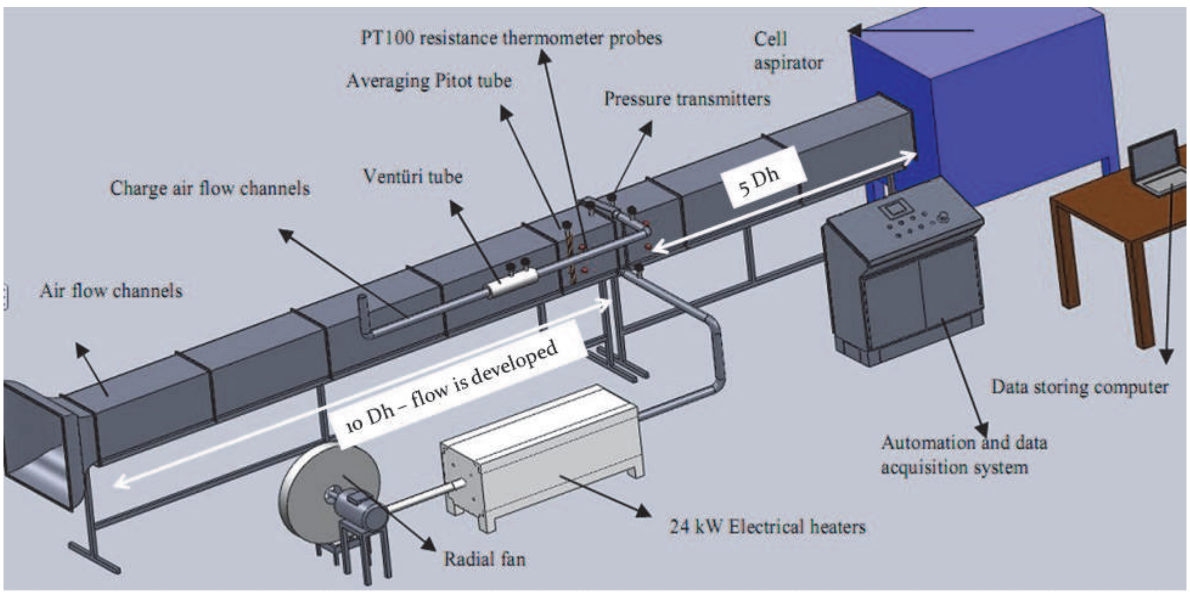
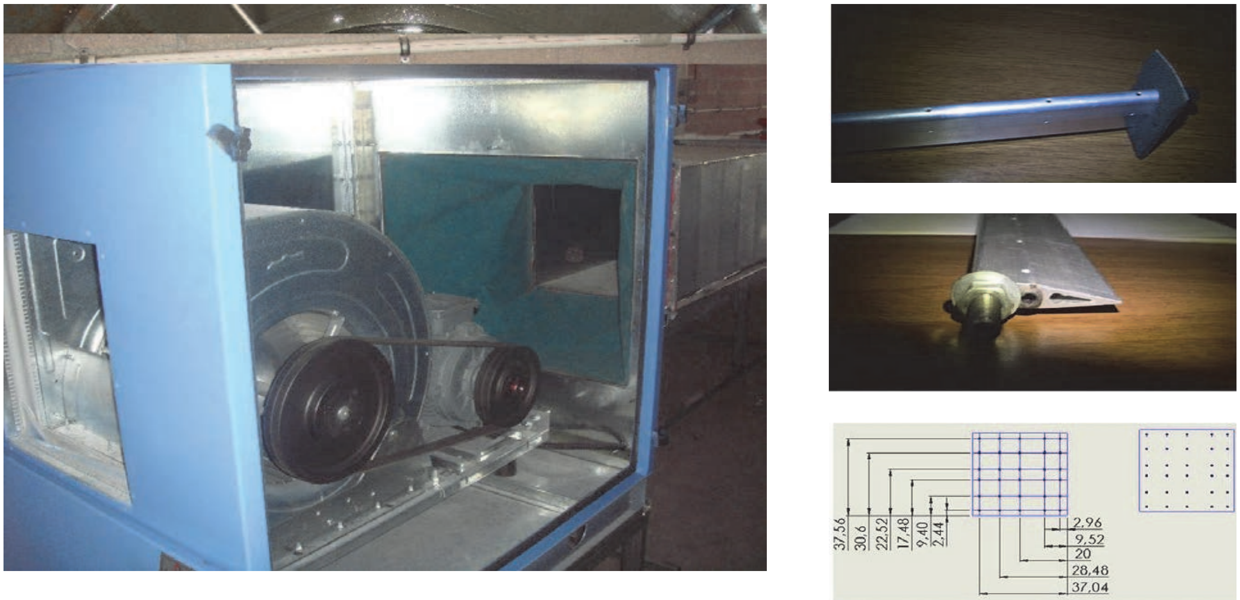


Figure 1. 3D solid model of experimental setup for intercooler tests.

In coolant air channel a cell type aspirator was used for forced convection and it was controlled with an electrical current frequency controller having same capacity with electrical motor of the aspirator. Selected electrical motor for the aspirator was 5.5 kW. Coolant air volumetric flow rate was changed between 0.4 and 1.9 m^3/s. Coolant air flow channel dimensions were 0.4x0.4 m. Thus its cross section was the first challenge which was too big to measure volumetric flow rate of the coolant air with a commercial flow meter. It was decided to measure the flow rate by using five averaging pitot tubes having 6 measurement points. Therefore, coolant air flow rate was measured from 30 points at a cross section. Pitot tubes were placed after ten hydraulic diameters from the entrance of air channel at upstream of heat exchanger slot to provide fully developed air flow. Log-Thebycheff traversing method was selected during determination of measurement point locations. The details of Log-Thebycheff method was given in ISO 3966 [28]. In Fig. 2, cell type aspirator, its interior, averaging pitot tubes and distribution of measurement points are demonstrated.



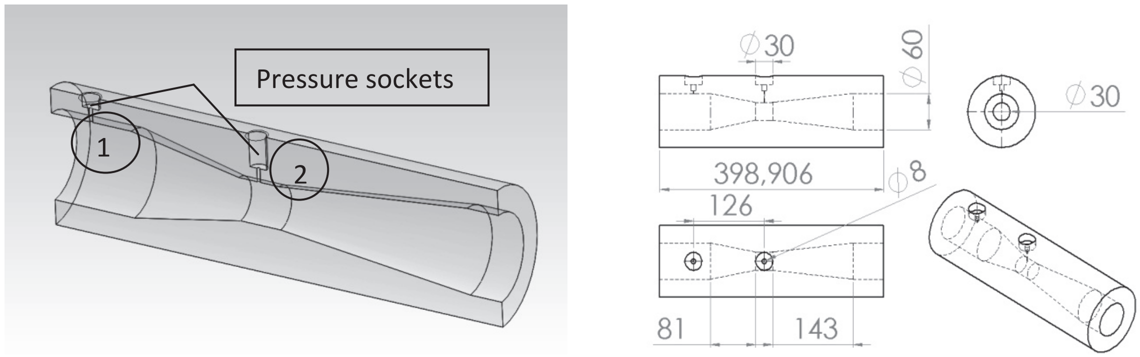
-a-

-b-

Figure 2. A-Cell type aspirator and its interior; B- averaging pitot tubes and distribution of measurement points.

Hot air flow rate was measured with a venturimeter in hot air channel. The venturimeter was designed according to ASME and DIN standards which were summarized in a study of Genceli [29, 30, 31]. Dimensions of venturi tube in millimeters and a sample solid model are presented in Fig. 3. Two pressure transmitters were used in venturimeter. In Fig. 4, a photograph belonging to venturimeteris introduced. A centrifugal fan was utilized for pressurized hot air channel. The air was then heated in an electrical resistance heater with 24 kW capacity. Electrical heater was consisted of three 8 kW heater blocks. A view of centrifugal fan and solid models of electrical heater are introduced in Fig. 5.

Two kinds of measurement were performed in the experimental setup; pressure and temperature measurements. All other properties used in evaluation of heat exchangers were calculated using these two values. For pressure measurement, single pressure transmitters with 4-20 mA output were used except for pitot tubes. A differential pressure transmitter was employed for pitot tubes since there were very little pressure difference and it was fluctuating rapidly. Two kinds of temperature measurement probes were utilized in the experimental setup. The majority was constituted by PT100 thermo element. Only two J type thermocouples were used while 15 PT100 were employed. Measurement data were collected by Siemens S7-200 PLC and transferred to a computer for data storage. Pressure and temperature measurement results which were recorded between 60 to 120 seconds for one measurement were averaged and recorded. Photographs of measurement points are given in Fig. 6.



a. b.

Figure 3. A-Cross section of venturimeter, B-Venturimeter for hot gas channel.

Figure 4. Venturimeter for hot gas channel.

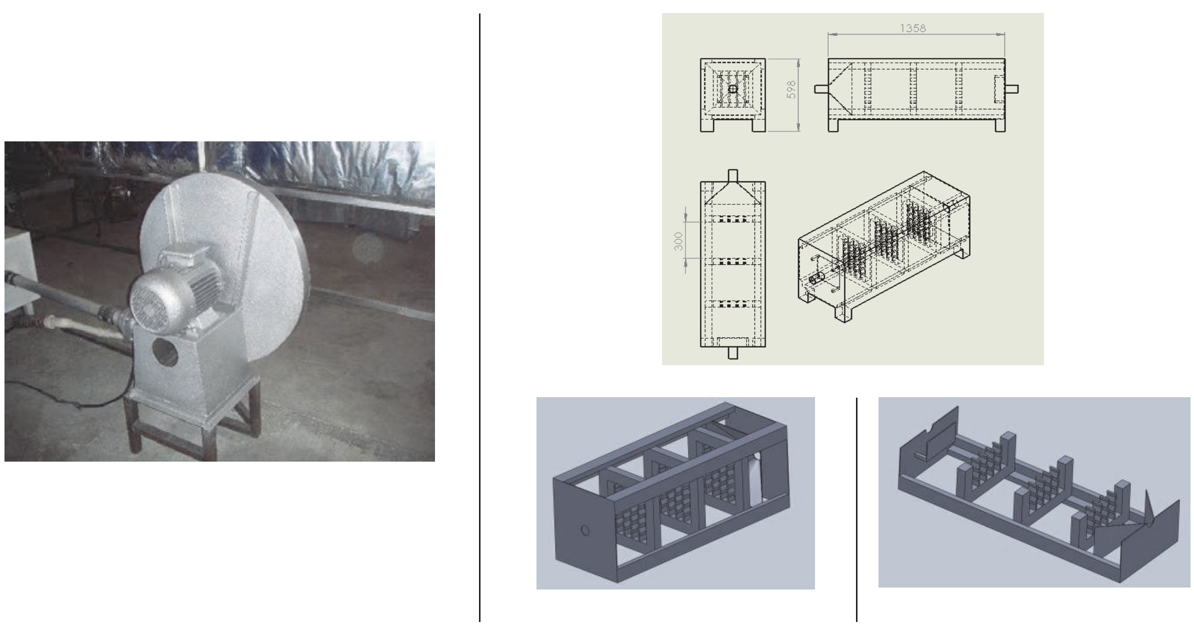


Figure 5. Centrifugal fan and electrical heater for hot air channel.

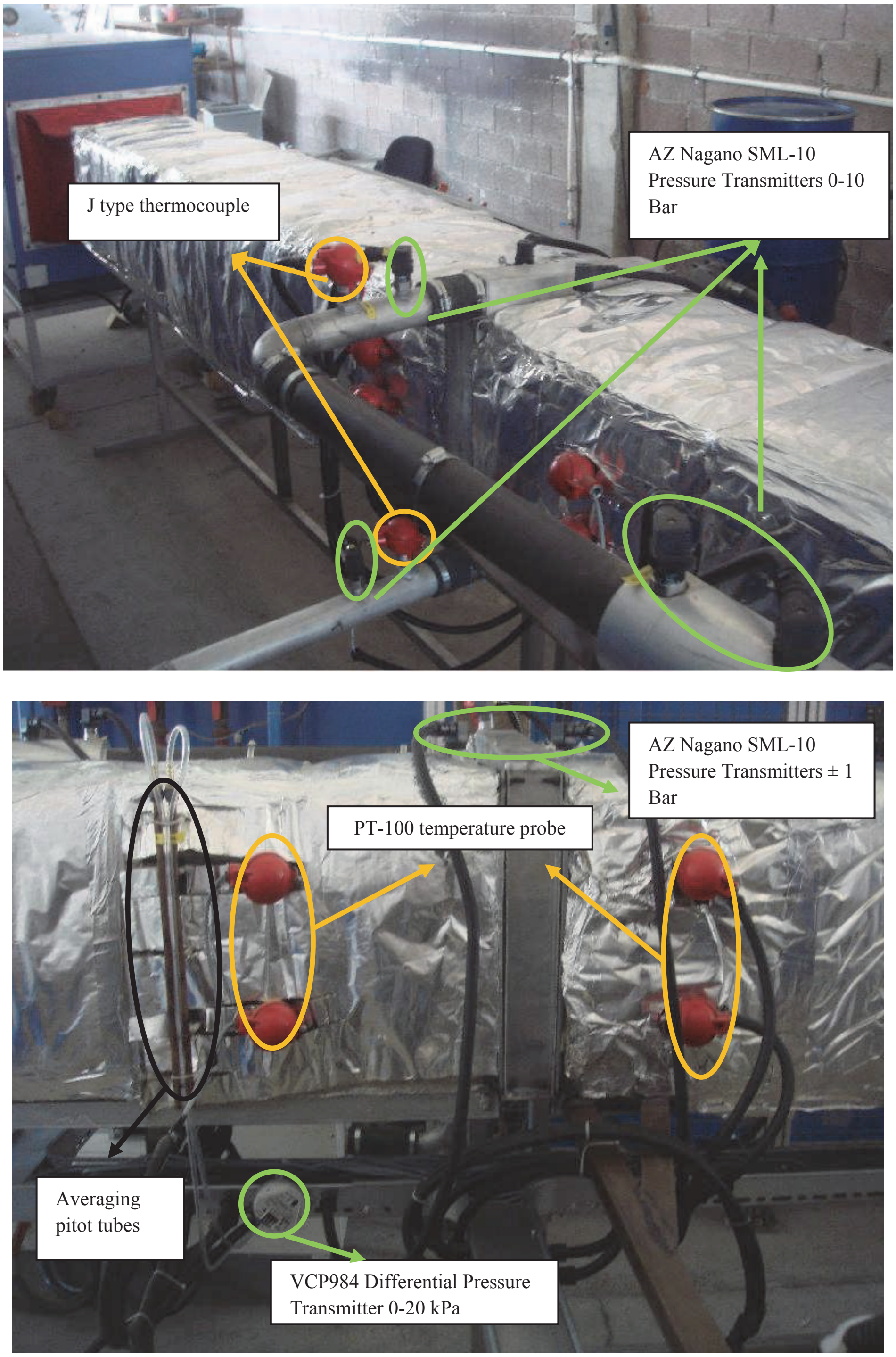


Figure 6. Measurement points in the experimental setup.

Technical drawings of the fins and their actual views are given in Fig. 7. Required dimensions about the heat exchanger are listed in Table 1. Fig. 8 contains two figures of the intercooler and the experimental setup.

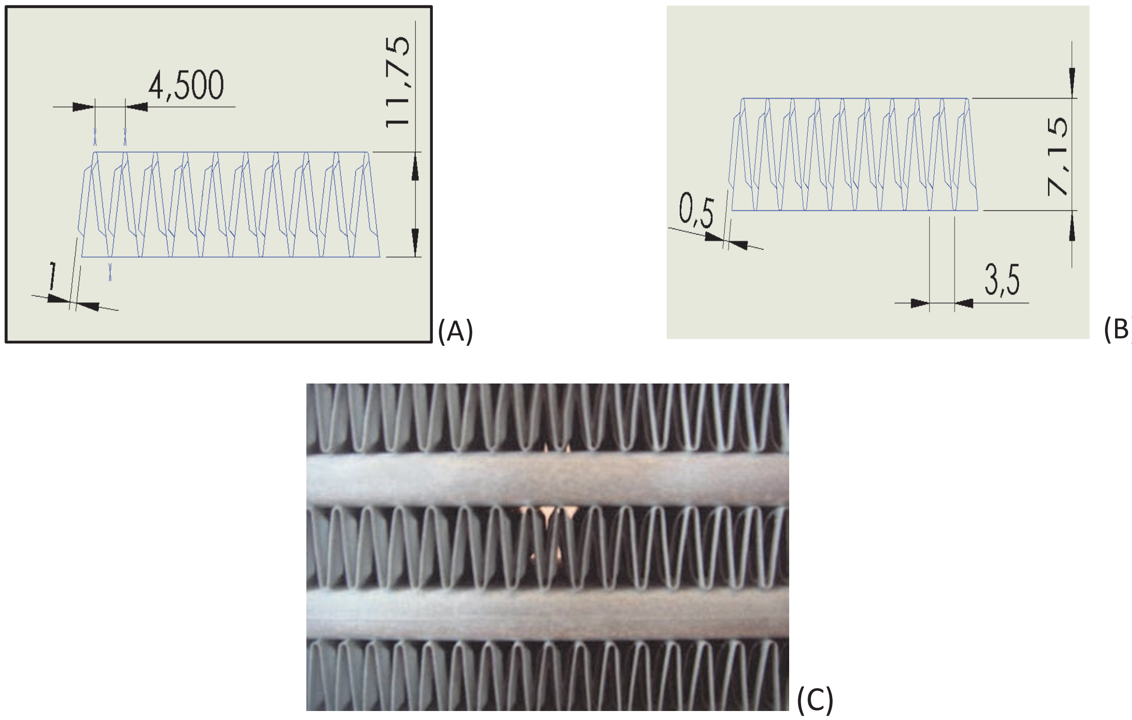


Figure 7. A- Fins in cooling air passage, B- Fins in charge air passage, C- Louvered fins.

Table 1. Required dimensions of the heat exchanger.

Total volume of the heat exchanger (m^3)	0.0113
Hot flow volume (m^3)	0.00278
Cold flow volume (m^3)	0.00789
Total heat transfer area of hot flow (m^2)	4.2688
Fin heat transfer area of hot flow (m^2)	3.343
Total heat transfer area of cold flow (m^2)	3.25
Fin heat transfer area of cold flow (m^2)	2.378
Compactness ratio (m^2/m^3)	664.6
Frontal area of hot flow (m^2)	0.027
Frontal area of cold flow (m^2)	0.1579
Free flow area of hot flow (m^2)	0.00509
Free flow area of cold flow (m^2)	0.03075
Hydraulic diameter of hot flow (m)	0.00186
Hydraulic diameter of cold flow (m)	0.00189

Figur0 8. Intercooler and experimental setup.

Ca2. 2 lculation procedure

Using pressure and temperature measurements, the following values were calculated;
Air densities changing with pressure and temperature according to ideal gas equation;

$$\rho_{H,D} = \frac{P_{H,D}}{RT_{H,D}} \tag{1}$$

Cooling air average velocity calculated by using pressure difference at averaging pitot tubes;

$$\upsilon_H = \sqrt{\frac{[2(P_{Total} - P_{Static})]}{\rho_H}} \tag{2}$$

Charge air average velocities calculated by using pressure difference at the venturimeter;

$$\upsilon_{D,O} = \frac{\sqrt{\frac{2(P_1 - P_2)}{\rho_{D,O}}}}{\sqrt{1-\beta^4}} \tag{3}$$

Volumetric flow rates can be determined as;

$$\dot{V} = A\upsilon \tag{4}$$

Mass flow rates can be determined as;

$$\dot{m} = \dot{V}\rho C_d Y \tag{5}$$

"Y" is expansion coefficient for compressible flow and it can be taken "1" for incompressible flow[31]. It was calculated as;

$$Y = \sqrt{\left(\frac{P_2}{P_1}\right)^{2/\gamma} \frac{\gamma}{\gamma-1} \frac{1-\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}\frac{\gamma-1}{\gamma}}}{1-\left(\frac{P_2}{P_1}\right)} \frac{1-\beta^4}{1-\left[\beta^4\left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}}\right]}} \tag{6}$$

Coefficient of discharge is;

$$C_d = \left(3\times10^{-29}\,\mathrm{Re}^5\right) - \left(4\times10^{-23}\,\mathrm{Re}^4\right) + \left(2\times10^{-17}\,\mathrm{Re}^3\right) - \left(7\times10^{-12}\,\mathrm{Re}^2\right) + \left(1\times10^{-6}\,\mathrm{Re}\right) + 0.9294 \tag{7}$$

At Re<100,000 circumstance and C_d= 0.98 at Re>100,000 circumstance [31].

Reynolds numbers were calculated as;

$$\mathrm{Re} = \frac{\rho \upsilon D_h}{\mu} \tag{8}$$

Dynamic viscosities changing with temperature were calculated as;

$$\mu_D = \left(0.4\times10^{-7}\,T_D\right) + \left(0.2\times10^{-4}\right) \text{forcharge air and} \tag{9}$$

For specific heat values of air changing with temperature, following equation was derived using thermodynamic property tables [31].;

$$c_{pH,D} = \left\{28.11 + \left[0.1967\times10^{-2}\left(T_{H,D}\right)\right] + \left[0.4802\times10^{-5}\left(T_{H,D}\right)^2\right] + \left[-1.966\times10^{-9}\left(T_{H,D}\right)^3\right]\right\} \Big/ 28.84 \tag{10}$$

for cooling air and hot air.

Heat transfer rate between the fluid streams can be calculated considering one side as;

$$\dot{Q} = \dot{m} c_p \Delta T \tag{11}$$

Effectiveness of tested heat exchanger can be calculated as;

$$\varepsilon = \frac{\dot{Q}}{C_{\min}\Delta T} \tag{12}$$

Analytical effectiveness equation for cross flow single passing unmixed fluid heat exchangers is [32];

$$\varepsilon = 1 - \exp\left\{ \frac{NTU^{0.22}}{Cr} \left[\exp\left(-CrNTU^{0.78} \right) - 1 \right] \right\} \tag{13}$$

Above equation can be solved by iteration in case of knowing heat capacity ratio of the fluids. Hence, number of transfer units (NTU) can be calculated as;

$$NTU = \frac{UA}{C_{\min}} \tag{14}$$

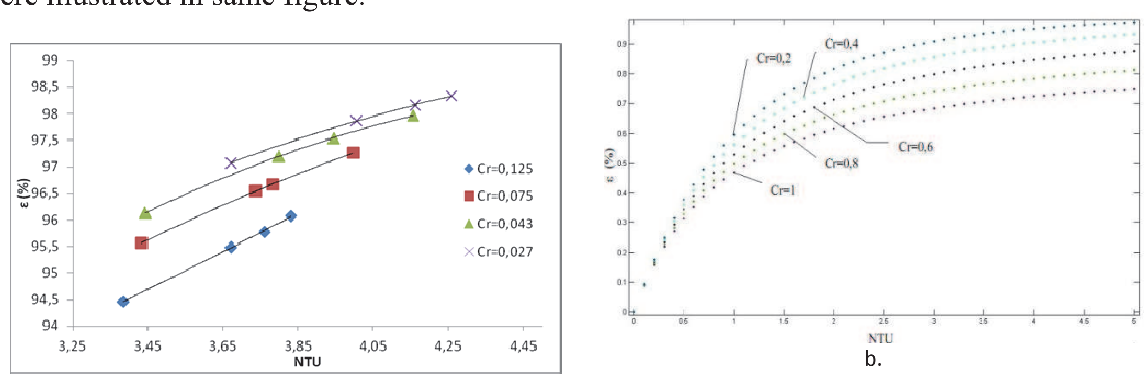
One can easily calculate overall heat transfer coefficient from (14).

Pressure loss coefficients were calculated with following equation;

$$f = \frac{\left[\left(\frac{\Delta P}{\frac{G^2}{\rho_1}} \right) - \left(1 - \sigma^2 + Kc \right) - 2\left(\frac{\rho_1}{\rho_2} - 1 \right) + \frac{\rho_1}{\rho_2}\left(1 - \sigma^2 - Ke \right) \right]}{\left(\frac{1}{4} \frac{\rho_1}{\rho_a} \frac{A}{A_c} \right)} \tag{15}$$

3. Results

With aforementioned experimental setup of this study, heat exchanger effectiveness can be determined, successfully. Fig. 9 gives a good example for this statement. In Fig. 10, intercooler effectiveness changing with NTU and Cr values were presented. Also equation (13) was calculated for five different capacity ratios for NTU values changing between 0-5 and results were illustrated in same figure.



a. b.

Figure 9. A-Intercooler ε-NTU,B-Calculated ε-NTU.

In this test interval, intercooler effectiveness changed between 94.5-98.5%. These heat capacity ratios can be seen practically in agricultural tractors. Considering acquired effectiveness values, selected intercooler can be used in agricultural tractors and similar heavy duty vehicles. The ε-NTU graphic of the intercooler is compromising with the specific literature forthe test interval considering NTU and capacity ratio values. However, effectiveness of the heat exchanger doesn't give opinion about the heat exchanger performance by itself. Cooling capacities of the heat exchanger used as intercooler is introduced in Fig. 10. Intercooler cooling capacity at this operational interval can be predicted with equation (16) which is given below. This equation was obtained by curve fitting method.

$$\dot{Q} = 4.62\times10^{-2}\,T - 13.927 \tag{16}$$

Hydraulic performance of the heat exchanger was also determined successfully with the experimental setup. Pressure drop and friction coefficient of the intercooler are provided in Fig. 11. Pressure drop changed from 5 kPa to 9 kPa which were 6 to 10% of total pressure value of charge air. Similar to the cooling capacity prediction, pressure drop in the intercooler can be calculated with equation (17) which is also a curve fitting equation and given below.

$$\Delta P = (67.146T) + 16945 \tag{17}$$

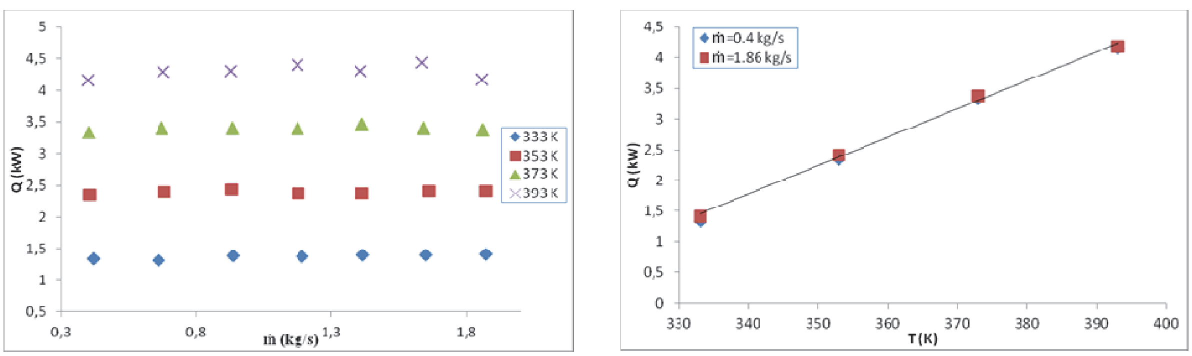


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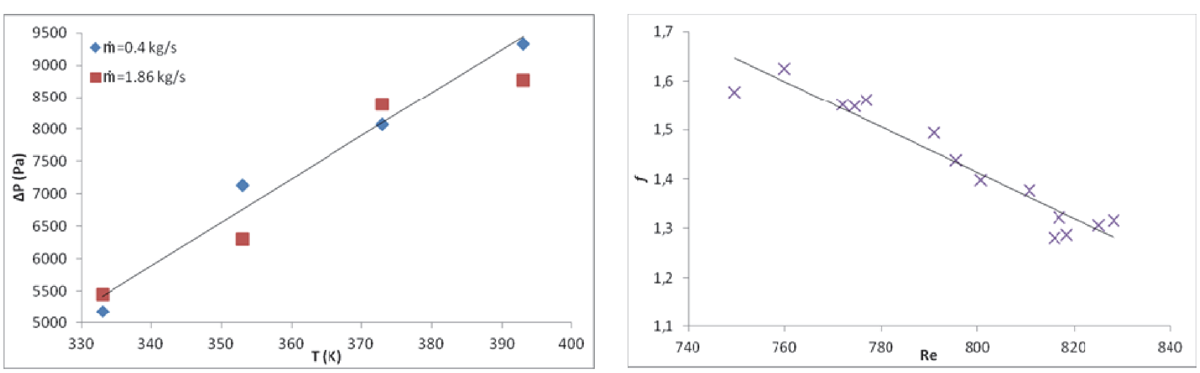
Figure 11. A-Intercooler cooling capacity a. atfour different charge air entrance temperatures changing with mass flow rate of cooling air,B-at maximum and minimum mass flow rates of cooling air changing with temperature.



a. b.

Figure 12. A- Pressure drop in intercooler changing with mass flow rate of cooling air, B-Friction coefficient changing with charge air Re numbers.

Uncertainty analysis of the performance test for the both heat exchangers is found in the range of values given in Table 2.

Table 2. Experimental uncertainties calculated according to experimental results.

$\rho_{H,O} = \pm\%0{,}03$	$\dot{m}_{H,O} = \pm\%0{,}6$	$\Delta P_{H,O} = \pm\%1$	$\varepsilon = \pm\%4$
$\rho_{D} = \pm\%0{,}1$	$\dot{m}_{D} = \pm\%0{,}4$	$\Delta P_{D} = \pm\%1$	$\dot{Q} = \pm\%4$

4. Conclusions

In this work, a louvered fin and plate intercooler characteristics and their effects to intercooling were experimentally investigated. Intercooler configuration is explained in an elaborate manner and results are presented in graphical form for cooling capacity, pressure drop in heat exchanger core and intercooler effectiveness. Details of experimental setup and method used in this study are also provided in the content of present paper.

Following conclusions and suggestions were obtained;

- Cooling capacity of the heat exchanger as intercooler was found between 1.5-4.5 kW.
- Pressure drop changed between 5 kPa to 9 kPa which were 6 to 10% of total pressure value of charge air.
- The intercooler effectiveness was found to be changing between 94.5-98.5% at 0,027-0,125 thermal capacity ratios.
- According to the experimental results, it was understood that designed and manufactured experimental setup can be effectively used for determining thermal and hydrodynamic performances of practically used intercoolers. During performance tests, hot and cold fluid flow rates can be changed, hence different operational conditions can be simulated. This process can be used for prediction of intercooler performance in field. Experimental setup can be enhanced by adding surface temperature measurement probes. By this way, heat convection coefficients can be determined.
- Obtained linear correlations can be used to predict the intercooler heat transfer and pressure drop performances at tested operational interval.
- Selected intercooler can be used in agricultural tractors and equivalent heavy duty vehicles in energy efficient manner owing to having higher effectiveness ratio.

5. Acknowledgements

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