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C. Albanakis, K. Yakinthos, K. Kritikos, D. Missirlis, A. Goulas, et al.. The effect of heat transfer on the pressure drop through a heat exchanger for aero engine applications. *Applied Thermal Engineering*, 2008, 29 (4), pp.634. 10.1016/j.applthermaleng.2008.03.034 . hal-00498972

HAL Id: hal-00498972

<https://hal.science/hal-00498972>

Submitted on 9 Jul 2010

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Accepted Manuscript

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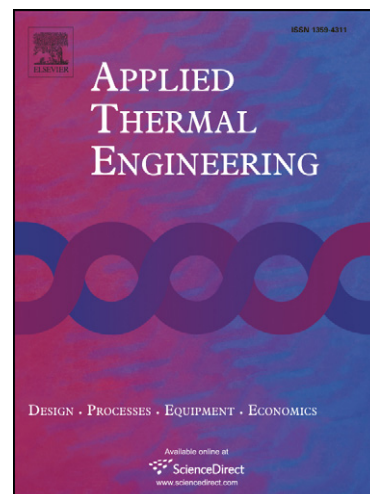
PII: S1359-4311(08)00166-X
DOI: [10.1016/j.applthermaleng.2008.03.034](https://doi.org/10.1016/j.applthermaleng.2008.03.034)
Reference: ATE 2460

To appear in: *Applied Thermal Engineering*

Received Date: 12 December 2007
Revised Date: 29 February 2008
Accepted Date: 14 March 2008

Please cite this article as: C. Albanakis, K. Yakinthos, K. Kritikos, D. Missirlis, A. Goulas, P. Storm, The effect of heat transfer on the pressure drop through a heat exchanger for aero engine applications, *Applied Thermal Engineering* (2008), doi: [10.1016/j.applthermaleng.2008.03.034](https://doi.org/10.1016/j.applthermaleng.2008.03.034)

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**The effect of heat transfer on the pressure drop through a heat exchanger
for aero engine applications**

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Abstract

This work is focused on the experimental study of the performance of a heat exchanger designed for aero engine applications. The heat exchanger is operating as a heat recuperator by taking advantage of the thermal energy of the exhaust gas of the aero engine in order to obtain a better combustion with less pollutant emissions. The experimental study has been performed in a wind-tunnel by taking detailed flow and

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thermal measurements on a 1:1 model of the heat exchanger under various operating conditions described by the hot-gas inlet mass flow-rates and its spatial direction (different angles of attack and inclination) towards the heat exchanger. The hot-gas has been modeled with preheated air. Six sets of measurements have been carried-out for different hot-gas inlet and outlet temperatures, including also isothermal measurements without any heat transfer in order to have a reference point for the pressure drop of the flow through the device. The experimental results showed that the effect of the angle of attack on the pressure drop is significant whilst the effect of the angle of inclination is negligible. Additionally, the pressure drop through the heat exchanger is greatly affected by the heat transfer.

Keywords: Heat exchanger; Pressure drop; Heat transfer; Aero engine

1. Introduction

Tubes bundle heat exchangers are widely used as coolers, heaters, evaporators, condensers, recuperators in the aerospace and power industry. The design of a heat exchanger of this type is a complex task, since many factors have to be considered, such as the thermal efficiency, the pressure drop which is imposed to the flow field and others. In the past, a lot of work has been carried-out regarding, also, the flow development through heat exchangers mainly focused on the effects of the tube shape to the flow development. For example, Boetler et al. [1], Brauer [2], Schulenberg [3], Bordalo and Saboya [4], Saboya and Saboya [5] and Rocha et al. [6] among others, have shown that regarding the pressure drop and heat transfer performance, better results could be achieved for the staggered banks of finned elliptic tubes instead of

staggered banks having circular tubes. Bouris et al. [7] presented a numerical evaluation of the heat transfer mechanism of alternative tube configurations in a heat exchanger where they concluded that the use of elliptic shape tubes provides increased heat transfer rates because the lower pressure drop imposed by the shape of the tube, allows tubes to be placed at a closer spacing. Longo and Gasparella [8] investigated experimentally the heat transfer coefficients and the pressure drop in a small-brazed plate heat exchanger operating with hydrofluorocarbon. They have found that the heat transfer coefficients show a great sensitivity on the saturation temperature. Finally, Jacimovic et al. [9] established a reliable correlation for the estimation of the pressure drop in a cross-flow over staggered tube bundle with plate fins.

In all these works, the basic aim was to correlate the thermal performance of a heat exchanger with the tube geometry, the operating fluid and other parameters with the pressure drop being the one with the greatest importance since it can affect the operation of an integrated system such as a gas-turbine or an aero engine. More specifically, regarding an aero engine, the integration of a heat exchanger can be used for various reasons, such as the cooling of some parts of the engine among others. The specific heat exchanger investigated in this work is a device designed and constructed by MTU Aero engines GmbH, under the concept of a recuperative engine. The basic idea is to use a less conventional, but more efficient, thermodynamic cycle for aircraft engines which is based on recuperation. The concept of the recuperative engine is shown in Fig. 1. Downstream of the turbine exit, the exhaust gas is directed through an installation of heat exchangers located inside the exhaust nozzle. The heat transfer is performed between the hot gas and the cooler compressor air. The latter is directed back to the combustion chamber resulting in a better combustion process and consequently lower levels of emissions and better fuel economy. Combined with other technological

engine improvements, the recuperative aero engine has the potential for fuel savings up to 20% and reductions up to 20% and 80% for CO₂ and NO_x emissions respectively.

Fig. 1. The recuperative aero engine

A detailed description of this technology can be found in the works of Broichhausen et al. [10], Wilfert and Masse [11] and in the MTU Aero Engine internet site [12]. Beyond the heat transfer efficiency of the heat exchanger and the investigation of its strength to the high and dynamical loads during the operation of the aircraft engine, a critical point of the integrated installation of the heat exchangers inside the nozzle, is the imposed pressure loss in the hot gas right after the low pressure turbine exit. Depending on its values, the characteristics of the aero engine turbocompressor can be changed and its operation can be affected as presented by Ruixian and Lixia [13]. Figure 2 shows a meridional view of the exhaust nozzle with the installed heat exchangers. Since the exhaust hot gas passes through the heat exchangers, the pressure losses must be minimum and very close to the prescribed ones downstream of the engine.

Fig. 2. Meridional view of the exhaust nozzle

The investigation of the coupling of the heat exchanger installation and the exhaust hot gas passing through them can be performed with the use of CFD. During the modeling of the integrated design, the main scope is to find the better arrangement of the heat exchangers inside the nozzle in order to have minimum pressure losses and a

maximum heat transfer rate. The numerical procedure can be performed by either solving the flow through the heat exchangers with a detailed modeling of the heat exchangers geometry or by using “pressure drop” zones, inside the flow domain, modeling the heat exchangers as porous media with a prescribed pressure drop. When the second modeling approach is performed, there is a need to derive the pressure losses through one individual heat exchanger for various flow conditions and primarily using experimental measurements. Yakinthos et al. [14] have presented a detailed analysis for the optimization of the design of such a recuperative heat exchangers by presenting a pressure drop law that has been already derived in the work of Missirlis et al. [15] for this specific type device, used in aircraft engines. Although the work is a detailed description of the steps needed to conclude to a pressure drop law, it cannot be characterized as an integrated work since the pressure drop law does not account for the heat transfer effects. In other words, the pressure drop law was derived by performing isothermal measurements on the heat exchanger. This law was applied for the modeling of the flow through the exhaust nozzle but without any heat transfer, meaning that the hot gas has been modeled as a fluid having ambient temperature. It is true that this modeling procedure is far from the real operating conditions of the aircraft engines, but in any case, since the computational results have been validated by using the data obtained from measurements carried-out in laboratory conditions (i.e. ambient temperature), it was a first step for the validation of the pressure drop model without taking into account the heat transfer effects.

In order to proceed to a more accurate modeling of the flow, a new measurement campaign has been initiated in the Laboratory of Fluid Mechanics & Turbomachinery. The basic aim, now, was to investigate and to relate closely the heat transfer mechanism with the pressure drop law through one individual heat exchanger. This investigation

has the potential to provide a more accurate law for the pressure loss that will relate all the critical parameters affecting the flow development through the heat exchanger such as the inlet conditions together with the heat transfer conditions. The new and improved pressure drop law will allow the modeling of the real operating conditions of the aircraft engine since it will be able to take into account the heat transfer effects.

2. Experimental setup

The heat exchanger is constructed by special profiled tubes (having an elliptic geometry) in order to achieve the, possibly, minimum pressure drop in the exhaust nozzle system and it consists of two manifold tubes, Fig. 3. The flow from the high pressure compressor enters the upper tube from both sides and is distributed into the U-shaped profiled tubes, which are brazed into the manifold tubes and form the core of the heat exchanger. The lower manifold tube (the collector tube) collects the preheated air and leads it back to the combustion chamber. The hot exhaust gas from the low-pressure turbine flows upwards through the heat exchanger and is cooled down while heating-up the air inside the profiled tubes.

Fig. 3. The heat exchanger and the U-shaped elliptic profiled tubes

For the experimental measurements, only a small part of the heat exchanger has been constructed. This part refers to the straight part of the device, excluding the bow-parts as shown in Fig. 3. The model heat exchanger was constructed in a 1:1 scale and consists of 144 elliptic straight tubes, placed in a 4-3-4 staggered arrangement (also shown in Fig. 3). The model heat exchanger is placed in wind-tunnel with a rectangular

cross-section area which is equal to $0.06 \times 0.076 \text{m}^2$. Figure 4 shows the test-rig with the heat-exchanger placed in the wind-tunnel. Regarding the cross-section area, a first computation has been made in order to ensure that no blockage effects (due to the developed wind-tunnel wall boundary layers) would occur that could potentially affect the measurements for the total range of the inlet flow velocities. The hot gas has been modeled with preheated air. For the heat transfer, cold water was circulated inside the heat exchanger tubes.

Fig. 4. The test-rig with the heat exchanger

A schematic diagram of the test-rig is shown in Fig. 5, where the air heater, the wind-tunnel, the chiller, the water tanks and the heat exchanger model are also shown, together with the measurement positions upstream and downstream of the heat exchanger.

Fig. 5. The experimental test-rig

The latter have been chosen to be 40mm upstream the model and 30mm downstream the model. The two positions were the same for both the velocity and pressure measurements. Especially for the cold medium (water), a closed loop was used with two water tanks connected with manifolds to the heat exchanger. For the water circulation, two small pumps were used, a triode mixing electrovalve and an electromagnetic flow-meter. In order to have a comparative relation for the pressure drop law with and without heat transfer phenomena, two sets of pressure drop measurements have been carried-out, the first with no heat transfer (isothermal) and the

second one with heat transfer (anisothermal). Additionally, regarding the inlet conditions of the heated air, a variety of imposed conditions have been used which were strongly related to the hot gas flow development inside the exhaust nozzle and to the manner that it enters the heat exchanger installation. In a preliminary, simplistic CFD numerical modeling [16] it was shown that the hot gas does not have a specific and preferable direction as it enters the heat exchanger, thus, it has been decided to test as many as possible inlet conditions. During the experiment, this has been modeled by applying various inlet flow angles for various mass flow rates. The various inlet conditions regarding the flow angle towards the heat exchanger have been obtained with the use of 49 specific inlet ducts, which led to a range of combined angles of attack, a_a and inclination, a_i , as defined in Fig. 6.

Fig. 6. Definitions of angle of attack and inclination

The values of the inlet air velocity were specified through the adjustment of the wind-tunnel mass flow rates, the latter being controlled by a small centrifugal blower installed at the exit of the wind-tunnel. For the anisothermal measurements, the air temperatures were set to specific values according to the operating conditions and were controlled by the adjustment of the air heater power. Additionally, the temperature difference through the heat exchanger was controlled by the chiller power. Before starting the measurements for each set corresponding to the anisothermal conditions, a certain time interval was needed in order to have a stable and constant temperature difference through the heat transfer process between the heat exchanger and the hot air.

2.1 Measurement devices and accuracy

The pressure and velocity measurements were carried-out using a 3-hole probe and a Pitot-static tube, capable to carry out measurements for the total and static pressure and, also, for the velocity and flow angle. The 3-hole probe and the Pitot tube have an accuracy of 2% referring to the dynamic pressure measurements, which leads to a maximum error of 2Pa. The 3-hole probe is connected to a digital manometer, which has an accuracy of 0.05% resulting-in a maximum error of 2Pa. The accuracy in the angle of attack and angle of inclination has been estimated at $\pm 1^\circ$ as presented at the work of Missirlis et al. [15] and the calibration procedure was the same with the one presented by Morisson et al. [17] and Yasar and Carpinioglou [18]. The accuracy of the positioning mechanism for the both the 3-hole probe and the Pitot tube is 0.25mm. During the pressure drop measurements it was necessary that the measurements would be carried out in such a way that would correspond to the average behavior of the heat exchanger. For this reason, and since the heat exchanger geometry is composed of numerous tubes with elliptic profile creating high and low velocity regions downstream the heat exchanger, it was necessary to proceed to a measurement step dependency study so as to find out and use the most appropriate measurement step.

At the initial stage of this work, preliminary measurements were carried out with varying measurement steps at the normal direction. For the inlet position, where the flow is expected to be relatively uniform, the measurement steps of 5, 10 and 15mm were used. However, for the outlet position, where the flow is in the wake regions of the elliptic tubes, much smaller values for the measurement step were used i.e., 0.5, 1, 1.5, 2, 2.5, 3 and 4 mm. The comparison of the average flow quantities, total and static pressure and velocity components, as they were derived for each case with a different

measurement step showed that in order to achieve values independent of the measurement step, the measurement step at the inlet position should be selected equal or less than 10mm while the measurement step at the outlet position should be selected equal or less than 2.5mm. Thus, the measurement step for the inlet position was chosen equal to 5mm while the one corresponding to the outlet position was chosen equal to 2mm. The selection of these step values provided an accuracy corresponding to less than 1Pa for the total and static pressure values and to approximately 0.01m/s for the velocity magnitude.

For the air temperature measurements, a K-type thermocouple was used with an error of $\pm 1.2^{\circ}\text{C}$. For the water temperature measurements, a Platinum Resistance Thermometers PT100 thermocouple was used with a resistance of 100 Ohms at 0°C and 138.4 Ohms at 100°C . The thermocouple provides a linear relationship between the resistance and the water temperature leading to an error of $\pm 0.3^{\circ}\text{C}$. The volumetric flow rate was measured with a MAGFLOW 5100W flowmeter with an average error of 0.5% for the range of the cold water flow rates applied to the experiment.

During the preliminary measurements, some problems appeared with condensation of the air on the external surface of the heat exchanger tubes caused by the ambient air humidity. Since the condensation process could affect the quality of the measurements, two dehumidifiers were used in order to make sure that no condensation would occur on the heat exchanger tubes. For all the sets of measurements, the relative humidity was ranging from 34 to 39%.

3. Experimental results

The aim of this work is the derivation of the pressure drop through the heat exchanger by investigating also the effect of the heat transfer on the pressure drop. The experiments were carried-out without heat transfer (isothermal study) and with heat transfer (anisothermal study). For the isothermal study, measurements for four sets of different inlet air temperature have been carried out, corresponding to $T_{\text{air,in}}=20^{\circ}\text{C}$, $T_{\text{air,in}}=40^{\circ}\text{C}$, $T_{\text{air,in}}=60^{\circ}\text{C}$, $T_{\text{air,in}}=80^{\circ}\text{C}$ ($T_{\text{air,in}}=T_{\text{air,out}}$) and for a range of mass flow rates starting from 0.015kg/s up to 0.037kg/s. Only in the isothermal model with $T_{\text{air,in}}=20^{\circ}\text{C}$ (Fig. 7) the maximum mass flow rate is 0.072kg/s. Since the fluid velocity varies along its path it is necessary to select a reference velocity in order to define the Reynolds number. According to the suggestions of Umeda and Yang [19], the reference velocity U_{max} is based on the minimum free area available for fluid flow A_{min} per unit length of the tube length D , regardless if the minimum area occurs in the transverse or diagonal opening. Thus, the Reynolds number is defined as: $\text{Re} = \frac{U_{\text{max}} D}{\nu}$ where ν is the fluid kinematic viscosity. The laminar-flow range is for $\text{Re} < 200$ and the turbulent-flow range for $\text{Re} > 6000$ with the transition-flow range being between $200 < \text{Re} < 6000$ [19]. For the experimental measurements of the present work, the corresponding Reynolds numbers are in the range of 280 and 921 and, consequently, the flow is within the transitional regime.

The anisothermal measurements have been carried-out for the same inlet mass flow rate range and for two air temperature differences, $T_{\text{air,in}}=60^{\circ}\text{C}$ and $T_{\text{air,out}}=31^{\circ}\text{C}$, $T_{\text{air,in}}=80^{\circ}\text{C}$ and $T_{\text{air,out}}=33^{\circ}\text{C}$. The temperature of the cold water was varying from 9 to 17°C depending on the air mass flow rate and the selected outlet air temperature. The

experimental conditions are listed and presented in Table 1. For all cases of measurements, isothermal and anisothermal, the evolution of the total and static pressure drop are rather similar because the dynamic pressure is very small. Therefore only the total pressure curves will be presented.

Table 1. Experimental conditions of air and water

3.1 Isothermal measurement conditions

Figure 7 shows the total pressure drop for the isothermal case with an inlet air temperature equal to 20°C and for seven different angles of attack in the ranging from 0° up to 60°. For this set of measurements, the angle of inclination was set to zero. In the same figure, the curves represent the polynomial fitting of the measured data for each case. From these measurements, it is clearly shown that for the same inlet air mass flow rate, there is an increase of the pressure drop and this can be attributed to the effect of the angle of attack on an elliptic-shaped airfoil. As already written, each tube of the heat exchanger has an elliptic shape. As the air-flow enters with a non-zero angle of attack, there is a significant increase of the pressure losses. As the flow continues to pass around the tubes, it tends to be straightened by the passages through the heat exchangers, but in any case, in the region of the first row of tubes, the flow is starting to form downstream large wakes, which increasingly affect the pressure drop through the heat exchanger. This observation is supported by our first preliminary CFD modeling, Missirlis et al. [16], where the flow through the heat exchanger has been investigated using some preliminary but very detailed isothermal measurements on an individual heat exchanger package. The fitting curves, are based on second order polynomials,

having a general expression of the form: $\Delta p = a\dot{m}^2 + b\dot{m} + c$ where Δp refers to the pressure drop, \dot{m} to the inlet mass flow rate and a , b , c to the coefficients from the polynomial curve fitting.

Fig. 7. Total pressure drop for isothermal-1 case for seven angles of attack and 0° angle of inclination

During the experiments, the effect of the angle of inclination has been also investigated by performing measurements for the isothermal case of the 20°C having an angle of attack equal to zero. Figure 8 show the variation of the total pressure drop, for various inlet air mass flow rates and for seven angles of inclinations ranging from 0° up to 60°. Additionally, in the same figure, the curve representing the polynomial fitting of the pressure drop variation for the zero angle of attack and angle of inclination combination is also shown. As it can be seen, the measurements for all the angles of inclination are concentrated in a region very close to the curve representing the 0°-0° distribution, leading to the conclusion that the effect of the angle inclination to the pressure drop is negligible.

Fig. 8. Total pressure drop for isothermal-1 case for seven angles of inclination and 0° angle of attack

In the previous section, we concluded that the effect of a non-zero angle of attack is quite significant since there is an increase of the pressure drop. Alternatively, the effect of the inlet air temperature to the pressure drop, having a constant inlet angle,

has been also investigated during the isothermal studies. In Fig. 9 the total pressure drop for the isothermal model and for the 0° - 0° angle combination, when the inlet air has four different temperatures (20° , 40° , 60° and 80°) together with the polynomial fitting curves, is also shown. In these plots, the effect of the air temperature is clearly shown: a larger value of temperature leads to a larger pressure drop. This behavior can be explained by two ways: in a first approach, the effect of the temperature on the pressure drop can be implicitly related to the higher values of the air dynamic viscosity leading to higher shear stress on the tubes surface, thus higher values for the aerodynamic drag and, by consequence, to a larger pressure drop. This explanation wouldn't be valid if the flow was turbulent, because in turbulent flows the effect of turbulent stresses would dominate. Since the Reynolds numbers in the experiments are small, although the flow is not fully laminar, the turbulence level is small. In an alternative approach, the effect can be related also to the lower values of the air density, leading to higher air velocities in order to always maintain a constant mass flow rate. This would imply an increase of the dynamic pressure, but from the analysis of the experimental results was found that the increase of the dynamic pressure is very small, almost 1Pa. Since the pressure drop is varying with the square of the air velocity, higher velocities lead to higher pressure losses.

Fig. 9. Comparison of the total pressure drop for all isothermal cases for 0° angles of attack and inclination

3.2 Anisothermal measurement conditions

Since it has been shown that the effect of the angle of inclination is negligible, the anisothermal measurements have been carried out by investigating only the effect of the angle of attack. The total pressure drop for the 60°C to 31°C and 80°C to 33°C inlet and outlet air temperatures are shown in Fig. 10 and Fig. 11 respectively. Here, the angle of attack varies from zero degrees up to sixty degrees. In these plots the polynomial curve fitting lines are also shown. For each test case (based on the temperature difference), the pressure drop is larger as the angle of attack increases.

Fig. 10. Total pressure drop for anisothermal case-1, for seven angles of attack and 0° angle of inclination

Fig. 11. Total pressure drop for anisothermal case-2, for seven angles of attack and 0° angle of inclination

In order to estimate the effect of the net temperature difference for an indicative angle of attack, Fig. 12 and Fig. 13 show the comparison for the total pressure drop for the two anisothermal cases when the hot air enters with zero and forty degrees angle of attack, having a zero angle of inclination for both cases. From these figures, the net effect of the heat transfer is clearly shown. Larger temperature differences lead to larger pressure losses, although it seems that for the higher angle of attack the pressure drop for both temperature differences is nearly the same.

Fig. 12. Comparison for the total pressure drop for the two anisothermal cases for 0° angles of attack and inclination

Fig. 13. Comparison for the total pressure drop for the two anisothermal cases for 40° angle of attack and 0° angle of inclination

3.3 Intercomparison between isothermal and anisothermal measurements

So far it has been shown that two major parameters govern the pressure losses of the flow through the heat exchanger: the angle of attack and the inlet temperature of the hot air. The effect of the heat transfer mechanism, compared to the cases when no heat transfer occurs, can be illustrated by the following plots for the total pressure presented in Fig. 14 and Fig. 15. Figure 14 refers to all the cases investigated (iso- and anisothermal) having a zero angle of attack and inclination while Fig. 15 to the cases having 40° angle of attack and a zero angle of inclination. The polynomial curve fitting lines are also shown. For both figures, the general trend is that the higher angle of attack together with the higher inlet temperature leads to larger values of pressure drop. On the other hand, a detailed investigation of these data shows that there is also an effect of the temperature difference when heat transfer is on (clearly shown in the previous section). For the isothermal case with inlet air temperature equal to 80°C , the pressure drop is larger than the one of the anisothermal case, which has the same inlet air temperature but the air is cooled. For both angles of attack, i.e. zero and 40° , when the air has a constant high inlet temperature the largest values for the pressure drop are presented. This holds also for the corresponding isothermal and anisothermal cases with inlet air

temperature equal to 60°C. This behavior can be related, in the same manner as it was done for the isothermal cases, to the variation of the density and the viscosity. As a general remark, it is clearly shown that air-cooling leads to lower pressure drop. This has a direct effect on the operation of the heat exchanger inside the exhaust nozzle and the imposed pressure drop downstream the turbine of the aero engine.

Fig. 14. Intercomparison for the total pressure drop for the four isothermal cases and for the two anisothermal cases for 0° angles of attack and inclination

Fig. 15. Intercomparison for the total pressure drop for the four isothermal cases and for the two anisothermal cases for 40° angle of attack and 0° angle of inclination

4. CONCLUSIONS

In this work, an experimental attempt was made to correlate the pressure drop through a specific type heat exchanger for aero engine applications with two main parameters: the inlet temperature and the angle of attack and inclination of the hot-air mainstream direction. These parameters have been investigated firstly by carrying-out isothermal measurements (with no heat transfer). Additionally, the effect of heat transfer on the pressure drop has been investigated, by carrying-out anisothermal measurements (with heat transfer). All these parameters are of critical importance since the heat exchanger is designed to operate as a recuperator downstream the aero engine by cooling the hot exhaust gas. Especially for the angles of attack and inclination, their role is very important since the flow development downstream the engine does not have a

prescribed direction and enters in the heat exchanger by having various angles of attack and inclination. The detailed measurements showed that:

- 1) There is a direct correlation of the pressure drop with the inlet air temperature. Higher temperatures lead to higher pressure drops, for the same inlet air mass flow rate.
- 2) The angle of attack plays also a major role to the pressure drop since; higher angles of attack lead to higher pressure drops.
- 3) The heat transfer rate regarding the cooling of the hot air is also a significant parameter. As the temperature difference between the hot and the cooled air increases, the pressure losses are also increasing.

Based on these conclusions, the next step is to derive an integrated law for the pressure drop through this type of heat exchanger, which will be able to correlate these three main parameters with the flow parameters (temperature, density, inlet velocity and viscosity). The derivation of such a pressure drop will allow the modeling of the operation of the heat exchangers inside the exhaust nozzle by creating “pressure drop zones” instead of modeling in detail every single flow-passage and the whole geometry of the heat exchanger.

Acknowledgments

This paper is part of the 03ED research project, implemented within the framework of the “Reinforcement Programme of Human Research Manpower” (PENED) and co-financed by National and Community Funds 25% from the Greek Ministry of Development-General Secretariat of Research and Technology and 75% from E.U.-European Social Fund. A part of this project has been also funded by MTU Aero Engines GmbH.

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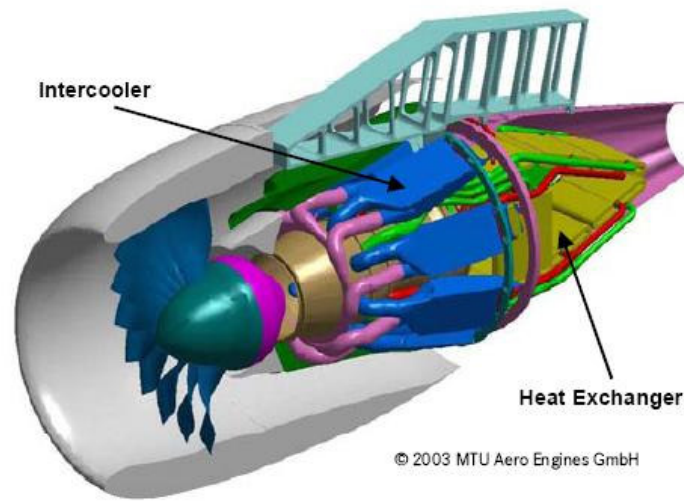


Fig. 1. The recuperative aero engine

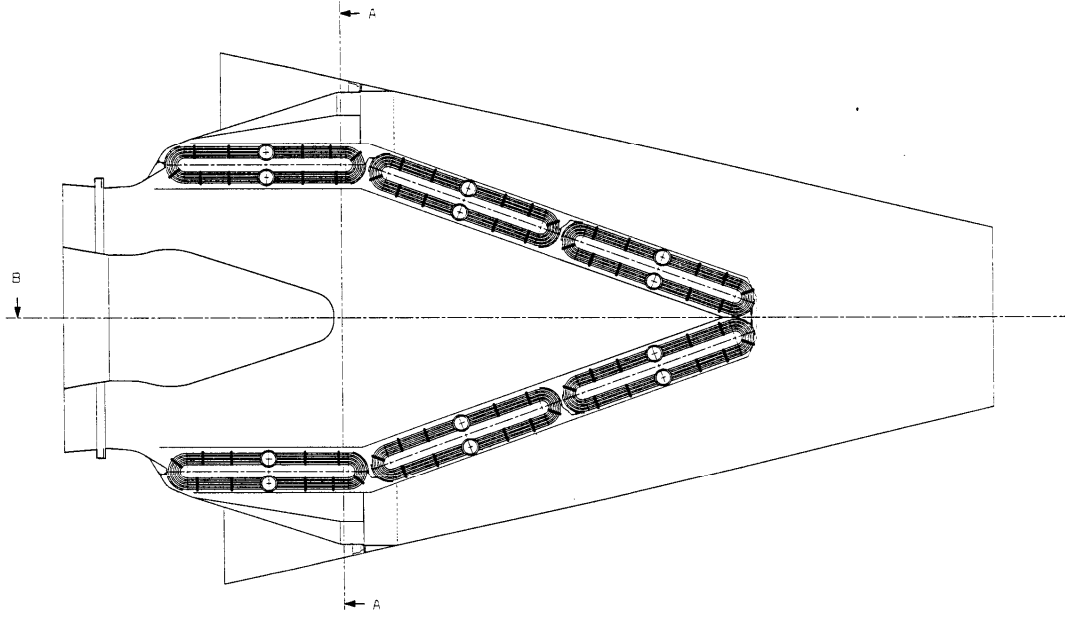


Fig. 2. Meridional view of the exhaust nozzle

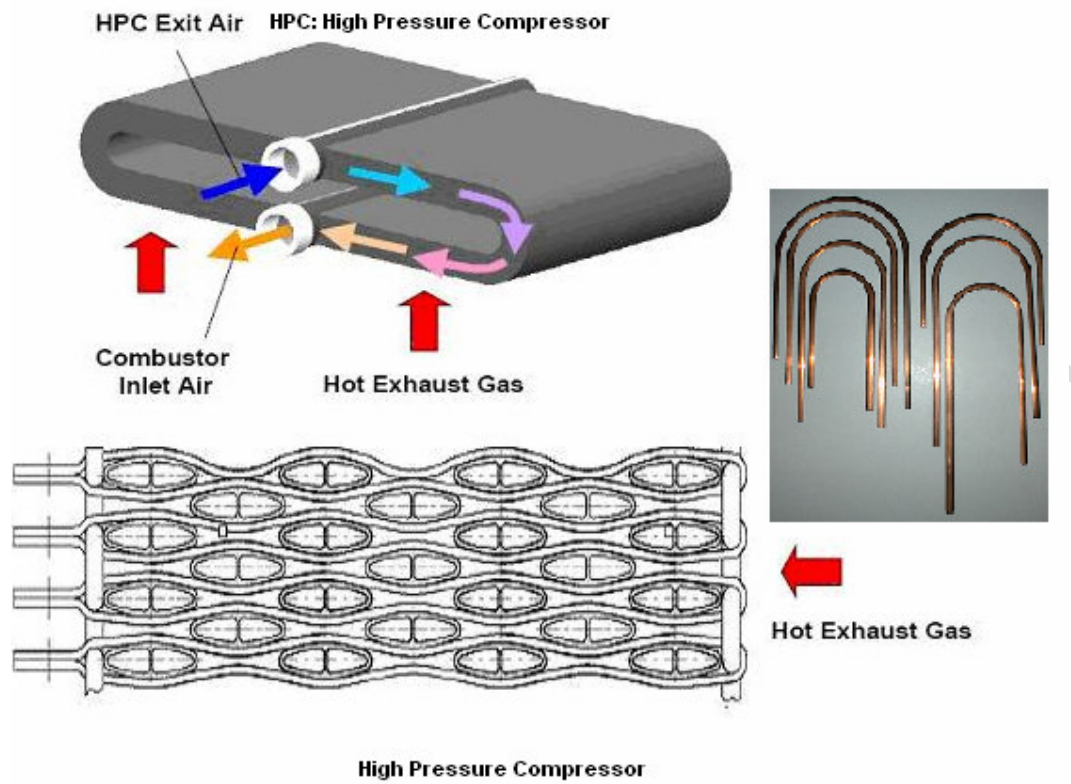


Fig. 3. The heat exchanger and the U-shaped elliptic profiled tubes



Fig. 4. The test-rig with the heat exchanger

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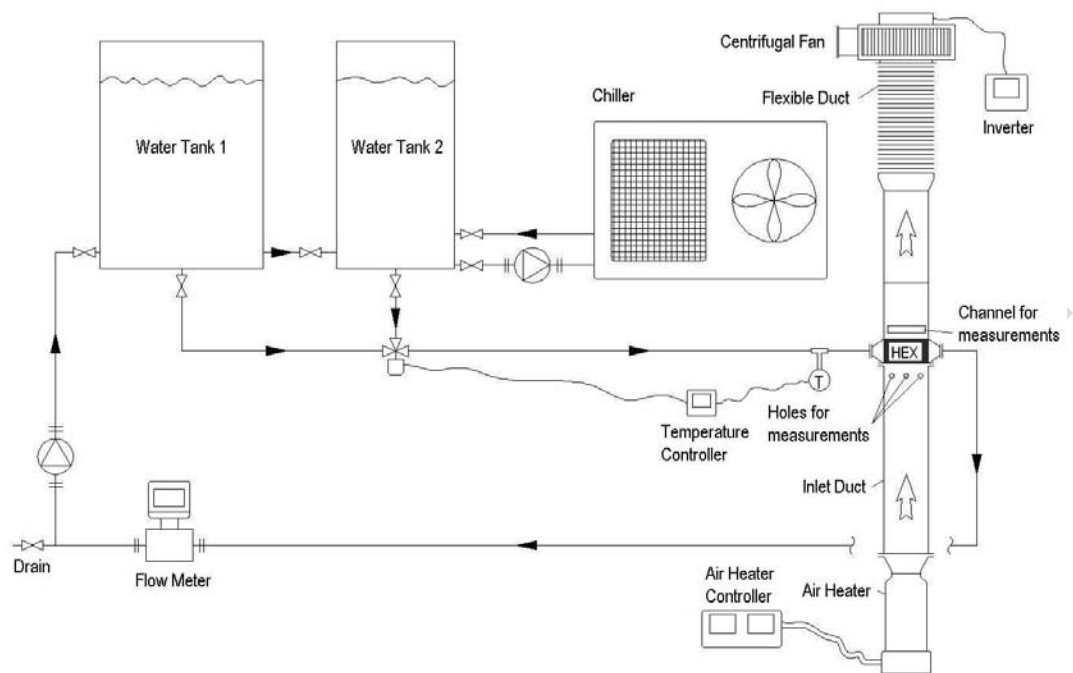


Fig. 5. The experimental test-rig

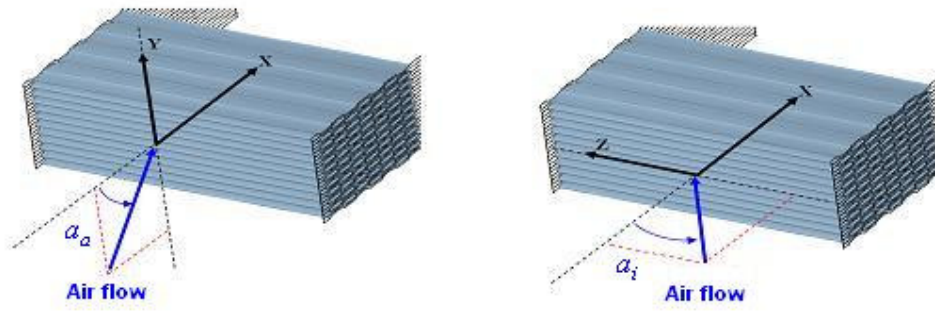


Fig. 6. Definitions of angle of attack and inclination

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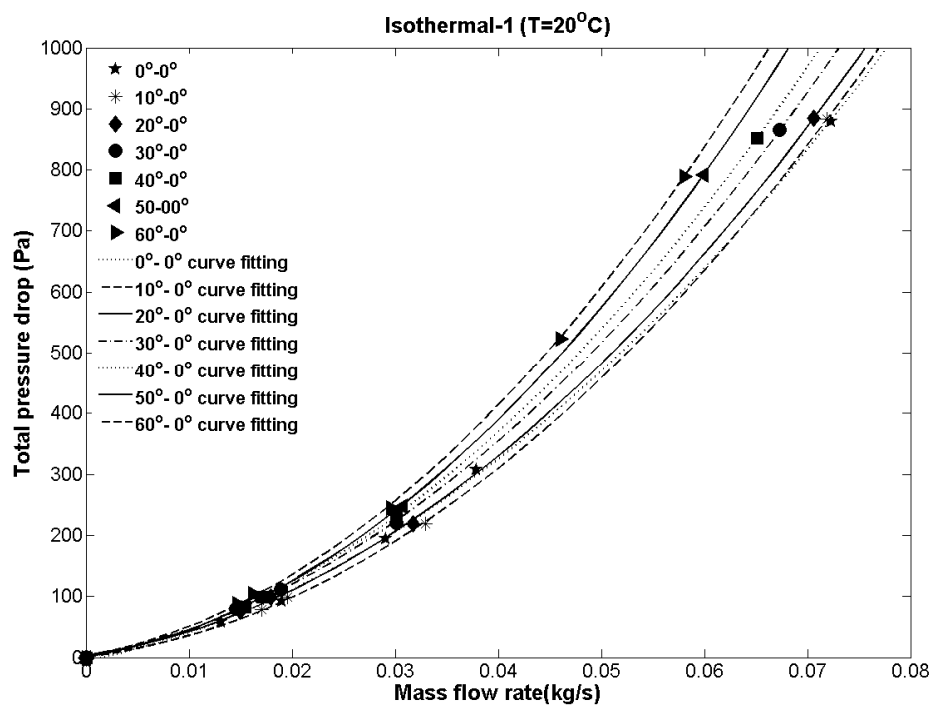


Fig. 7. Total pressure drop for isothermal-1 case for seven angles of attack and 0° angle of inclination

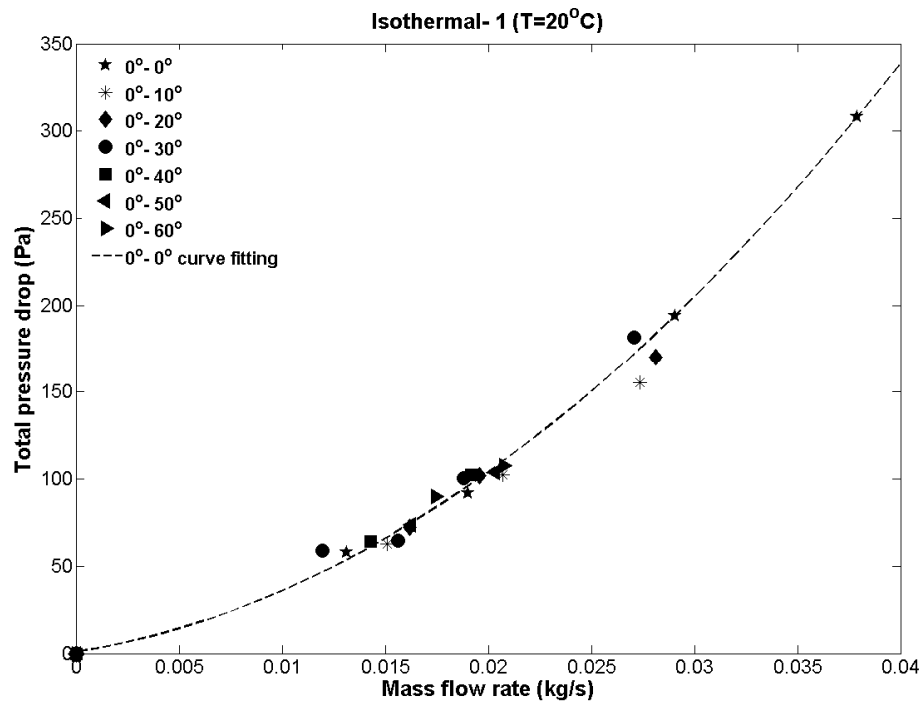


Fig. 8. Total pressure drop for isothermal-1 case for seven angles of inclination and 0° angle of attack

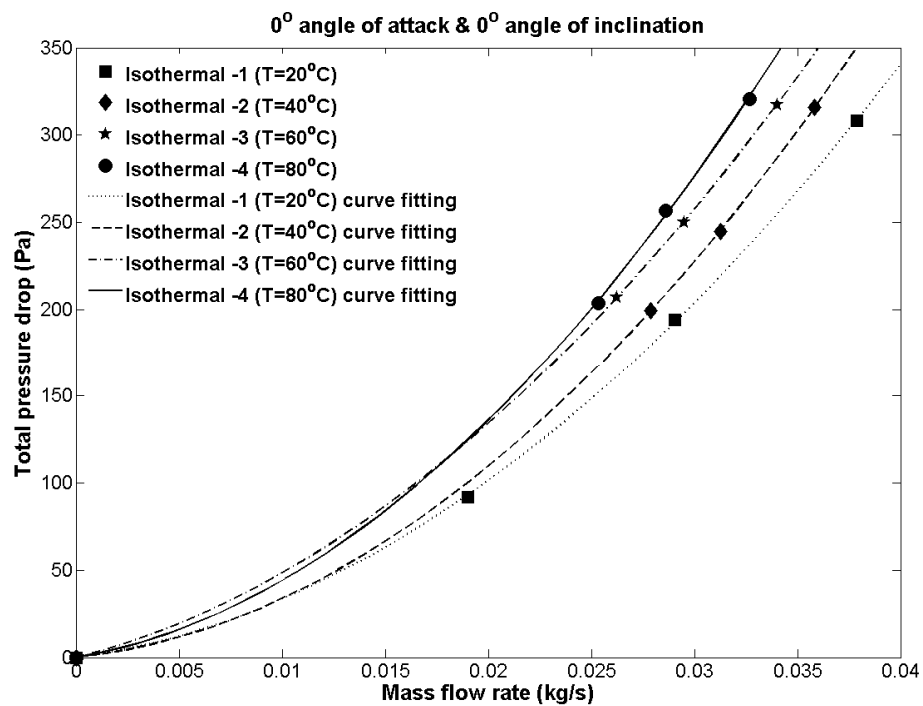


Fig. 9. Comparison of the total pressure drop for all isothermal cases for 0° angles of attack and inclination

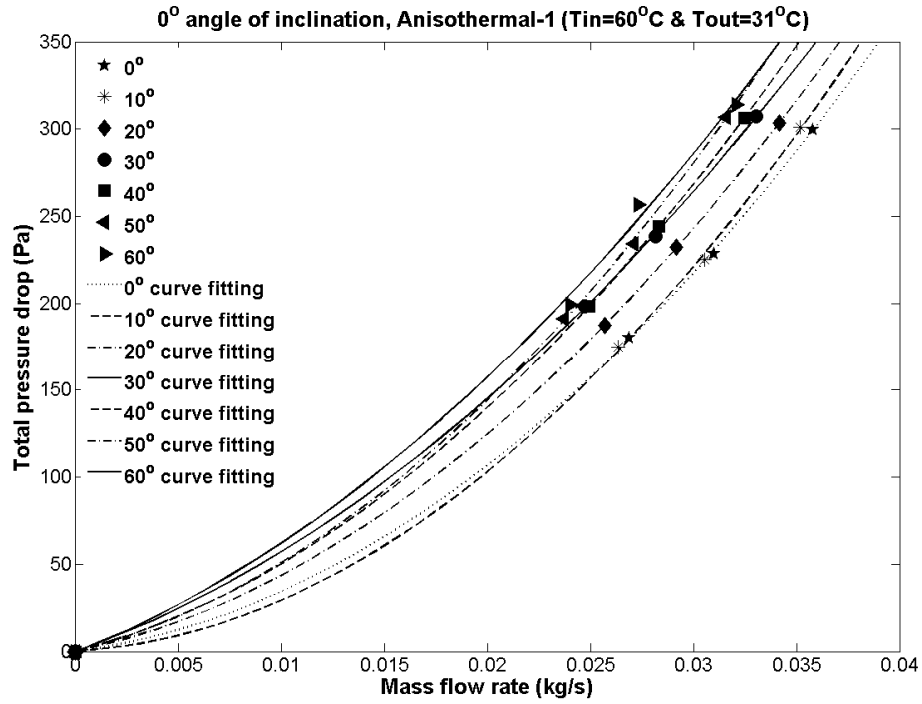


Fig. 10. Total pressure drop for anisothermal case-1, for seven angles of attack and 0° angle of inclination

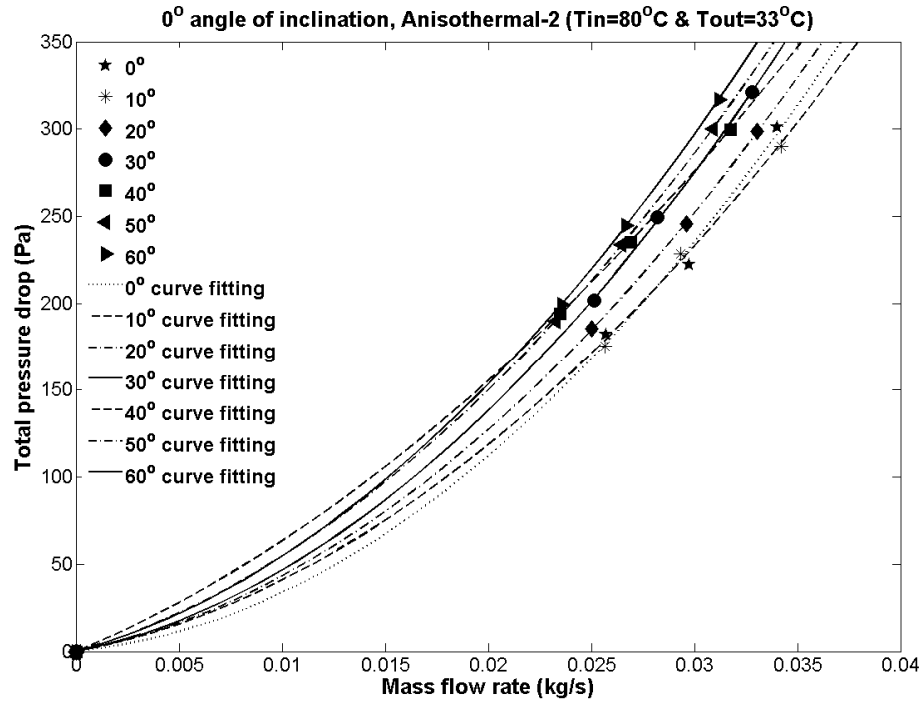


Fig. 11. Total pressure drop for anisothermal case-2, for seven angles of attack and 0° angle of inclination

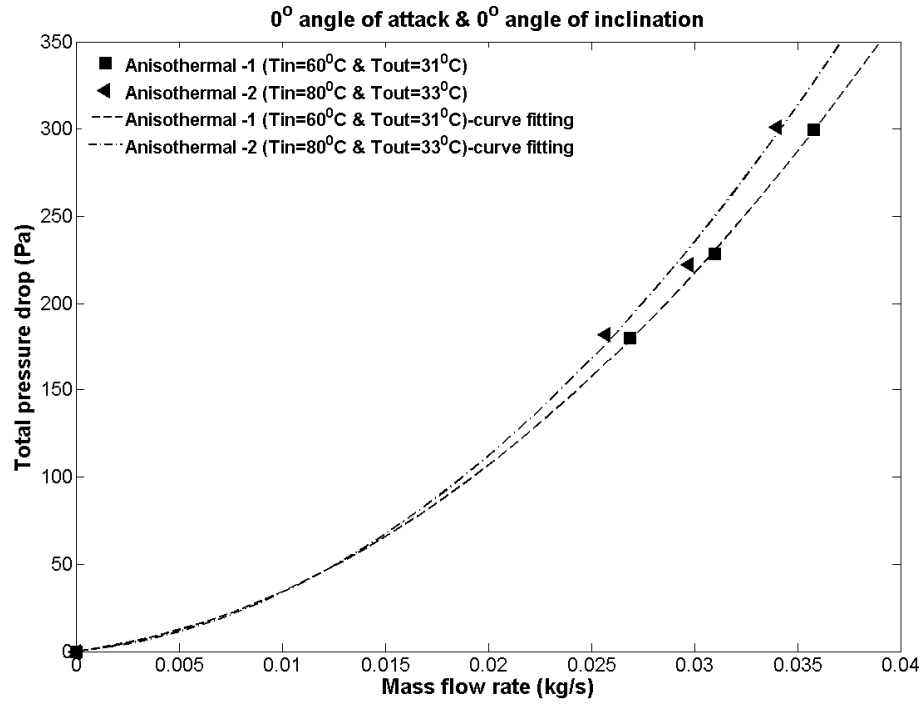


Fig. 12. Comparison for the total pressure drop for the two anisothermal cases for 0° angles of attack and inclination

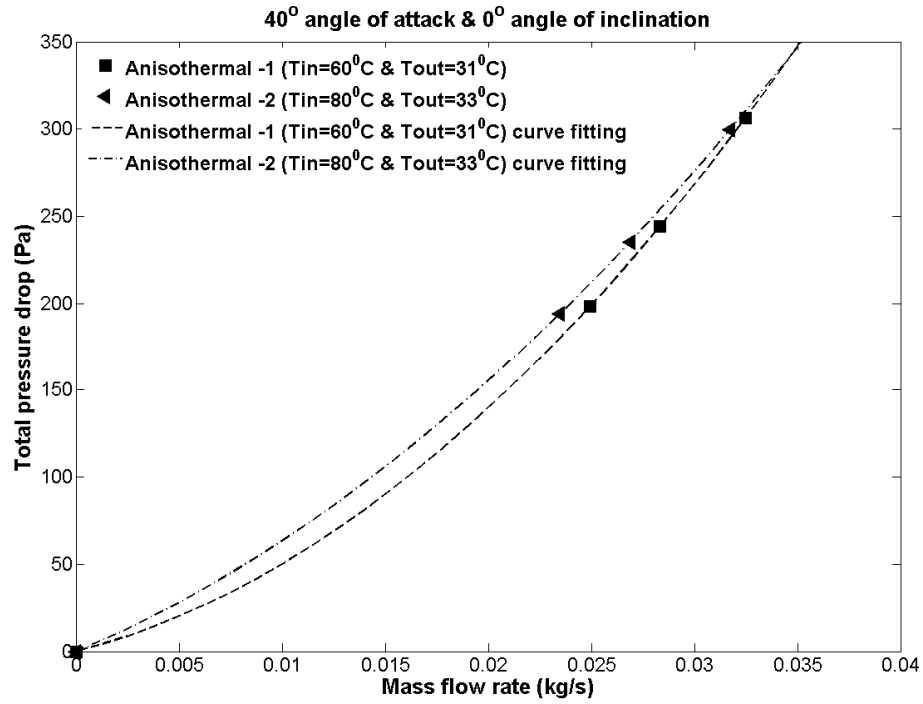


Fig. 13. Comparison for the total pressure drop for the two anisothermal cases for 40° angle of attack and 0° angle of inclination

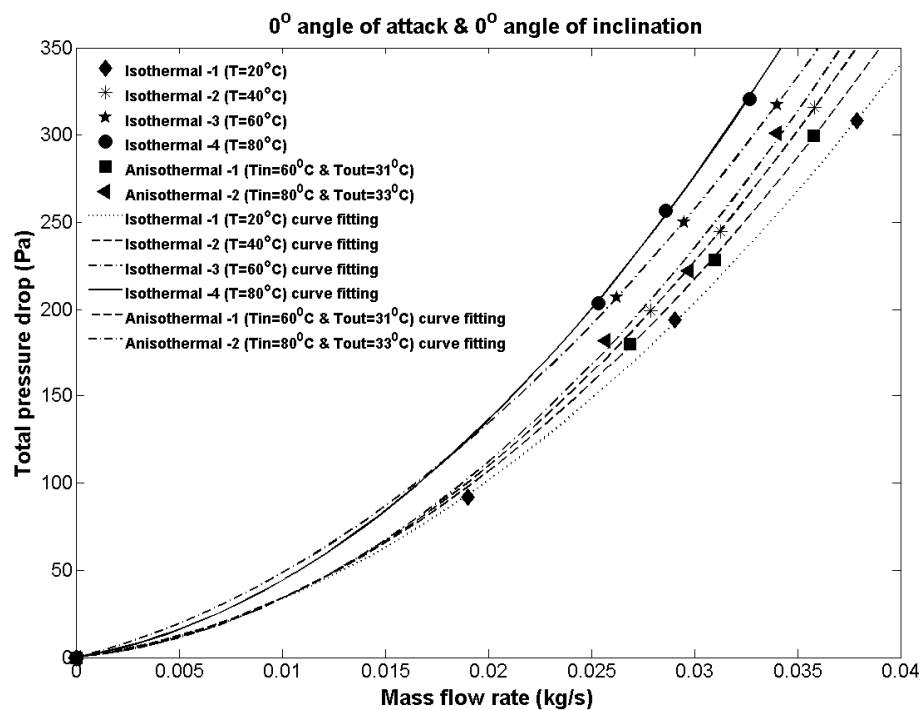


Fig. 14. Intercomparison for the total pressure drop for the four isothermal cases and for the two anisothermal cases for 0° angles of attack and inclination

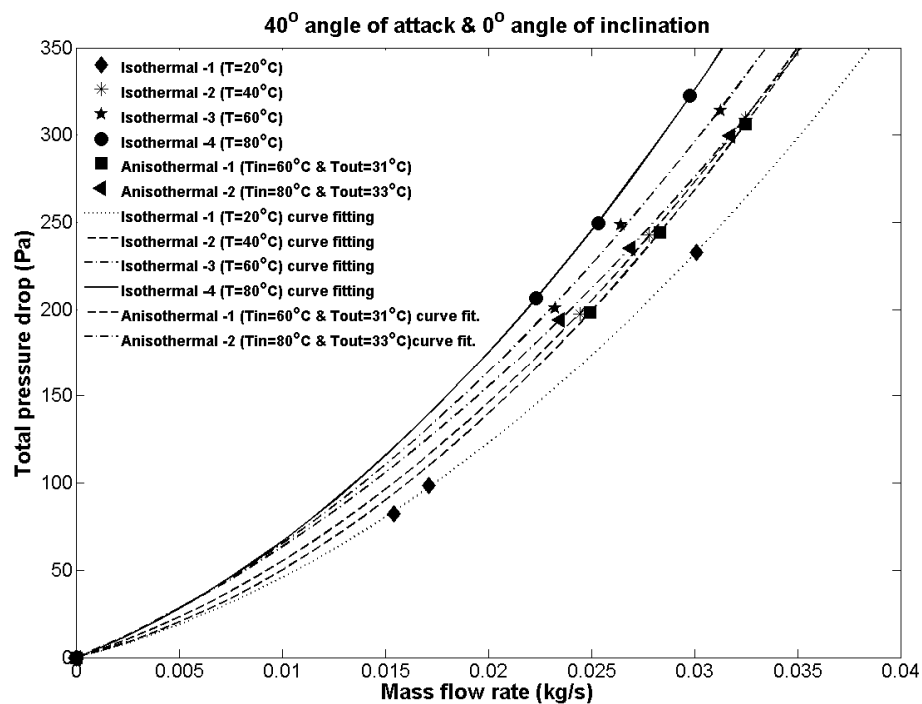


Fig. 15. Intercomparison for the total pressure drop for the four isothermal cases and for the two anisothermal cases for 40° angle of attack and 0° angle of inclination

Measurements Cases	$T_{\text{air,in}}$ (°C)	$T_{\text{air,out}}$ (°C)	ΔT (°C)	Air mass flow range (kg/s)	$T_{\text{water,in}}$ (°C)	Water mass flow range (lit/h)
Isothermal	20	20	0	0.015-0.037	-	715-910
	40	40	0		-	
	60	60	0		-	
	80	80	0		-	
Heat Transfer	60	31	29		14-17	
	80	33	47		9-12	

Table 1. Experimental conditions of air and water