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Preliminary Investigation of the Heat Transfer and Pressure Drop of a Plate-Fin Heat Exchanger for Aero Engine Applications

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The present work describes the experimental setup for the investigation of the performance characteristics of condenser specimens. These test specimens will provide valuable data to be used in the sizing of condensers which will be integrated in the Water-Enhanced Turbofan (WET) engine. The WET engine concept uses steam in the combustion chamber to lower the temperature and increase the engine efficiency. The condenser aims to convert the steam present in the core flow into liquid while transferring heat energy to the bypass flow. This recovered liquid water enables the WET cycle to operate as a self-sustained system, eliminating the need for a large external water reservoir. The design and selection of appropriate condensers is of significant importance since the flow conditions include two-phase flow and condensation in the presence of non-condensable gases. Thus, a dedicated experimental setup was designed to be able to reproduce the desired flow conditions to test the performance of the condensers. The condenser testing is focused on the heat transfer performance and pressure drop, providing Colburn factor j and Fanning friction factor f correlations for different condenser designs and evaluating the optimum design based on the optimal characteristics. The tests intended to measure multiple condenser designs, with different flow conditions, such as temperature, pressure, water to air mass fraction, mass flow rates, Reynolds number and orientation, to give an overall picture of each condenser performance. The experimental tests confirmed that the test rig design can be used for conducting accurate heat transfer and pressure drop measurements in two-phase flow heat exchangers.

1. Introduction

The WET (Water-Enhanced Turbofan) engine concept lies in its potential to enhance thermal efficiency and reduce greenhouse gas emissions in aviation, as thoroughly explained in Kaiser et al. (2022). By injecting water into the engine cycle, it has the potential to lower combustion temperatures, reduce nitrogen oxide (NO_x) emissions, and improve fuel efficiency. This results in a more environmentally friendly propulsion system that aligns with the industry goals for carbon neutrality and sustainable aviation. The incorporation of a condenser into the WET cycle is critical, as it enables water recovery from the exhaust, allowing for a closed-loop system that minimizes additional water supply.
The primary objective of the condenser design is to achieve the fuel burn target set by the baseline engine cycle. Its heat transfer performance, pressure drop, weight, and integration impact play a crucial role in determining the overall engine efficiency and fuel consumption. Similar work on heat exchanger optimization for aero engine applications was done in Salpingidou et al. (2017). A well-functioning condenser not only ensures sufficient water recovery but also minimizes penalties associated with pressure losses and added weight. The difference between a functional and a fully optimized condenser design strongly impacts fuel burn. The effectiveness of

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the condenser on the cold side affects the amount of flow required to transfer a certain amount of heat and its size. From a design perspective, low pressure losses are necessary to be targeted on the bypass flow and on the core flow, emphasizing the need for low pressure drop while maintaining high heat transfer, as also presented in the work of Germakopoulos et al. (2018). Achieving this balance requires careful consideration of the flow channel geometry, fin design and fluid properties. The experimental setup was therefore designed to assess the performance of crossflow plate-fin heat exchangers with different geometrical characteristics on both the hot and cold sides. Plate-fin heat exchangers were selected due to their high efficiency, compact size and lightweight construction. The condenser testing is focused on evaluating heat transfer performance and pressure drop characteristics, to derive empirical correlations for the Colburn factor (j) and Fanning friction factor (f) for various fin configurations and operating conditions, for a mixture of air and steam, during condensation in the presence of non-condensable gases, for which there are no dedicated correlations available in literature.

2. Experimental Setup

The experimental setup consists of equipment for the flow stream generation and instrumentation for the flow characteristics measurements, as presented below.

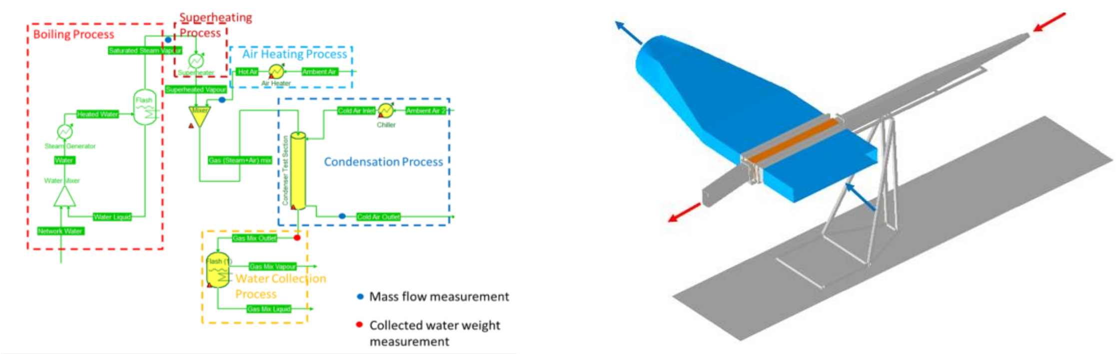
2.1 Equipment

The testing of the condensers needs to be conducted in proper conditions that resemble the real conditions of the bypass flow and the exhaust gases. The hot-gas stream consists of combustion hot gases having a fraction of water in vapor state. Thus, in experimental conditions, the equivalent hot-gas stream will consist of water vapor mixed with hot dry air. The crossflow stream consists of ambient cold air, modelling the atmospheric bypass flow. The experimental conditions were selected to achieve representative Reynolds number and water mass fraction ratios to the estimated cruise and take off conditions for the WET engine, taking also into consideration the experimental equipment limitations. In the experimental test-rig, the condenser specimens will be tested for varying operating conditions, inlet temperatures, mass flow rates and steam-air mixture mass fractions. All experiments will be performed for a close to ambient static pressure value. Furthermore, additional isothermal measurements, using only dry air, will be performed to derive the pressure drop characteristics of the specimen for isothermal conditions. The targeted range of experimental conditions is presented in Table 1.

Table 1: Targeted experimental conditions range.

	Cold air	Hot gas (steam and air mixture)	Dry air	Units
Pressure level	atmospheric	atmospheric	atmospheric	-
Inlet temperature	283-293	320-350	300-340	K
Reynolds number	1,000-1,400	1,000-6,000	1,000-6,000	-
Water mass fraction	-	0.1-0.3	-	
Mass flow rate	0.8-1	0.02-0.12	-	kg/s

For the estimation of the necessary inlet conditions of the experimental equipment, an equivalent thermodynamic model was developed in COCO free software platform, as presented in Figure 1.



(a) (b)

Figure 1: (a) Thermodynamic model of the test rig experimental equipment. (b) Wind tunnel and condenser experimental setup. The red arrow shows the hot stream flow, and the blue one shows the cold stream flow.

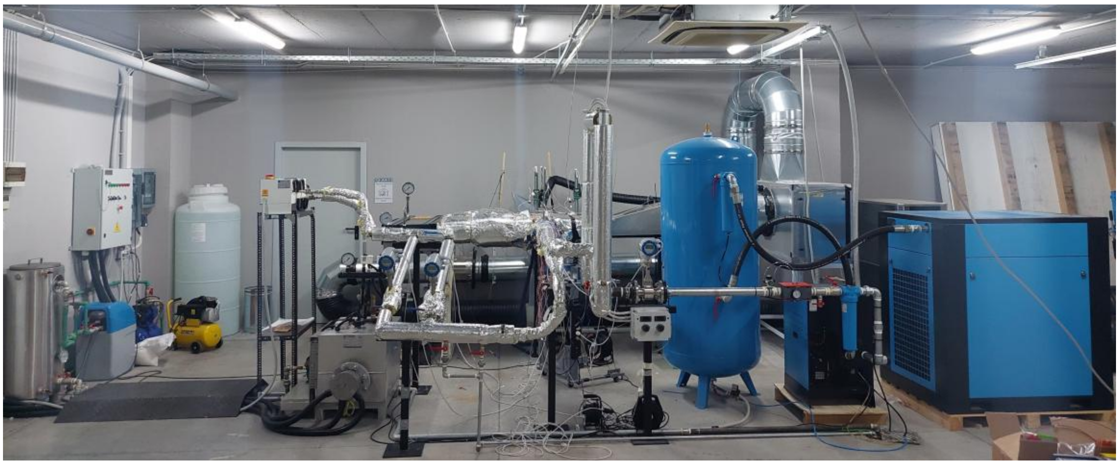


Figure 2: Experimental test-rig.

As shown in Figure 1, there are 5 processes to be implemented for the experiment. Firstly, the boiling process, in which the water goes into the boiler and turns into saturated steam. The boiler has a pressure regulator to control the outlet pressure of the saturated steam. Then, the superheating process where the saturated steam goes into the superheater, if the condenser has a superheated section at the front. For the cases that superheated steam is not needed, the superheater can be bypassed, and the saturated steam can be used directly from the boiler. For the air heating process, an air compressor and an air dryer are used, to generate the dry air flow at the desired mass flow rate. After the air compressor, a pressure regulator controls the outlet pressure of the air. Lastly, extra length is added to the pipes, to create the necessary length for the heating of the air with the use of thermal resistances installed inside the flow. These thermal resistances are controlled by a panel to achieve the required temperature. These two flows, the saturated steam and the hot dry air are mixed inside a Y-junction acting as a mixing chamber right before entering the condenser. The test specimen is mounted on a custom-design wind tunnel which has the capability to change the inclination angle of the condenser in order to have the capability to experimentally investigate the effect of the inclination on its performance. On the other stream, the cold air flows, generated by a centrifugal fan that sucks air, and it goes through an air chiller right before it enters the condenser cold side. After the condenser, a water separator is installed that collects the liquid water to measure it real-time on a weight scale.

2.2 Measuring Instrumentation

Regarding the measuring instrumentation of the test-rig, the latter consists of various air and steam flowmeters, thermocouples, Pitot-static tubes, digital manometers, barometers, humidity sensors and auxiliaries. More specifically, a DN50 flowmeter is placed at the hot dry air inlet before the air is mixed with the steam, to measure the mass flow rate, pressure and temperature of the hot dry air stream. Two DN40 and DN15 vortex flowmeters are placed at the steam inlet before it is mixed with the hot air and they measure the volumetric flow rate of the superheated steam for high and low flow rates respectively. A DN300 vortex flowmeter is placed at the outlet of the cold air and measures its mass flow rate. Pressure and temperature measurements at the condenser inlet and outlet can be measured within the hot gas and cold air streams with Pitot static tubes and T-type thermocouples. Additionally, temperature measurements of the test-rig and condenser specimen surfaces can be carried out with PRT sensors at the inlet and outlet. These measurements can be recorded by two temperature loggers and two digital manometers and saved for later analysis. Also, for the measurement of the condensate, there is an electronic scale that measures the weight of the water in real time and records it directly on a computer. More details about the experimental setup can be found in Dimitriadou et al. (2024).
The uncertainty of the measurements, based on selected equipment, is shown below:

Table 2: Uncertainty of measurements based on selected equipment.

Measurement	Uncertainty	Measurement	Uncertainty
Temperature cold side	≤ 0.1 % (± 1.0 K)	Differential pressure	≤ ± 0.20 % FSO
Temperature hot side	≤ 0.1 % (± 1.0 K)	Mass flow rate of hot side fluid	< 2 %
Pressures	≤ ± 0.20 %	Mass flow rate of cold side fluid	< 2 %
Ambient pressure	≤ ± 0.20 %	Mass flow rate for condensate flow at hot side exit	< 1 %

3. Performance parameters calculation

The parameters that are used for the evaluation of a condenser's performance are the Colburn factor which represents the heat transfer coefficient and the Fanning friction factor which represents the pressure drop of the condenser.

3.1 Heat transfer – Colburn Factor

The calculation of the heat flux along the condenser is based on measurements on the cold side inlet and outlet with thermocouples installed at the exact opposite positions. Consequently, the condenser is divided into parts, as shown in Figure 3, to calculate the heat flux of each part individually.

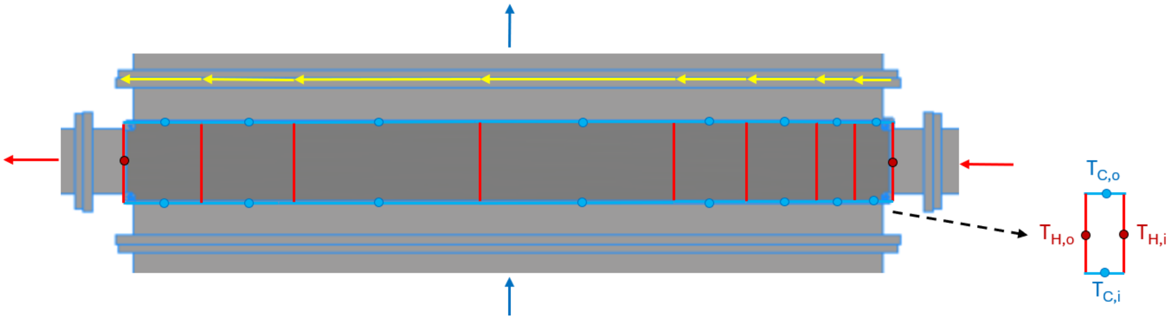


Figure 3: Division of condenser to parts, as shown by vertical red lines, for each measuring station. Red arrows – hot stream, blue arrows – cold stream, yellow arrows – measuring stations length.

The following equations, based on Shah and Sekulic (2003), are used for the calculation of the heat flux of each part and for the whole condenser.
Heat flux of each part of the condenser:

$$q_i = \dot{m}_c c_{p_{C,i}} \left(T_{C,o,i} - T_{C,i,i} \right) \tag{1}$$

Heat flux of the whole condenser:

$$q_{total} = \sum_{1}^{i} q_i \frac{x_i}{L} \tag{2}$$

Overall heat transfer coefficient:

$$UA_i = \frac{q_i}{F_{G,i} \Delta T_{LMTD,i}} \tag{3}$$

Given the logarithmic mean temperature difference as:

$$\Delta T_{LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \text{, with } \Delta T_1 = T_{H,i} - T_{C,o} \text{, and } \Delta T_2 = T_{H,o} - T_{C,i} \tag{4}$$

Then, for the calculation of the heat transfer coefficient for each stream individually, the following equation is used:

$$\frac{1}{UA} = \frac{1}{\eta_{o,h} h_h A_h} + \frac{t_{wall}}{k_{wall} A_{wall}} + \frac{1}{\eta_{o,c} h_c A_c} \tag{5}$$

After the calculation of the heat transfer coefficient of the hot stream, the Colburn factor j can be calculated by the following equation, using the hot side flow characteristics:

$$j = \frac{h Pr^{2/3}}{c_p G} \tag{6}$$

Table 3: Heat transfer equations parameters calculation.

Symbol	Parameter	Calculated by
$\dot{m}_c$	Cold air mass flow rate	Flowmeter
$c_{p_{c,i}}$	Specific heat of cold air	Coolprop properties for Humid Air
$F_{G,i}$	Correction factor	Equals to 1 for condensers
$\eta_{o,h}$	Hot side fin effectiveness	To be calculated from Eq(5)
h_h	Hot side heat transfer coefficient	To be calculated from Eq(5)
A_h, A_c	Hot and cold side total heat transfer areas	Calculated from condenser geometry
$\frac{t_{wall}}{k_{wall}A_{wall}}$	Condenser wall thermal resistance	Calculated from condenser geometry and materials
$\eta_{o,c}$	Cold side fin effectiveness	Calculated from existing correlations
h_c	Cold side heat transfer coefficient	Calculated from existing correlations

3.2 Pressure drop – Fanning Friction Factor

For the calculation of pressure drop in each stream, the total pressures are measured before the inlet and after the outlet of the condenser using the pitot static tubes, which means the contraction and expansion losses are included in the total pressure drop and must be calculated.
Pressure drop for each stream is given below:

$$\Delta p = p_{c,i} - p_{c,o} \tag{7}$$

Then the Fanning friction factor is calculated based on equations developed by Kays and London (1984):

$$f = \frac{r_h}{L}\frac{1}{(1/\rho)_m}\left[\frac{2g_c\Delta p}{G^2} - \frac{1}{\rho_i}(1 - \sigma^2 + K_c) - 2\left(\frac{1}{\rho_o} - \frac{1}{\rho_i}\right) + \frac{1}{\rho_o}(1 - \sigma^2 - K_e)\right] \tag{8}$$

Table 4: Pressure drop equations parameters calculation.

Symbol	Parameter	Calculated by
r_h	Hydraulic radius	Based on geometry
L	Fluid flow length on cold side	Based on geometry
ρ_m	Mean density	Based on COCO calculations
ρ_i, ρ_o	Density at inlet, outlet	Calculated from static pressure at inlet, outlet
G	Mass flux	Flowmeter
σ	Ratio of free flow to frontal area	Based on geometry
K_c, K_e	Contraction and expansion loss coefficients	Based on literature diagrams

4. Results

The following results were measured using the maximum steam, hot air and cold air mass flow rates and they were intended to compare the experimental outlet condensate flow rate with the estimation from the thermodynamic model, in order to assess the accuracy of the experimental measurements on the heat transfer. The condenser specimen tested in this experiment is a crossflow plate fin heat exchanger, with plain fins on hot side and herringbone fins on cold side.

Table 5: Experimental conditions of conducted wet tests.

	Steam	Hot air	Hot mixture (Steam + Hot air)	Cold air
Mass flow (kg/hr)	83	323	406	3,800
Reynolds number	-	-	6,900	1,960
Inlet temperature (K)	376	299	342	289
Water mass fraction	-	-	0.204	-
Total pressure drop (%)	-	-	6.5	0.64
Water recovery (%)	-	-	71	-

The condensate mass flow rate was determined experimentally using an electronic scale, which recorded the real-time mass of the condensed water exiting the system. To validate the experimental results, a thermodynamic model was developed using the COCO simulator. The model was initialized using the measured inlet conditions of the steam-air mixture and the total heat flux, which was estimated based on cold side (air) temperature measurements. The model was then used to predict the outlet conditions and the resulting condensate flow rate.

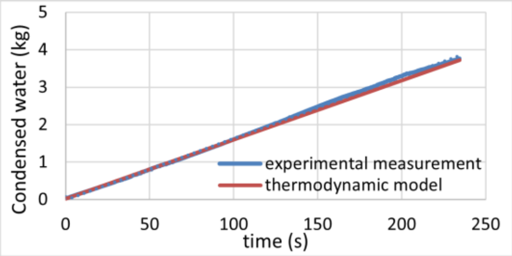


Figure 4: Experimentally measured water mass collection rate compared to thermodynamic model estimation.

As shown in Figure 4, the experimental data closely match the thermodynamic model estimation. More specifically, the mean deviation is only 4 % and the water mass recovery percentage is 71 %, which suggests that the temperature measuring stations on the cold side were sufficient to capture the heat flux accurately. This also implies that heat losses to the environment were minimal, and the experimental setup was well insulated. Also, there was a 6.5 % total pressure drop on the hot side and 0.64 % on the cold side. These values preliminarily estimate the maximum pressure drops expected during condenser operation, as the tests were conducted under conditions of maximum mass flow rates.

5. Conclusions

The designed experimental setup enables controlled testing of both two-phase and single-phase heat exchangers with water-air mixtures across a wide range of flow rates, inlet temperatures, and pressures. The installed instrumentation allows comprehensive data collection along the heat exchanger. Finally, the experimental results closely matched the thermodynamic model and confirmed the setup's accuracy. This strong agreement confirms the setup's reliability for conducting future high accuracy measurements of the condenser's performance parameters. Further work will quantify how specific experimentally tested heat exchanger designs affect WET cycle performance, in terms of heat transfer and pressure drop. After experimental characterization, the results will also be used to develop dedicated correlations for condensation with non-condensable gases.

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