

IMPACT OF NON-CONDENSABLE GASES ON TWO-PHASE HEAT TRANSFER IN AERO ENGINE HEAT EXCHANGERS

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Abstract

Several of the revolutionary flight propulsion concepts studied today comprise heat exchangers as essential components. This could be either as part of unconventional gas turbine cycles as the Water-Enhanced Turbo-fan concept (WET) or as part of the thermal management system of electrified propulsion systems. Two-phase heat exchangers as condensers can contribute to significant improvements. Often the condensing medium is mixed with components that cannot condense in the present temperature range. A common case is the condensation of steam in the presence of air. This presence of non-condensable gases can lead to a significant reduction in the condenser's heat transfer performance. Therefore it is crucial to assess its effect in conceptual design. Two well-established literature methods for modeling the non-condensable-gas-effect are compared with a far simpler alternative approach. The observed deviations in overall heat transfer rate using this latter approach do not exceed 6.6%, whereas modeling complexity and computation time are reduced. This observation enables an efficient assessment of condensation in the early phase of flight propulsion concept evaluation.

Keywords

Heat Exchanger; Thermal Management; Condensation

NOMENCLATURE

Symbols

ΔH	change in heat content of unit volume	J/m ³
D_h	hydraulic diameter	m
$f()$	function	
G	mass flux	kg/(m ² s)
$g()$	heat transfer deterioration factor	
h_{tc}	heat transfer coefficient	W/(m ² K)
j	Colburn factor	
K	molar mass transfer coeff.	mol/(sm ² Pa)
k	thermal conductivity	W/(m*K)
kd	diffusion coefficient	m ² /s
L	flow length in heat exchanger	m
λ	latent heat	W/kg
M	molar mass	kg/mol
μ	dynamic viscosity	Pa s
p	pressure	Pa
p_r	reduced pressure, $p/p_{critical}$	

pp	partial pressure	Pa
Pr	Prandtl number	
Q	heat transfer rate	W
q	heat flux	W/m ²
R	thermal resistance	K/W
Re	Reynolds number	
ρ	mass density	kg/m ³
T	temperature	K
x	mass fraction	

Indices

AIR	air within the hot gas
$bulk$	in midst of the stream
c	cold side
$condH_2O$	pure steam condensation
$condNCG$	steam condensation with NCG-effect
$cool$	coolant (cold side)
eff	effective; defined w.r.t. parting sheet heat transfer area
$film$	condensate film

<i>FUEL</i>	injected fuel leading to hot exhaust gas
<i>g_f</i>	logarithmic mean of NCG in bulk vs. adjacent to condensate film surface
<i>h</i>	hot side
<i>H₂O</i>	water
<i>in</i>	inlet of heat exchanger
<i>L</i>	entity with all-mass-flow-liquid assumption
<i>lat</i>	latent (vs. sensible)
<i>NCG</i>	non-condensable gas
<i>sens</i>	sensible (vs. latent)
<i>surf</i>	at the condensate film surface
<i>total</i>	total (sensible + latent parts)
<i>v</i>	vapor
<i>wall</i>	parting sheet between streams

Abbreviations

HTC	heat transfer coefficient
NCG	non-condensable gas

1. INTRODUCTION

The reduction of detrimental human-caused climate impact is a common goal of society, governments and industries. Ambitious international strategies as the European Union's *European Green Deal* give significant incentive for research and technology development, also in the field of aviation [1].

The propulsion system of an aircraft is one of the important levers to reduce its overall emissions. In the past decades, significant improvements were achieved in fuel consumption, NO_x emissions and noise footprint. Nowadays, the conventional gas turbine aero engine seems to have reached a state at which further evolutionary improvements may not suffice anymore to reach the ambitious goals such as proposed by the *European Green Deal*. Therefore, aero engine manufactures as well as research institutes work on revolutionary aero engine concepts that might enable the next, significant improvement in aviation emission reduction, required by authorities and society.

Several of these revolutionary concepts incorporate heat exchangers as important components in the cycle. MTU Aero Engines' Flying Fuel Cell™ as well as the Water-Enhanced Turbofan (WET) and Reaction Engines' SABRE Engine are examples for the promising impact of heat exchangers in flight propulsion.

Therefore, the aviation industry and academia have an inherent interest to strengthen their capabilities in the assessment of heat exchangers. In cases as the

WET concept, which condenses water from the hot exhaust gas to enable the operation of the wet cycle, two-phase heat exchangers are incorporated. Advanced modeling capabilities are required to assess this change in fluid phase.

One important phenomenon in two-phase heat exchangers can be the non-condensable gas (NCG)-effect during condensation: in cases where the hot side fluid is a mixture of multiple components, of which some do and some do not condense within the temperature range at which the heat exchanger is operated, those NCGs will have a detrimental impact on the heat transfer rate. A common example is a small amount of air in a steam-condenser. During the process of condensation, the concentration of NCG increases adjacent to the evolving condensate film and acts as resistance against further condensation from the bulk to the condensate film. Even small amounts of NCG (in the order of 1%) in a condensing fluid can reduce the achieved heat transfer coefficient (HTC) significantly (in the order of 50%). [2–6]

Therefore, it is substantially important to assess the impact of NCG in condenser-based flight propulsion systems.

The detrimental impact of NCG on heat transfer during condensation is a known phenomenon within the research community. The main focus of investigation is commonly on systems, where only a small percentage of NCG is present in the condensing flow, e.g. due to air leaking into a closed vapor heat exchanger system. Several correlations were developed to model this scenario, e.g. Arai et al. [7], Dehbi & Guentay [8] and Vierow & Schrock [6], to name just a few.

For a condenser such as proposed to be used in the WET concept, the NCG concentration is substantially higher. Here, the phase-changing substance is the smaller part of the flow, as commonly the case for wet air. Several correlations were developed for this scenario also, e.g. Silver [9], Jia et al. [5], Lei et al. [3], Liu et al. [10] and Xu et al. [11], whereas it is still the less treated case [5].

Both of these scenarios have heat transfer rates smaller than those expected without NCG in the flow. As can be seen within the cited literature, correlations for the modeling of NCG-effect on heat transfer tend to be complex and in need of advanced fluid property information, which is not commonly available in every flight propulsion design tool. Therefore, a simpler yet sufficiently accurate approach needed.

In this work, two well-established methods for the modeling of NCG-effect during condensation, the Colburn-Hougen (1934) and the Silver (1947) methods, are compared with the far simpler approach from Arai (2002), that was originally designed for small amounts of NCG but is applied for high-NCG-fractions within this work. The capabilities, advantages and limits of this extended Arai method will be shown.

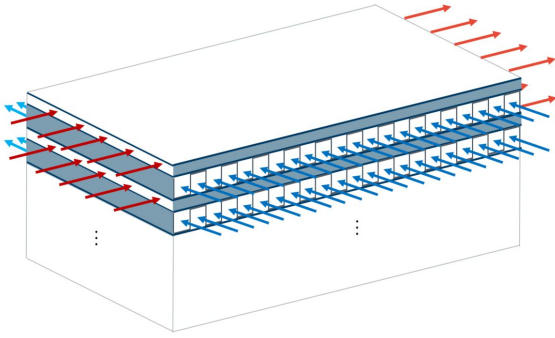


FIG 1. Illustration of the modeled plate-fin heat exchanger with alternating hot and cold passages in cross-flow arrangement.

2. MODEL SETUP

2.1. General Heat Exchanger Model

2.1.1. Condenser Definition

The studies are conducted with a numerical one-dimensional two-phase heat exchanger tool developed by MTU Aero Engines. The condenser is chosen as plate-fin heat exchanger with no fins on the hot side and offset-strip fins on the cold side. Hot and cold sides are stacked in alternating order. Cross-Flow arrangement is chosen. The hot fluid is wet air with a defined water mass fraction x_{H_2O} . The cold fluid is dry air. The latter choice is based on the experience that most often environmental air is the only effective heat sink available in flight propulsion concepts.

The sketch in figure 1 illustrates the modeled plate-fin heat exchanger.

2.1.2. Discretization

Due to the change in temperature, pressure, vapor quality (mass fraction of gaseous water) and fluid properties of the hot flow within the heat exchanger, an equidistant discretization in direction of the hot flow is chosen. The cold flow direction is not discretized to reduce complexity as well as due to the rather small heat transfer per cell in a model with appropriate hot-flow-discretization. For each cell, the respective cell inlet state and properties are assumed to be constant throughout the cell.

2.1.3. Pressure Change

Generally, pressure change within the heat exchanger matrix is calculated as sum of three components: the frictional, accelerational and gravitational pressure changes. As long as the pressure change within the flows is not uncommonly high, its impact on the goal of this work can be assumed to be negligible. Therefore, the pressure change calculation is limited

to the non-reversible frictional pressure change. Accelerational and gravitational pressure change will be neglected.

2.1.4. Heat Transfer

The heat transfer within each calculation cell is determined via the ϵ -NTU method which takes the overall conductance of the heat flow from hot to cold fluid as input. This overall conductance is calculated via a thermal resistance model out of the involved convection, conduction and condensation HTC.

Since parting sheets and fins are assumed to be of thermally well-conducting material of small thickness, fin efficiencies of 100% are assumed and the thermal resistance of the heat conduction through the parting sheets between hot and cold fluid is neglected.

The hot flow heat transfer in the single-phase regime is calculated by the common heat transfer correlations for laminar, transient as well as turbulent flow of VDI Heat Atlas [12] (ch. Ga). The constant heat flux case is assumed and the mean value equations are chosen. The steam-air mixture is modeled as single component fluid with the fluid properties of the mixture.

In the two-phase regime, the hot flow heat transfer is calculated based on NCG-correlations. These are described in the next paragraph.

In order to show the detrimental impact of NCG on the heat transfer, a reference case is defined, which neglects this NCG-effect. For this, the widely accepted correlation of Shah [13] for film condensation of single-component fluids is used. For this, the air-steam mixture is assumed to behave as condensable single-component with the properties of the mixture in the vapor phase and liquid water properties in the liquid phase.

$$(1) \quad htc_L * [(1 - x_v)^{0.8} + \frac{3.8x_v^{0.76}(1 - x_v)^{0.04}}{p_r^{0.38}}] \quad htc_{condH_2O} =$$

with

$$(2) \quad htc_L = 0.023Re_L^{0.8}Pr_L^{0.4}k_L/Dh$$

where htc_{condH_2O} is the pure vapor condensation HTC and htc_L is the theoretical HTC of the liquid-only case (all mass flow assumed to be liquid). x_v is the vapor mass fraction, p_r the reduced pressure, Re_L the liquid-only Reynolds number, Pr_L the Prandtl number of the liquid, k_L the thermal conductivity of the liquid and Dh the hydraulic diameter.

The Shah correlation was originally designed for flow in pipes with circular cross-section. Via the use of the hydraulic diameter Dh within equation 2, its applicability for rectangular channels is assumed.

Since condensation correlations (with or without NCG-effect) commonly do not consider the growth of the thickness (and therefore thermal resistance) of

the liquid condensate film along the hot flow length, this growth is neglected here. Due to the comparably low thermal resistance of a thin liquid water film, the thermal resistance of the condensate film shall be neglected in the course of this paper. The cold flow heat transfer is calculated via the correlation of Manglik & Bergles [14] for offset-strip fins.

2.2. Non-Condensable-Gas-Effect Methods

There are several methods in literature for the calculation of heat transfer during condensation with NCG present. Two of the best established ones are the work of Colburn & Hougen, as precise and comprehensive approach, as well as the publication of Silver, who recommends a simplified approach, with still reasonable accuracy. These two broadly accepted correlations shall be compared with the much simpler approach of Arai et al.

2.2.1. Colburn-Hougen Method

Colburn & Hougen [15] emphasize that the condensation in the presence of NCG is a heat *and* mass transfer problem. The resulting heat transfer depends not only on sensible heat transfer but also on the diffusion of vapor molecules through the bulk gas mixture to the condensate film surface and their latent heat, which is released during phase change. The authors propose a thermal resistance network from the hot side flow (mix of condensable and non-condensable gases) to the cold side flow (e.g. liquid heat sink). The flows are separated by a flat wall (parting sheet) of high thermal conductivity. The condensed vapor is assumed to form a liquid film of small thickness on the hot side of the plate. The condensation process from bulk hot flow to the free surface of the condensate film is modeled by two mechanisms in parallel: the sensible heat transfer from bulk gas to liquid surface, and the mass transfer of vapor from the bulk to the liquid surface including the latent heat transfer to the condensate film. The physical system including the thermal resistance model is depicted in figure 2. Colburn & Hougen also considered an additional potential cold side thermal resistance due to fouling, which is neglected in the figure.

They state, that the heat flux from hot bulk flow to the condensate film surface, $q_{condNCG}$ has to be same as the heat flux through the film, the parting sheet and into the coolant in series, $q_{film,wall,cool}$, which is a reasonable assumption.

$$(3) \quad q_{condNCG} = q_{film,wall,cool}$$

The thermal resistances R_{film} , R_{wall} , R_{cool} can be calculated by common literature correlations for convection or conduction heat transfer. A combined heat transfer $htc_{film,wall,cool}$ can be derived. The sensible

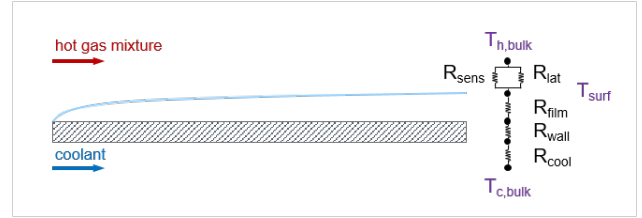


FIG 2. Thermal resistance model of the Colburn-Hougen method for determining the condensation heat transfer in the presence of non-condensable gas

HTC htc_{sens} during condensation in presence of NCG can be calculated by a common single-phase convection correlation as the ones from VDI Heat Atlas. For the calculation of the latent part of the condensation in presence of NCG, an advanced fluid model, providing all required fluid properties as well as partial pressures, as well as equation 5 for the molar mass transfer coefficient K is required. For the Colburn coefficient j the respective same value as in htc_{sens} can be used. The term pp_{gf} is the logarithmic mean of the the NCG's partial pressure in the bulk vs. adjacent to the condensate film. Since the diffusion coefficient kd was not available in the used fluid model, this work follows a calculation within the Colburn & Hougen publication [15], for which $(\mu/(\rho * kd))^{2/3} = 0.71$ is assumed.

$$(4) \quad \begin{aligned} &htc_{sens} * (T_{h,bulk} - T_{surf}) + \\ &+ K * M_v * \lambda * (pp_{v,bulk} - pp_{v,surf}) \\ &= htc_{film,wall,cool} * (T_{surf} - T_{c,bulk}) \end{aligned}$$

with

$$(5) \quad K = \frac{j * G}{M_{bulk} * pp_{gf} * (\mu/(\rho * kd))^{2/3}}$$

where htc_{sens} is the sensible HTC from bulk hot gas to the condensate film and $htc_{film,wall,cool}$ is the combined HTC of the heat flow through condensate film, parting sheet and into the coolant. $T_{h,bulk}$, T_{surf} and $T_{c,bulk}$ are the temperatures of the bulk hot gas, the condensate film surface and the bulk cold gas respectively. K is the molar mass transfer coefficient, M_v is the molar mass of the vapor and M_{bulk} the molar mass of the bulk hot stream. The fluid property λ is the latent heat of vaporization, μ is the dynamic viscosity, ρ the mass density and kd the diffusion coefficient, each for the hot fluid. The partial pressures $pp_{v,bulk}$ and $pp_{v,surf}$ refer to the partial pressures of vapor in the bulk hot flow and at the condensate film surface respectively. The entity pp_{gf} is the logarithmic mean of the partial pressure of NCG in the bulk hot flow and the one at the condensate film surface. G is the mass flux of the hot flow and j the Colburn factor representing htc_{sens} .

Based on these equations, the condensate film surface temperature T_{surf} is the only unknown within equation 4 and can be determined e.g. iteratively.

As stated above, the thermal resistances of film and parting sheet shall be neglected within this work. Therefore, the HTC in presence of NCG, $htc_{condNCG}$, can be written as in equation 6:

$$(6) \quad htc_{condNCG} = htc_{sens} * (T_{h,bulk} - T_{surf}) + K * M_v * \lambda * (pp_{v,bulk} - pp_{v,surf})$$

The Colburn-Hougen method was published in 1934 and is often used and cited in the research field of condensation in presence of NCG. Still, it is quite tedious to implement and requires an advanced fluid model and internal equation solving.

2.2.2. Silver Method

Silver [9] introduces his work by pointing out that the modeling of condensation in presence of NCG is a complex problem and that this becomes also obvious in good but complex modeling approaches as the one from Colburn & Hougen described above. Initiatives to reduce the method's complexity by simplifications often led to inaccurate results. The author presents a simplified method that does not have the same loss in accuracy as others.

The focus lies on crude gas, which is assumed to be saturated or nearly saturated with water vapor at the inlet of the condenser. One can observe that the sensible heat transfer from the saturated gas mix to the heat sink and the latent heat transfer via condensation behave in such a way that the gas mix remains saturated during the condensation process. Therefore, the water vapor heat content and the total gas mix heat content can be calculated if the local temperature is known. The heat content of condensable parts of the crude gas mix, which are not water, can commonly be neglected.

It has been shown that the mass transfer of water vapor to the film surface, where it releases latent heat during phase change, results in a far higher heat transfer than the sensible heat transfer from hot gas mix to condensate film. The sensible HTC remains about constant over the temperature change throughout the condenser, as was demonstrated experimentally. Silver discovered that the ratio of sensible heat capacity to the total gas mix heat capacity at any temperature is the same as the ratio of sensible HTC and the total HTC at that temperature. Therefore, the total HTC can be derived from the sensible HTC at any temperature of the saturated crude gas, if the ratio of heat capacities, and more generally the ratio of the changes in sensible and total volumetric heat contents, ΔH_{sens} and ΔH_{total} , is known. This finding is stated in equation 7.

$$(7) \quad htc_{condNCG} = htc_{sens} * \frac{\Delta H_{total}}{\Delta H_{sens}}$$

The sensible HTC can be determined by common literature correlations. Here, the respective VDI Heat Atlas equations stated above will be used. The frac-

tion of heat contents of unit volume of gas results from a capable gas model.

Silver combines $htc_{condNCG}$ with the HTCs of condensate film, parting sheet and coolant, as described in the Colburn-Hougen method (cp. figure 2). In order to stay consistent with already stated simplifications, the thermal resistance of both condensate film and parting sheet will be neglected in the following.

Silver lays special focus on the correct choice of the mean temperature difference of both sides, which is in the case of saturated condensation not the common logarithmic mean temperature. This approach is not incorporated in the current paper's model since the latter incorporates a sufficient degree of spatial discretization of the heat exchanger. The temperature difference can therefore be well assumed to be the one of the inlet temperatures of each cell.

Whereas Silver came up with a far more compact approach than Colburn & Hougen, his work has the downsides of requiring a fluid model being capable of calculating the fraction of heat contents ΔH at the required temperatures and pressures as well as the reduced accuracy through the use of the observation that the ratio of HTCs equals about the ratio of total vs. sensible heat contents per volume. Additionally, he addresses the deteriorating effect of NCGs on heat transfer only as far as it is incorporated in the fluid model used for the calculation of the heat content ratio.

2.2.3. Arai Method

Compared to these broadly acknowledged approaches, Arai et al. [7] propose a correlation that does not need any advanced gas model information, as e.g. partial pressures or heat of vaporization. The authors investigated the effect of NCG in a horizontal pipe heat exchanger for the use in a passive containment cooling system for nuclear power plants. They state that vertical condensation with NCG present were investigated much more than horizontal condensation with NCG. Therefore, they conducted a test with a single horizontal U-tube with condensation inside the tube. The tested parameter ranges are a mass flux between 20 and 130 kg/(m²s) and an inlet NCG volume fraction of 0. to 0.2. They related the results to empirical correlations modeling pure steam condensation HTCs. Thereby they came up with a heat transfer deterioration factor due to the NCG-effect. The respective correlation is:

$$(8) \quad htc_{condNCG} = htc_{condH_2O} * g(x_{NCG})$$

for $x_{NCG} \leq 0.2$:

$$(9) \quad g(x_{NCG}) = 1 - 1.34 * x_{NCG}^{0.85}$$

for $x_{NCG} > 0.2$:

$$(10) \quad g(x_{NCG}) = 1 - 0.99 * x_{NCG}^{0.66}$$

where the heat transfer deterioration factor $g(x_{NCG})$ is a function of only the NCG mass fraction x_{NCG} .

For the calculation of the pure steam HTC, Arai et al. used the approach of Dobson [16] for the annular flow regime and the approach of Chato [17] for the stratified flow regime. Instead of using two different correlations, in this work the Shah correlation [13] shall be used for the calculation of the HTC without NCG-effect, since it is reasonably accurate and applicable for different flow regimes and orientations [18]. Additionally, Wu, Tokuma and Vierow compared the Arai equation with own experimental data and found that the combination with the Shah correlation gave the best congruence [19].

It should be noted, that in the following the Shah correlation is not applied on the water-fraction only, but it treats the NCG-steam mixture as quasi-single-component in gaseous state.

The resulting $htc_{condNCG}$ can be combined in series with the HTCs of condensate film, parting sheet and coolant as for the other two presented methods. According to above, the thermal resistance of condensate film and parting sheet will be neglected.

With the use of a compact pure steam HTC correlation as Shah, the Arai approach requires significantly fewer input and works with simpler fluid model information than the approaches of Colburn-Hougen and Silver. Additionally, fewer calculation steps are required as with Colburn-Hougen. Its simplicity is a main advantage of the Arai correlation. A downside is the limited experimental coverage of higher NCG volume fractions, which is restricted to volume fractions up to 0.2. This is far from the expected value in some important flight propulsion applications, which operate volume fractions of about 0.8 and similar. The assumption of exclusively horizontal condensation is a limitation as well. The accuracy of this equation with respect to the more detailed Colburn-Hougen approach was not shown in the original work.

3. RESULTS

The described model and NCG-methods were applied in several studies. The most relevant results are summarized here.

As exemplary mission point the following input is chosen: wet air with $T_{h,in} = 428.15K$, $p_{h,in} = 1bar$, $x_{H_2O} = 0.2$, $x_{FUEL} = 0$. on the hot side and dry air with $T_{c,in} = 278.15K$, $p_{c,in} = 0.5bar$ on the cold side. The mass fluxes are $G_h = 10.00kg/(m^2s)$ and $G_c = 20.25kg/(m^2s)$. The hydraulic diameters are $Dh_h = 0.00789m$ and $Dh_h = 0.00293m$ with overall heat exchanger dimensions of 1.4m in hot flow direction and 0.3m in cold flow direction. When some of these values are varied in the following course of studies below, it will be stated.

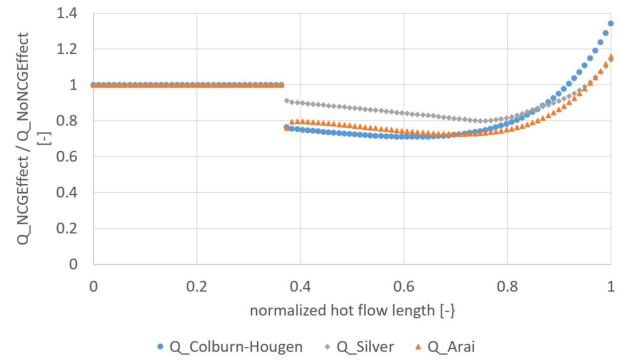


FIG 3. Impact of NCG-effect on local heat transfer rate throughout the condenser, normalized by the theoretical case which neglects the NCG-effect. NCG-effect modeled by three different approaches.

	Colburn-H.	Silver	Arai
overall heat transf.	-19.5%	-11.6%	-18.9%
calculation time	+67%	+27%	+3%

TAB 1. Overall performance of the three NCG-methods, in terms of deviation from the case without any NCG-effect (Shah method)

3.1. Impact of Non-Condensable Gases

In a first step, the general impact of the NCG-effect is shown. As discussed, the NCG-effect is expected to reduce the heat transfer during condensation. More specifically, the HTC of the condensing side is diminished.

The NCG-effect was modeled by the three methods Colburn-Hougen, Silver and Arai. The theoretical reference case without any NCG-effect was modeled by the Shah correlation.

The deterioration impact of NCGs on the overall heat transfer rate is shown in figure 3. From the single-phase inlet of the condenser to the start of condensation, all four cases are identical, since the NCG-effect is only effective in the two-phase region. In the first part of the two-phase region, the detrimental effect on heat transfer can be clearly seen for all three NCG-methods. In the rear part of the two-phase region there is an increase in heat transfer. This latter phenomenon is due to the remaining temperature difference between both sides in this region. The case without NCG-effect transfers much more heat in the first part of the two-phase region and has therefore less temperature difference remaining in the rear part, leading to reduced heat transfer in this region. Overall, the reference case neglecting the NCG-effect transfers significantly more heat than the cases considering NCG-effect, as can be seen in table 1.

It should be noted that all three NCG-methods come up with quite similar heat transfer curves in the given example.

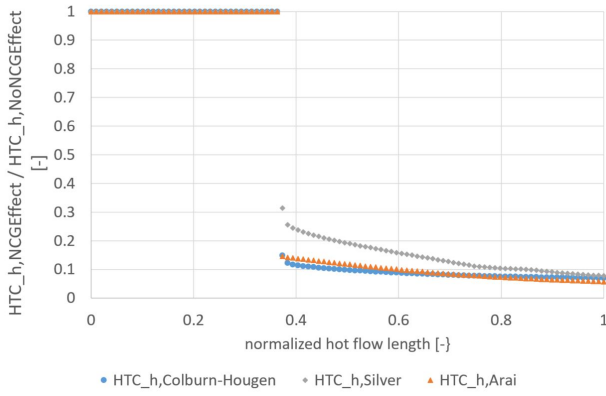


FIG 4. Deteriorating impact of NCG-effect on hot side HTC, the effect calculated with different approaches. Very similar results for Colburn-Hougen and Arai methods.

The heat transfer is influenced by both the hot and the cold side heat transfer capability. For an undisturbed investigation of the NCG-effect, the impact of NCGs on the hot side HTC is shown in figure 4. The drastic decrease in HTC of about 70% and more emphasizes the importance of considering the NCG-effect. The compact Arai-method returns results that are surprisingly close to the two others, especially to the broadly accepted but complex Colburn-Hougen method.

As a third means of comparison, the computation time for all three NCG-methods was compared with the Shah-reference-case. For the given example, the calculation with Colburn-Hougen method took 67% longer than the reference case, the one with Silver method 27% and the one with Arai method 3% (cp. table 1). Although this is not a representative comparison, the Arai method requires by far the lowest computation time of the three NCG-methods, as was expected due to its compact form.

These first results of the qualities of the Arai correlation motivate further studies on the comparability with the broadly acknowledged Colburn-Hougen method. Both the heat transfer rate distribution as well as especially the local HTC results are comparable, with far fewer computation time using the Arai method. The Silver method shall not be further considered in the following investigations.

3.2. Sensitivity Studies on Local HTC

For the given example, the similarity of the Colburn-Hougen and the Arai hot side HTC results is surprisingly strong. And this is the case although the inlet NCG volume fraction is about 0.7 and the hot side mass flux is $10 \text{ kg}/(\text{m}^2 \text{ s})$, values both far outside the original limits of the Arai method, stated above. It shall be shown in the following that this good comparability in output values is not limited to the presented exemplary input data set. The following study shall investigate the congruence of the two method's results further.

The input parameters of the Arai and the Colburn-Hougen method can be described as follows:

$$(11) \quad \begin{aligned} &htc_{condNCG,Arai} = \\ &f(T_h; p_h; x_{h,H_2O}; x_{h,FUEL}; G_h; Dh_h) \end{aligned}$$

$$(12) \quad \begin{aligned} &htc_{condNCG,Colburn-Hougen} = \\ &f(T_h; p_h; x_{h,H_2O}; x_{h,FUEL}; G_h; Dh_h; L_h; \\ &T_c; htc_{cool,eff}) \end{aligned}$$

with the hot and cold side temperatures T_h and T_c , the hot side pressure p_h , the hot side water and fuel mass fractions x_{h,H_2O} and $x_{h,FUEL}$, the hot side mass flux G_h , the hot flow hydraulic diameter Dh_h , the heat exchanger hot flow length L_h and the cold side HTC $htc_{cool,eff}$.

The $x_{h,fuel}$ represents fuel constituents within the flow. Generally, $x_{h,AIR} + x_{h,H_2O} + x_{h,FUEL} = 1$.

The impact of $htc_{cool,eff}$ in the Colburn-Hougen method is due to the condensate film surface temperature T_{surf} impact on $htc_{condNCG,Colburn-Hougen}$. The $htc_{cool,eff}$ entity is the htc_{cool} value already recalculated to refer to the same heat transfer area as the $htc_{condNCG}$.

In the following, the input parameters stated in equations 11 and 12 are varied independently from each other and the resulting deviation in Arai's $htc_{condNCG}$ from the respective $htc_{condNCG}$ value of the Colburn-Hougen method is plotted over the normalized hot flow length.

A fully independent calculation of both methods as in the calculation for figure 4 impedes a direct comparison of local values, since each local result depends on the preceding local results which are its input. These results will differ between both calculations, even if often not significantly, and therefore the input to the following calculation cells will not be completely comparable anymore.

In order to omit this problem, the following sensitivity study is calculated with the Colburn-Hougen method. In each calculation point over the hot flow length, the Arai method is applied also and its resulting HTC is compared with the respective Colburn-Hougen HTC at this point. Therefore, both methods will always use the same boundary conditions. The Arai results will not impact the overall calculation, they are only used for comparison purposes. This enables a clear comparison of both methods over the parameters of interest.

The first parameter of consideration is the hot side inlet temperature $T_{h,in}$. It can be seen in figure 5 that a change in this parameter value has a strong impact on the position of condensation start within the heat exchanger. But the comparability of the HTC is almost unaffected by the change.

In contrast, the hot side inlet pressure variation in figure 6 affects both the position of condensation onset in the expected way - lower pressure results in condensation at lower temperatures, which is further

down the hot flow length - as well as the comparability of the two-phase HTC. The two methods correlate better for lower inlet pressures.

The impact of the water mass fraction x_{h,H_2O} on the onset of condensation is as expected (cp. figure 7): the more water in the wet air, the higher the saturation temperature, which corresponds with an earlier onset of condensation. The parameter has some impact on the level of the typical curve. The amplitude of the curve stays nearly constant, though. A thorough understanding of the impact of x_{h,H_2O} on the HTC comparability requires a more detailed parameter sweep.

The plot of the $x_{h,FUEL}$ variation with the values 0.000, 0.025 and 0.050 is omitted here, since the change in comparability due to changes in this parameter showed to be negligible.

The impact of hot side mass flux G_h variations, shown in figure 8, is one of the most interesting observations. Higher mass fluxes lead to a higher underprediction of the HTC by Arai with respect to the according Colburn-Hougen result. Furthermore, the typical falling trend of the curves over the hot flow length is turned into a quite flat curve for higher mass fluxes. This can be helpful for further studies as described at the end of this section. The major difference in shape between the $G_h = 10 \text{ kg}/(\text{m}^2\text{s})$ curve and the other two is due to different correlations used for them. The former is in the transient heat transfer region, the latter are fully turbulent.

A similar effect can be seen in figure 9 which depicts the impact of hot side inlet Reynolds number on the comparability of both NCG-effect methods. The three Reynolds numbers correspond to mass flux values of $9 \text{ kg}/(\text{m}^2\text{s})$, $18 \text{ kg}/(\text{m}^2\text{s})$ and $27 \text{ kg}/(\text{m}^2\text{s})$.

The main impact of a change in Dh_h is the onset of condensation, which occurs later for higher Dh_h values, due to a reduced heat transfer in the single-phase part. In the two-phase part, there is a slight increase in deviation amplitude with rising Dh_h . The corresponding plot was omitted.

The value of L_h was not varied in this sensitivity study. It would continue the shown plots to the right. This could be interesting for some applications, but is not further discussed here.

Since the cold side contributes to the $htc_{condNCG}$ of the Colburn-Hougen method, also two influence parameters of this second side are considered. The cold side inlet temperature $T_{c,in}$ plot is shown in figure 10. The parameter variation's impact on the position of condensation onset is as expected. A higher overall temperature delta (lower $T_{c,in}$) leads to higher heat transfer and therefore earlier condensation onset. Since the thermal resistance of the hot side is significantly higher than the cold side one, the condensate surface temperature T_{surf} is strongly affected by the change in $T_{c,in}$. This results in a noticeable change in comparability.

The effective cold side HTC $htc_{cool,eff}$ is defined for the same heat transfer area as $htc_{condNCG}$ and

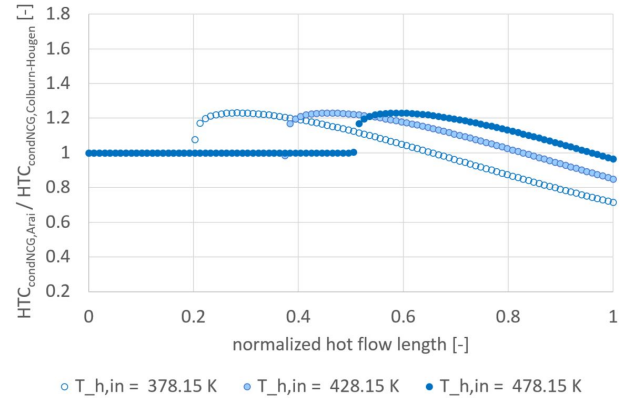


FIG 5. Impact of hot side inlet temperature variation on the local comparability of Colburn-Hougen and Arai method.

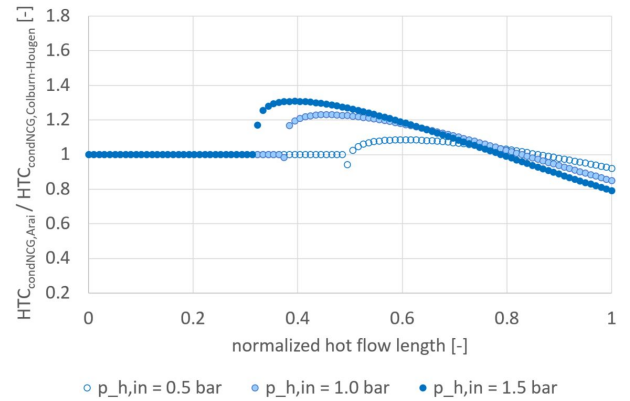


FIG 6. Impact of hot side inlet pressure variation on the local comparability of Colburn-Hougen and Arai method.

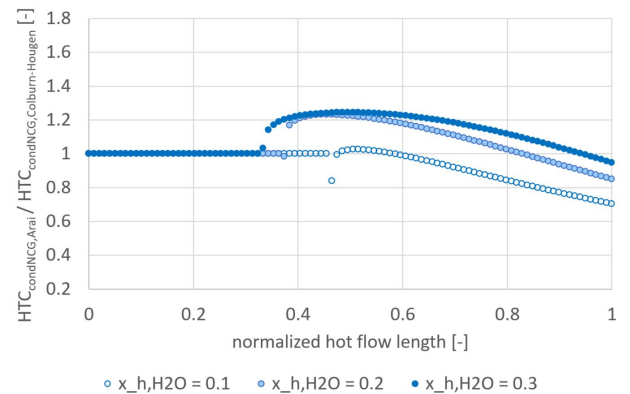


FIG 7. Impact of hot side water mass fraction variation on the local comparability of Colburn-Hougen and Arai method.

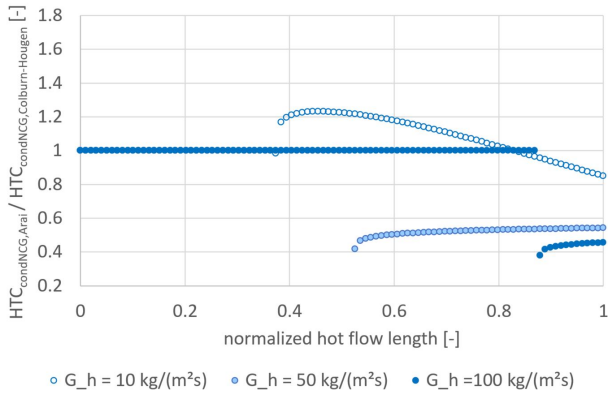


FIG 8. Impact of hot side mass flux variation (induced by mass flow variation) on the local comparability of Colburn-Hougen and Arai method.

is therefore directly comparable with the latter. The variation of $htc_{cool,eff}$ in figure 11 has impact on the level of the curve. The closer the $htc_{cool,eff}$ is to the $htc_{condNCG}$ values (which are about 170 to 770 W/(m²K) in the given case), the lower the level of the curve.

The main outcomes out of these sensitivity studies are as follows: the comparability of the local $htc_{condNCG}$ values of Arai vs. Colburn-Hougen is increased by a low hot side inlet pressure and especially by a hot side mass flow (Re number) value in a certain region of about $G_h = 10 \text{ kg}/(\text{m}^2\text{s})$ or $Re_{h,in} = 3000$ to 6000. The water mass fraction x_{h,H_2O} has an impact that should be investigated further. From the cold side, both the flow's inlet temperature $T_{c,in}$ as well as its effective HTC have significant impact on $htc_{condNCG}$. On the contrary, the impact of $x_{h,FUEL}$ can be neglected. A variation in $T_{h,in}$ and Dh_h has a major impact only on the start of the condensation, far less on the level of local comparability.

The results of the sensitivity studies above show an ambivalent result for the comparability of Arai and Colburn-Hougen method. For most parameter variations around the first exemplary point, the deviations between the two methods stay in the range of about $\pm 30\%$. Therefore, the original comparability is not restricted to a single exemplary point but exists for larger ranges of certain input parameter values. For some parameter variations (as G_h , $Re_{h,in}$ and $T_{c,in}$), the deviations increase to values up to the order of 60%, though.

These results of the sensitivity studies should be used to amend the Arai method in a way that improves the comparability, especially for variations in the parameters with most impact.

3.3. Integral Heat Transfer Rate

Although the comparability of the local HTC values of original Arai and Colburn-Hougen method has

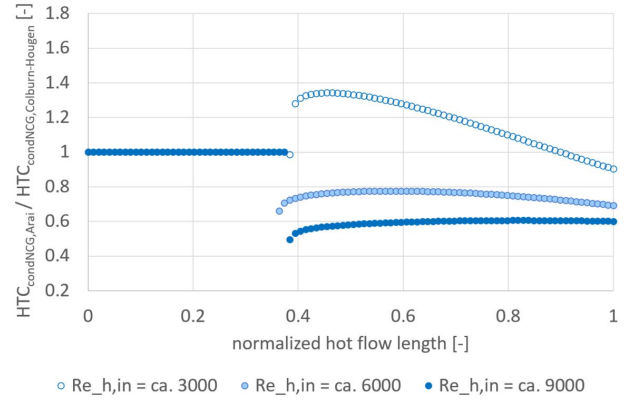


FIG 9. Impact of hot side inlet Reynolds number variation (induced by mass flow variation) on the local comparability of Colburn-Hougen and Arai method.

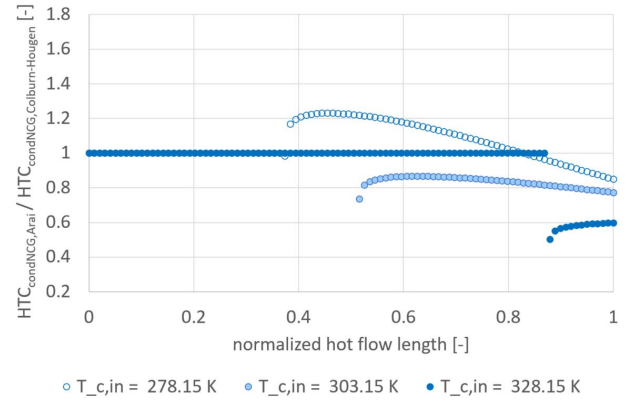


FIG 10. Impact of cold side inlet temperature variation on the local comparability of Colburn-Hougen and Arai method.

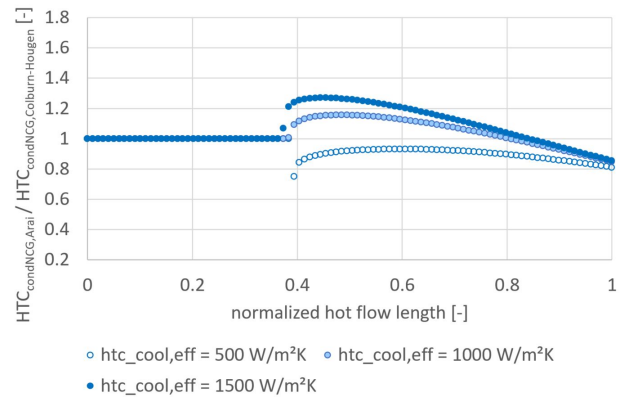


FIG 11. Impact of cold side effective HTC, defined with respect to the same parting sheet transfer area as the HTC of the hot side, on the local comparability of Colburn-Hougen and Arai method.

Parameter	lower	mid	upper
$T_{h,in}$	-0.7%	+0.9%	+2.0%
$p_{h,in}$	+0.3%	+0.9%	+0.8%
x_{h,H_2O}	-2.9%	+0.9%	+1.8%
$x_{h,FUEL}$	+0.9%	+0.9%	+0.8%
G_h	+0.9%	-1.4%	-0.2%
$Re_{h,in}$	+2.3%	-3.7%	-3.6%
Dh_h	+2.0%	+5.9%	+6.6%
$T_{c,in}$	+0.9%	-3.2%	-2.6%
$htc_{cool,eff}$	-2.0%	+0.3%	+1.2%

TAB 2. Overall condenser heat transfer rate calculated by the Arai method compared to the respective results of the Colburn-Hougen method for several input parameters for the lower, mid and upper values specified in the sensitivity study above.

to be improved, the overall (integral) heat transfer rate resulting of both methods is already in good accordance for the tested input parameter variations.

This comparability in overall heat transfer rate is shown in table 2.

The shown deviation level is not restricted to a certain ratio of hot and cold side thermal resistances. Still, the cold side has noticeable impact as can be seen in the values for $T_{c,in}$ and $htc_{cool,eff}$ in table 2.

The high deviation for the Dh_h variation should not be understood as indicator for a high importance of this parameter. Rather, it points out a limitation of the conducted studies: The comparability between Arai and Colburn-Hougen in overall heat transfer rate depends on the section of the respective HTC curve that contributes for the given input parameters. It depends on the position of condensation onset within the condenser as well as the overall length of the condenser. In the specific example of Dh_h , the condensation onset for higher values of Dh_h was so far to the rear that only the positive signed deviations of the HTC-deviations curve contributed, therefore the especially high positive deviation in overall heat transfer value.

This interaction should be investigated further in future work, since it affects the overall heat transfer rate comparability.

The results in table 2 indicate that for the tested input parameter value ranges the overall heat transfer results by Arai and Colburn-Hougen method differ in order of only some percent. With respect to computation time, the method of Arai was quicker than the Colburn-Hougen method by at least a two-digit percent value, sometimes even reaching a 40% reduction and more.

4. SUMMARY & OUTLOOK

4.1. Summary

It was shown that the diminishing effect of NCG on the heat transfer during condensation is well known in research. Several methods exist that address the modeling of this NCG-effect. Two well-established representatives of these, the methods of Colburn & Hougen and of Silver, were implemented for a plate-fin heat exchanger with wet air on the hot side and dry air on the cold side. In addition, the compact empirical correlation of Arai et al. was implemented. It was shown that all three methods came up with similar heat transfer rate and HTC results. Especially the comprehensive Colburn-Hougen and the compact Arai methods returned very similar results. Further comparisons of the HTC and overall heat transfer rate results of these two methods over an enlarged input parameter range investigated the comparability of the results of the two methods. The deviations in local HTC values should be improved by means of correction factors. Even without correction factors, the maximal deviation in overall heat transfer rate within the conducted calculations was +6.6%, most deviations being far smaller in value. The comparability is especially well for a lower hot side pressure level as e.g. in the WET concept. These observations can be very helpful for the design of future aero engine concepts which incorporate condensers, since the Arai method needs far less implementation effort, fluid model capabilities and computation time than the investigated other, more commonly used methods.

4.2. Outlook & Future work

In a next step, the detailed results of the sensitivity study shall be used to modify the Arai method in a way that improves the comparability of its local HTC results with the Colburn-Hougen method. Furthermore, a sensitivity study with combined, multidimensional parameter sweeps could be conducted to assess the parameter space in more detail. Based on this, recommended parameter ranges for the use of the (modified) Arai method can be defined. The impact of the point of condensation onset and the length of the condenser should be considered in this context also. Finally, the modeling results should be validated by experimental data in order to assure their reliability and applicability.

This will enable flight propulsion engineers to assess revolutionary aero engine concepts comprising condensers with NCGs in an efficient way.

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