COMPACT CONDENSING SYSTEM WITH HEAT EXCHANGE SURFACE AND WATER INJECTION FOR SMALL SCALE COMBINED CYCLE MOBILE UNITS

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Abstract **– Theoretical studies developed so far indicate the Small Scale Combined Cycle Units (SSCCU) as an attractive alternative to diesel engines (the actual leader) in the terrestrial propulsion systems field. The biggest problem in this case is to adapt the condensing system to the new conditions, which means small size and low water consumption. Because of these required features, in all the above mentioned studies the Heller condensing system was considered. This solution makes possible the use SSCCU as propulsion system for locomotives and trucks but can't be applied on cars because the radiator of the condensing system is to big to allow the SSCCU framing in the propulsion system allotted space. So, a compaction of the radiator or its elimination should be done in this case.**
The paper presents the result of a study regarding the radiator compaction by water – air heat transfer intensifying. Solution analyzed is the injection of water in the cooling air flow of the radiator.

Keywords: *compact condensing system, mobile unit, heat transfer intensifying, water injection.*

1. INTRODUCTION

It is well known that efficiency increasing is one of the most important objectives of the researchers in the power generation field, where terrestrial propulsion systems represent an important branch.
The studies developed so far proved the possibility to build Small Scale Combined Cycle Units (SSCCU) more efficient and less pollutant than diesel engines – the actual leader in the terrestrial propulsion systems field [1], [2], [3]. Consequently, SSCCU came as an opportune alternative to diesel engines in the terrestrial propulsion systems field.
As long SSCCU operate as terrestrial propulsion systems, their water consumption must be very low. This exclude the possibility to include a conventional condensing system (with cooling tower) in the SSCCU scheme. That is why all SSCCU schemes previously studied are based on a condensing system with heat exchange surface (Heller system). In this case, the water consumption is theoretically zero, so the only major problem regarding the use of a SSCCU as terrestrial propulsion system is to fit it in the terrestrial propulsion system allotted space. This means to reduce the size of SSCCU.
The dimensional analysis of all schemes proved that condensing system is the biggest component of SSCCU. So, the most efficient way to reduce the SSCCU size is to reduce the size of the condensing system, which means to reduce the size of the radiator. The solution adopted in the present study is the increasing of the heat-transfer coefficient by humidification of the cooling air. Obviously, in this case water consumption is not zero anymore. So, beside the heat-transfer coefficient increasing, the water consumption must be estimated too.

2. EXPERIMENTAL INSTALLATION

Schematic of the experimental installation is presented in fig. 1. The components of the installation are: HV – vessel for water heating; P – pump; F – water flow-meter; $TC_1 \ldots TC_4$ – thermocouples; Rad – radiator; UT_1, UT_2 – U-shaped tubes; AT – Anubar tube; C – compressor; WID – water injection device; $V_1 \ldots V_3$ – water vessels; t_1, t_2 – taps; WCMD – water consumption measurement device.
The AT – UT_2 kit allows to estimate the air velocity and, further, to estimate the air flow. For a higher precision, UT_2 leans under the angle $\alpha = 8^\circ$ relative to the horizontal plane.
WCMD consists of vessels $V_1 \ldots V_3$. Every measurement starts noting the water level in V_1 (h_i). The water flow drained into V_2 is adjusted (acting the tape t_1) so as to compensate the water flow injected by WSD. The aim of this adjustment is to keep the maximum level of water in V_2 (in order to keep constant the suction head) but minimizing the water flow drained from V_2 in V_3; this way the time required for V_2 filling is maximized, so the injected water flow is determined more preciously. The water drained from V_2 in V_3 (by opening the tape t_2) is moved in V_1. Thus, results the final level of water (h_f). The injected water flow is expressed as

$$\dot{m}_p = \frac{h_f - h_i}{\tau_p} \quad [kg/s], \qquad (1)$$

where τ_p is the period of time, expressed in seconds.

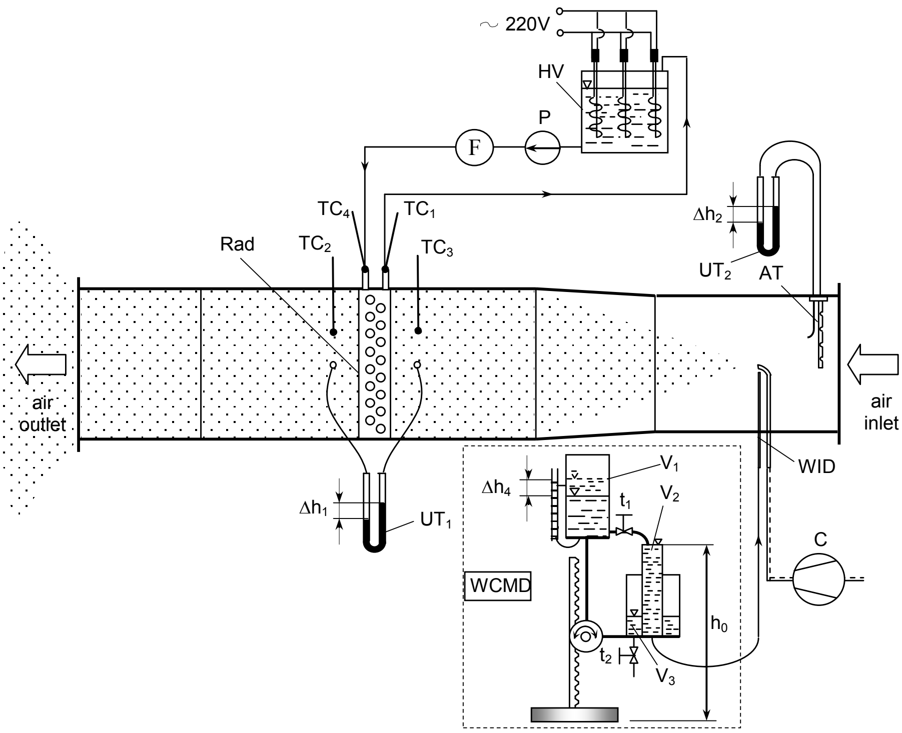

Fig. 1. Schematic of the experimental installation

The experimental installation is fitted into an aerodynamic tunnel provided with a 17 kW fun.

3. EXPERIMENTAL METHOD

The goal of the study was to establish the influences of the air velocity through radiator (w_{rad}) and specific water loss, expressed as

$$\psi = \frac{\dot{m}_p}{\dot{m}_{air\,u}} \quad [\text{kg of water / kg of dry air}], \qquad (2)$$

over the overall heat-transfer coefficient (k_{rad}); $\dot{m}_{air\,u}$ is the dry air flow, expressed in kg/s. In order to establish these influences, the most relevant curves are $k_{rad} = f(\psi)$ − drawn on several values of w_{rad} and $k_{rad} = f(w_{rad})$ − drawn on several values of ψ.

Variation of ψ was made by changing the value of h_0 (see fig. 1) while $\dot{m}_{air\,u}$ was kept constant.

The dry air flow is calculated with formula

$$\dot{m}_{air\,u} = S_{AT} \cdot w_{AT} \cdot \rho'_{AT} \quad [\text{kg/s}], \qquad (3)$$

with:

- S_{AT} = $4{,}665 \cdot 10^{-2}$ m^2 − cross section of the experimental installation in the point where Anubar tube is placed;
- w_{AT} − air velocity in AT cross section;
- ρ'_{AT} − air density in AT cross section.

In order to express ρ'_{AT} it was necessary to determine the air pressure in sections 2 and 3.

Admitting that maximum pressure drop on radiator is the maximum pressure achieved by the fan, namely 300 mm H_2O (0,0294 bar), it means that air density varies 3% at most. In this case we can consider

$$p_2 = 1{,}013 \quad [\text{bar}] \qquad (4)$$

and

$$p_3 = p_2 + \Delta p_1 = p_2 + 9{,}81 \cdot 10^{-5} \cdot \Delta h_1 \quad [\text{bar}]. \qquad (5)$$

Knowing p_3, the air density ρ'_{AT} can be expressed as

$$\rho'_{AT} = \frac{10^5 \cdot p_3}{R_{air} \cdot T_{amb}} \quad [\text{kg/m}^3], \qquad (6)$$

where R_{air} = 287,04 J/kg·deg is the air constant while T_{amb} [K] is the ambient temperature.

It was admitted formula (6) because T_{AT} - T_{amb} < 1 deg, which means an air density variation less than 0,3%.

The air velocity in section AT is given by the calibrating equation of the Anubar tube [4]

$$w_{AT} = 5{,}058263 + 0{,}217592 \cdot \Delta h_2 - \frac{6{,}519252}{\Delta h_2} \quad [\text{m/s}]. \qquad (7)$$

The air velocity through radiator is given by the similar formula

$$w_{rad} = 18{,}410727 + 0{,}7999344 \cdot \Delta h_2 - \frac{23{,}686573}{\Delta h_2} \quad [m/s] \quad . \quad (8)$$

In order to calculate k_{rad} it was necessary to determine the following parameters:

- Water flow, expressed as

$$\dot{V}_w = \frac{V_f - V_i}{\tau} \quad [m^3/s]\,, \quad (9)$$

  where V_f / V_i are the initial / final indication of F while τ is the period of time, expressed in seconds;
- Temperatures of water on radiator inlet / outlet – t_4 / t_1 [°C], measured by TC_4 / TC_1;
- Enthalpies of water on radiator inlet / outlet (i_4 / i_1); they are calculated using formula [5]

$$i = A_1 + A_2 \cdot y + A_3 \cdot y^2 + A_4 \cdot y^6 + z\left[B_1 + B_3 \cdot y^6 + \frac{B_4}{(y+0{,}5)^5}\right] + z^2\left(C_1 + C_2 \cdot y + C_3 \cdot y^6\right) + D_1 \cdot z^4 \cdot y^{12} \quad [kJ/kg]\,, \quad (10)$$

  where $y = t/100$ while $z = (500-p)/100$; t and p are the water temperature [°C] and pressure [bar]; coefficients A_n, B_n, C_n, D_n are given in table 1.

n	A_n	B_n	C_n	D_n
1	49,4	-9,25	$-7{,}3\cdot10^{-2}$	$3{,}39\cdot10^{-8}$
2	$4{,}025\cdot10^{2}$	1,67	$7{,}9\cdot10^{-2}$	–
3	4,764	$7{,}36\cdot10^{-3}$	$6{,}8\cdot10^{-4}$	–
4	$3{,}333\cdot10^{-2}$	$-8\cdot10^{-3}$	–	–

Table 1: Coefficients used for water enthalpy calculation.

- Average density of water in radiator, given by [5]

$$\frac{1}{\rho_{mw}} = E_1 + E_2 \cdot y + E_3 \cdot y^2 + E_4 \cdot y \cdot (y - 1{,}5)^3 + F_1 \cdot X^4 \cdot Y^{12} + z\left[G_1 + G_2 \cdot y + G_3 \cdot y^6 + \frac{G_4}{(y+0{,}5)^3}\right] + z^2\left(H_1 + H_2 \cdot y^3 + H_3 \cdot y^9\right) \quad (11)$$

  The values of coefficients E_n, F_n, G_n, H_n are given in table 2.

n	E_n	F_n	G_n	H_n
1	$9{,}771\cdot10^{-4}$	$1{,}1766\cdot10^{-13}$	$3{,}225\cdot10^{-6}$	$3{,}7\cdot10^{-8}$
2	$1{,}774\cdot10^{-5}$	–	$1{,}3436\cdot10^{-6}$	$3{,}588\cdot10^{-8}$
3	$2{,}52\cdot10^{-5}$	–	$1{,}684\cdot10^{-8}$	$-4{,}05\cdot10^{-13}$
4	$2{,}96\cdot10^{-6}$	–	$1{,}432\cdot10^{-7}$	–

Table 2: Coefficients used in water density calculus.

As long P is a circulating pump (variations of water density and enthalpy are insignificant − lower than 0,05% when pressure varies from 1 bar to 2 bar), in all calculations we considered $p_1 = p_4 = 1{,}013$ bar;

- Heat transferred from water to air, calculated as

$$Q = \dot{m}_w \cdot (i_4 - i_1) = \dot{V}_w \cdot \rho_{mw} \cdot (i_4 - i_1) \quad [kW]\,; (12)$$

- Temperatures of air before / after radiator t_3 / t_2 [°C], measured by TC_3 / TC_2;
- Logarithmic temperature difference, given by

$$\Delta t_{lg} = \frac{|\Delta t_1 - \Delta t_2|}{\left|\ln \frac{\Delta t_1}{\Delta t_2}\right|} \quad [deg]\,, \quad (13)$$

  where Δt_1 and Δt_2 are the temperature differences between fluids changing heat:

$$\Delta t_1 = t_1 - t_3\,;$$

$$\Delta t_2 = t_4 - t_2\,.$$

The overall heat transfer coefficient can now be calculated with formula

$$k_{rad} = \frac{10^3 \cdot Q_c}{S_{rad} \cdot \Delta t_{lg}} \quad [W/m^2 \cdot deg]\,, \quad (14)$$

where $S_{rad} = 2{,}496\ m^2$ is the heat transfer surface of the radiator.

4. RESULTS

In order to appreciate the advantages and drawbacks of the experimented system, the similar system without water-injection (with dry cooling air) was firstly tested. In this simplified case, the characteristic curve $k_{rad} = f(w_{rad})$ is the one presented in fig. 2. As can be seen in this figure, when w_{rad} increases from 12,9 m/s to 42,5 m/s, then k_{rad} increases from 91,07 W/m²·deg to 187,39 W/m²·deg.

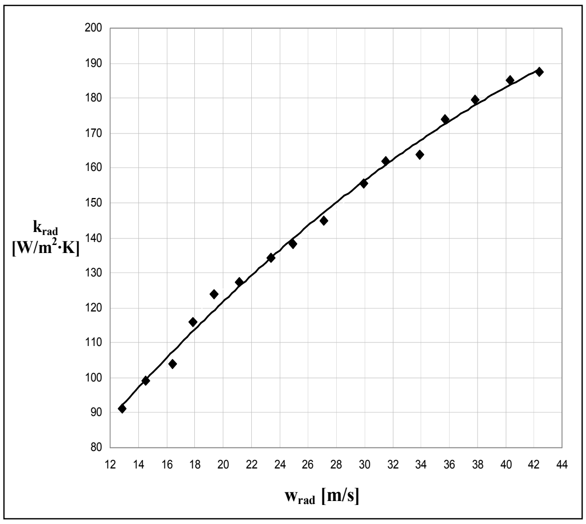

Figure 2: Variation of k_{rad} with w_{rad} in the case of the cooling system with dry air.

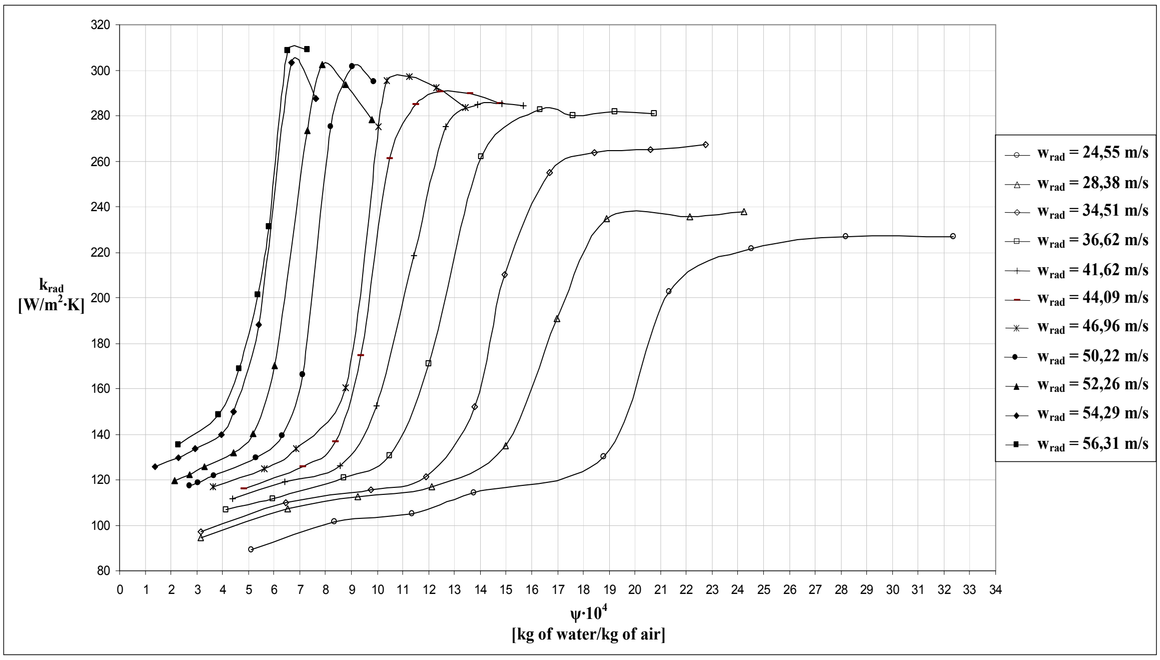

Figure 3: Variation of k_{rad} with ψ for different values of w_{rad}.

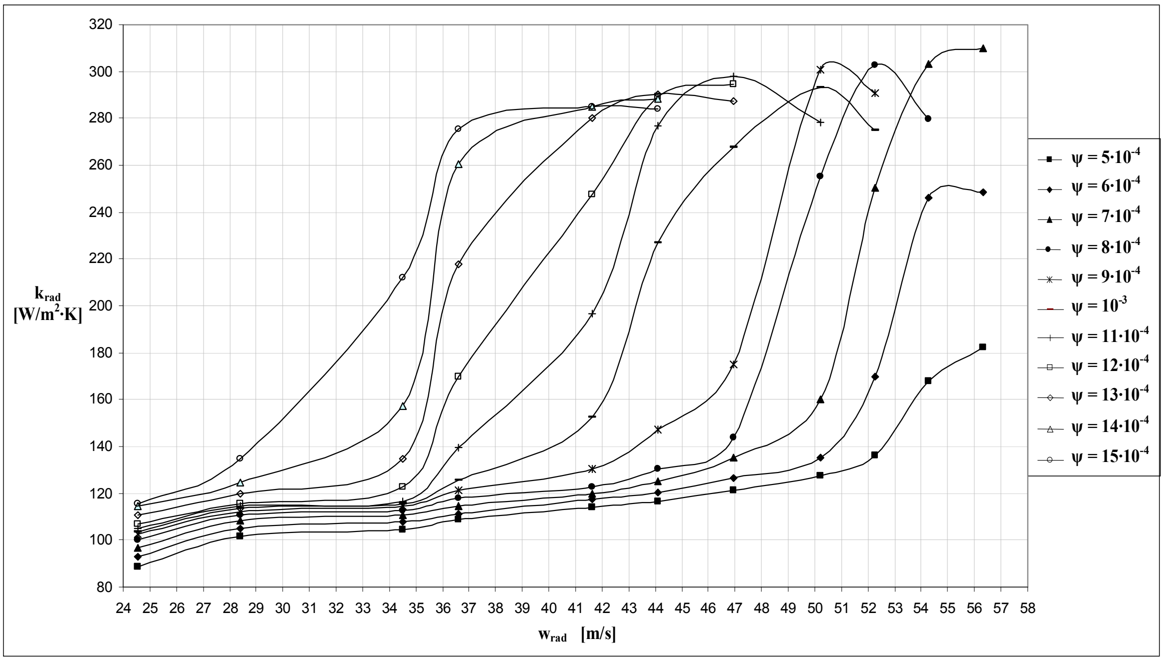

Figure 4: Variation of k_{rad} with w_{rad} for different values of ψ.

As was mentioned in chapter 3, the most relevant curves for the study of the condensing system with humidification of the cooling air are $k_{rad} = f(\psi)$ − drawn for several values of w_{rad} and $k_{rad} = f(w_{rad})$ − drawn for several values of ψ.

These curves are presented in fig 3 and 4. Can be observed that ψ has an optimum value (ψ_{opt}) − described by an optimum value of k_{rad} ($k_{rad\ opt}$) − for any value of w_{rad}. When w_{rad} is high, ψ_{opt} correspondes to the maximum value of k_{rad}. The curves $k_{rad} = f(w_{rad})$, drawn for different values of ψ (see fig. 3) indicate that w_{rad} has also an optimum value when ψ is constant.

Fig. 2 also shows that $k_{rad\ opt}$ increases while ψ_{opt} decreases when w_{rad} increases. This is benefic because it means heat-transfer intensifying by water consumption reduction. Thus, maximum value

$k_{rad\ M}$ = 308,9 W/m^2·deg, reached experimentaly, is describeded by the maximum velocity $w_{rad\ M}$ = 56,3 m/s and $\psi_{opt} = 7{,}29 \cdot 10^{-4}$ kg of water / kg of dry air. Taking into account the specific air flow of $14{,}26 \cdot 10^{-3}$ kg of dry air / kJ of generated energy − required to cool the water passing the radiator in the case of the optimized SSCCU configuration [4] − it means that specific water consumption is $1{,}04 \cdot 10^{-5}$ kg of water / kJ of generated energy. So, the water consumption (injected water flow) of a SSCCU generating 150 kW is $1{,}56 \cdot 10^{-3}$ kg/s = 5,62 kg/h.

Obviously, maximum w_{rad} means maximum energy consumed by the fan for air circulation. So, the overall energetic balance may indicate another (lower) value of w_{rad} as the optimum one. We mention that this approach is not the subject of the present paper and will be studied in a further step of the research.

In order to mark the limits for k_{rad} and ψ_{opt}, it is necessary to analyze the other extreme case, described by the minimum value of w_{rad}, namely $w_{rad\ m}$ = 24,55 m/s. In this case, $k_{rad\ opt}$ = 221,21 W/m^2·deg while $\psi_{opt} = 24{,}52 \cdot 10^{-4}$ kg of water / kg of dry air. So, the water consumption (injected water flow) of the optimized SSCCU and generating 150 kW becomes $5{,}25 \cdot 10^{-3}$ kg/s = 18,9 kg/h.

5. CONCLUSIONS

The results of the study prove the significant increasing of the overall heat-transfer coefficient when water injection is used for cooling air humidification. Thus, k_{rad} can increase 1,5…1,65 times (on the same value of w_{aer}) if inject water in the cooling air stream. Unfortunately, the analyzed solution implies an water consumption of 5,62 kg/h (for $w_{rad\ M}$ = 56,3 m/s) up to 18,9 kg/h (for $w_{rad\ m}$ = 24,55 m/s) in the case of an optimized SSCCU for terrestrial propulsion, generating 150 kW. Taking into account the great benefits of this solution (important reduction of the radiator size and, consequently, of SSCCU size; lower investment costs due to lower materials costs), the water consumption (even on minimum w_{rad}) can be agreed for a terrestrial propulsion system of 150 kW.

Can be concluded that analyzed cooling system allows the use of SSCCU as propulsion system even for cars.

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