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Sensitivity Analysis on the Performance of a Natural Draft Direct Dry Cooling System for a 50 MWe CSP Application

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***Abstract* -** Natural draft direct dry cooling systems (NDDDCSs) provide a new method of directly condensing the working fluid in power cycles compared to traditional forced-draft air-cooled condensers or indirect dry cooling towers. Minimal research has been conducted on the natural draft direct dry cooling system with regards to concentrated solar power (CSP)-based applications. With no need for additional fan drives, the operational costs of these systems are kept to a minimum, while simultaneously increasing the net power output of power cycles due to reduced parasitic loads. Previous work developed and validated a one-dimensional model to simulate the performance of a natural draft air-cooled condenser under no wind conditions, by simultaneous solution of the relevant energy- and draft equations. This study uses the model to conduct a sensitivity analysis on the performance of a NDDDCS for a 50 MWe CSP application, while varying major geometric ratios of the cooling tower. The main drivers of the presented sensitivities are identified and the effects on the various loss factors are highlighted. Results show increased performance for higher total height to inlet diameter- (H_5/d_3) and outlet diameter to inlet diameter (d_5/d_3) ratios and that air mass flow rates dominate performance benefits, with reduced steam velocities also playing a secondary role. The velocity distribution at the tower outlet is optimized at a (d_5/d_3) ratio of 0.61. Decreasing the ratio of inlet diameter to inlet height (d_3/H_4) makes the tower significantly less susceptible to cold inflow due to higher exit air velocities. Reference tower dimensions demonstrate the relative size of NDDDCSs capable of achieving the required performance (under no wind conditions) as a stand-alone system.

***Keywords*:** Air-cooled condenser, heat transfer, natural draft, numerical simulation.

1. Introduction

The focal point of global power generation has shifted towards the implementation of sustainable and clean energy [1]. In the current political and economic climate of the world, it has also become important that countries become more self-reliant when it comes to power generation and the resources required to produce electricity. The future of the human race is closely linked to the amount of energy that can usefully be converted to do work. With the global population ever increasing and critical resources like water becoming scarcer, the way that electricity is produced is becoming more important [2]. Modern electricity generation cycles have much higher requirements to be feasible than in the past. These systems must be of a renewable nature, and resource wastage must be kept to an absolute minimum. A large part of plant performance is determined by the cooling system, where adverse wind effects and high ambient temperatures affect normal fan-driven air-cooled condensers, causing vacuum related load losses and trips in extreme cases [3,4]. On the other hand, indirect natural draft cooling systems have reduced thermal efficiency and require shell-and-tube condensers, which increase system cost and complexity.

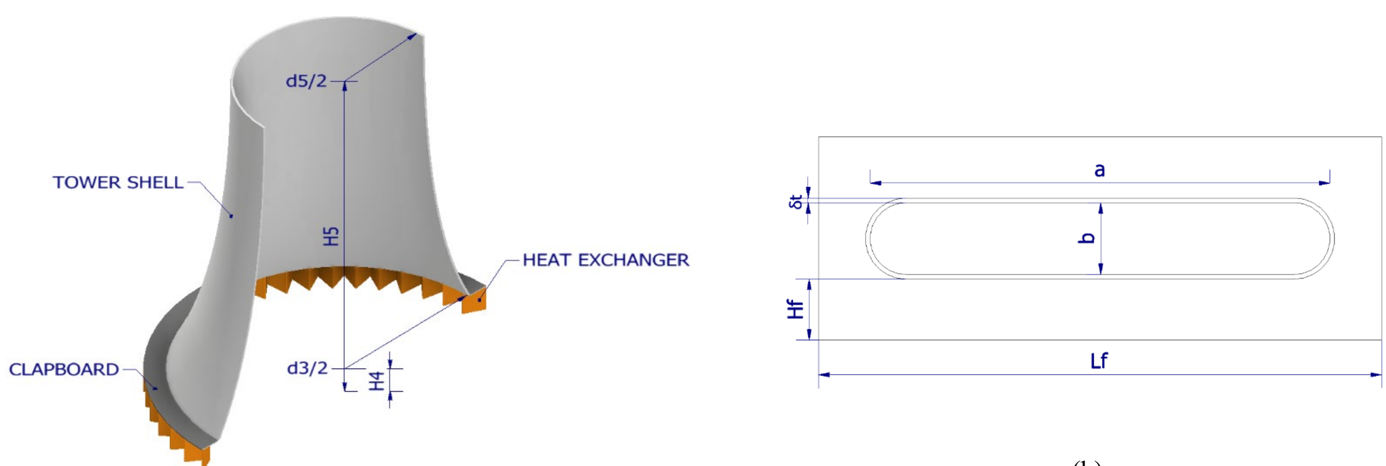
The implementation of a NDDDCS with vertically arranged heat exchangers is proposed that negates the aforementioned disadvantages of traditional air-cooled condensers and retains the advantageous characteristics present in indirect dry cooling systems.

It is well established that Kröger [5] laid the foundation for the performance of forced draft ACCs and indirect natural draft cooling systems with his one-dimensional approach. The NDDDCS has been attracting academic interest due to its potential advantages. Kong et al. [6,7] established that increasing the apex angle of a NDDDCS with vertically arranged heat exchangers improves the system performance under wind and no-wind conditions. This is mainly due to the incoming air having a larger radial velocity component for larger apex angles relative to smaller apex angles. The presence of high tangential velocity components causes the formation of vortices within the tower shell. Similarly, installing external windbreakers for a large scale NDDDCS causes the redirection of air in the radial direction, improving performance [8]. The installation of internal windbreakers for medium- to small scale NDDDCSs has been investigated as well [9]. These windbreakers might be essential for smaller scale NDDDCSs due to the lower driving force present, resulting in drastic system performance degradation under high wind conditions. These medium- to small scale systems suffer from performance degradation due to cold inflow as well, which is a phenomenon that has not been studied in detail for large scale systems. These large scale systems are less susceptible to this effect because the exit velocity of air is sufficiently high in these systems [10]. According to Dai et al. [11], these smaller scale towers benefit from installing platforms that induce a swirling effect on the air.

This paper employs a one-dimensional thermo-fluid model, developed and validated in previous work [12] from the foundations of Kröger [5], to conduct a sensitivity analysis on the performance of a NDDDCS under no wind conditions. Key geometric ratios are varied and the effects on the system performance evaluated. Due to the advantages of the NDDDCS, it can potentially increase the viability of CSP plants. The study therefore performs the analysis on a 50 MWe CSP application. A 40% steam cycle efficiency is assumed for the nominal output point, which requires the system to reject 75 MW of heat to the atmosphere. Incidentally, the 50 MWe Khi Solar One CSP plant, operating in South Africa, employs one of the first commercial NDDDCS systems [13] as part of its integrated 205 m high, 37 m diameter central receiving and cooling tower.

2. Geometry

A schematic of the NDDDCS system and a section of the finned tube used in this study is shown below in Figure 1, where H_5 is the total tower height, H_4 is the inlet height, d_5 is the outlet diameter and d_3 is the inlet diameter.



(a) (b)

Fig. 1: a) NDDDCS geometry b) finned tube

Exhaust steam enters the large diameter steam ducting headers, where it then flows towards the finned tube bundles. The steam condensates against the inner wall of the finned tube, transferring heat energy to the tube wall that is then convectively absorbed by the air flowing over the outside of the finned tube. This causes increased buoyancy of the air within the tower, leading to an updraft and replenishment with fresh colder air at the heat exchanger inlets. The condensate then flows to the bottom of the heat exchanger where it can be pumped for further cyclic use. For this analysis, the finned tube bundles are taken from the work of Kröger [5]. The pressure drop- and heat transfer characteristics of a two-row finned tube bundle, along with the steam-side cross sectional area is combined to form a modern single-row finned tube, similar to the finned tubes employed by Kong et al. [6,7,8]. All results are generated by starting at a reference tower size and subsequently incrementing the relevant dimension to see the effect on the thermo-flow flow performance of the tower. The reference ratios for (H_5/d_3), (d_5/d_3) and (d_3/H_4) are 1, 0.5 and 10 respectively. The delta delta apex angle for all cases is 60°. Tables 1 and 2 show the tube- and tower dimensions respectively.

Table 1: Finned tube dimensions

Description	**Symbol**	**Value**
Tube major axis (m)	a	0.1326
Tube minor axis (m)	b	0.025
Tube thickness (m)	δ_t	0.0015
Tube pitch (m)	P_t	0.067
Fin length (m)	L_f	0.18
Fin height (m)	H_f	0.0195
Fin thickness (m)	δ_f	0.00025
Fin pitch (m)	P_f	0.0023

Table 2: Reference tower dimensions

Description	**Symbol**	**Value**
Tower height (m)	H_5	70
Tower inlet height (m)	H_4	7
Tower inlet diameter (m)	d_3	70
Tower outlet diameter (m)	d_5	35

3. Modelling

3.1. Mathematical model

To solve the thermal performance of a NDDDCS, the energy- and draft equations (Equations 1 and 9) are solved using an iterative numerical model developed in MATLAB. The mathematical model is based on the work of Kröger [5] by combining the mathematics from the air-side of an indirect dry cooling tower with the steam-side of an ACC. Equation 1 governs the heat transfer through the NDDDCS and is represented by:

$$\dot{Q} = \dot{m}_a \, c_{pa} \, (T_{ao} - T_{ai}) = \dot{m}_a \, c_{pa} \, \varepsilon \, (T_{vm} - T_{ai}) \tag{1}$$

where $\dot{m}_a$ is the mass flow rate of air, c_{pa} is the specific heat of air taken at the average air temperature over the heat exchanger, T_{ao} is the outlet air temperature, T_{ai} is the inlet air temperature at half the inlet height, T_{vm} is the mean saturated steam temperature and ε is the heat exchanger effectiveness. The latter is calculated via:

$$\varepsilon = 1 - \exp\left(-\frac{UA}{\dot{m}_a \, c_{pa}}\right) \tag{2}$$

where UA is the overall heat transfer coefficient determined by the following relation:

$$UA = \left(\frac{1}{h_{ea}A_a} + \frac{1}{h_c A_c}\right)^{-1} \tag{3}$$

The term h_{ea} is the effective air-side heat transfer coefficient, A_a is the total air-side surface area, h_c is the steam-side heat transfer coefficient and A_c is the total wall area of the finned tube on the steam side. The effective air-side heat conductance is determined by:

$$h_{ae}A_a = k_a Pr_a A_{fr} Ny \tag{4}$$

where k_a is the thermal conductivity of air, Pr_a is the Prandtl number of air, A_{fr} is the frontal area of the heat and Ny refers to characteristic heat transfer parameter, given by:

$$Ny = 366.007945\, Ry^{0.433256} + 360.588007\, Ry^{0.47037} \tag{5}$$

The nondimensional pressure loss coefficient over the heat exchanger is described by:

$$K_{he} = 4177.08481\, Ry^{-0.4392686} \tag{6}$$

The characteristic flow parameter Ry is evaluated using:

$$Ry = \frac{\dot{m}_a}{\mu_a\, A_{fr}} \tag{7}$$

where μ_a is the dynamic viscosity of air. The steam-side heat transfer coefficient is calculated via:

$$h_c = 0.9245 \left[\frac{k_c^3\, \rho_c^2\, g\, i_{fg}\, H_4}{\mu_c \left(\frac{\dot{m}_a}{2\, n_{tb}\, n_{cc}}\right)\, c_{pa}\, (T_{vm} - T_{ai}) \left[1 - \exp\left\{-\frac{h_{ae}A_a}{\dot{m}_a\, c_{pa}}\right\}\right]} \right]^{0.333} \tag{8}$$

where ρ_c is the density of condensate, g is the gravitational acceleration, i_{fg} is the latent heat of vaporization, n_{tb} is the number of tubes per bundle and n_{cc} is the total number of heat exchangers. The draft equation is given by:

$$p_{a1}\left[\left(1 - 0.00975\frac{H_4}{2\,T_{a1}}\right)^{3.5}\left\{1 - 0.00975\frac{H_5 - \frac{H_4}{2}}{T_{ao}}\right\}^{3.5} - \left(1 - 0.00975\frac{H_5}{T_{a1}}\right)^{3.5}\right] = (K_{il} + K_{he\theta} + K_{ct} + K_{ts})_{he}\left(\frac{\dot{m}_a}{A_{fr}}\right)^2\left(\frac{1}{2\rho_{a23}}\right)\left[1 - 0.00975\frac{\left(H_5 - \frac{H_4}{2}\right)}{T_{ao}}\right]^{3.5} + (K_{to} + a_{e5})\left(\frac{\dot{m}_a}{A_5}\right)^2\left(\frac{1}{2\rho_{a5}}\right) \tag{9}$$

where T_{a1} is the ambient temperature at ground level, p_{a1} is the atmospheric pressure at the ground level, ρ_{a23} is the mean density of air flowing through the heat exchanger, ρ_{a5} is the air density at the tower outlet, K_{il} is the loss due to the inlet louvres and K_{ts} is the loss due to tower supports. K_{ct} is the loss due to the redirection of air at the tower inlet diameter:

$$K_{ct} = 2.21 - 0.42\left(\frac{d_3}{H_4}\right) + 0.091\left(\frac{d_3}{H_4}\right)^2 \tag{10}$$

K_{to} is the outlet loss coefficient of the tower, given by:

$$K_{to} = -0.129\left(Fr_D\frac{d_5}{d_3}\right)^{-1} + 0.0144\left(Fr_D\frac{d_5}{d_3}\right)^{-1.5} \tag{11}$$

The velocity correction factor, a_{e5}, is determined from:

$$a_{e5} = 1.004 + 5.8\left(\frac{d_5}{d_3}\right)^9 + \left[0.007 + 0.043\left(\frac{d_5}{d_3}\right)^{2.5}\right] Fr_D^{-1.5} \tag{12}$$

where Fr_D is the densimetric Froude number, described by:

$$Fr_D = \frac{\left(\frac{\dot{m}_a}{A_5}\right)^2}{[\rho_{a5}\,(\rho_{a6} - \rho_{a5}\,)g d_5]} \tag{13}$$

where ρ_{a6} is the density of ambient air at the elevation of the tower outlet and A_5 is the outlet area.

4. Results and Discussion

Simulation results were generated based on two protocols: 1) incrementing the (H_5/d_3) and (d_5/d_3) ratios while keeping d_3 constant and 2) fixing H_5, then incrementing the (d_3/H_4) ratio while keeping the heat exchanger area constant and incrementing the (d_5/d_3) ratio while keeping d_3 constant. The first part of protocol 2 results in towers with large inlet heights and small inlet diameters (narrow towers) or small inlet heights and large inlet diameters (wide towers). A constant heat load of 75 MW was enforced on the system with an ambient temperature of 313.15 K (a typical worst-case ambient temperature for CSP applications in South Africa), while the steam back pressure is allowed to vary to achieve the required heat rejection rate. Reference tower dimensions are given in Table 3 and all results are nondimensionalised relative to the reference case. Tower sizing was checked to ensure that steam back pressures below 10 kPa(a) (typical lower limit for dry-cooled systems) were not calculated for any of the configurations.

Table 3: Reference tower results

Description	Value
Inlet steam temperature (K)	333.8
Steam back pressure (kPa)	20.5
Outlet air temperature (K)	331.1
Air mass flow rate (kg/s)	4142.5
Inlet steam velocity (m/s)	10.4
Steam condensation rate (kg/s)	31.8
Tower loss (K_{ct} ; nondimensional)	7.11
Pressure loss coefficient ($K_{he\theta}$; nondimensional)	38.6
Outlet loss coefficient + velocity correction factor ($K_{to} + a_{e5}$; nondimensional)	0.77

Results generated according to the first protocol are shown in Figure 2. Trends indicate improved cooling system performance (resulting in reduced steam back pressure) with increased tower height and outlet diameter. A higher tower results in a greater draft, or driving potential available, between the column of hot air in the tower versus a column of cold air in the atmosphere outside. This boosts the air mass flow rate through the system. Larger mass flow rates are also experienced as the outlet diameter is enlarged. A greater outlet diameter causes an enhanced flow area at an equivalent draft, which improves the mass flow rate. The reduction seen in draft is linked to a reduced saturated steam temperature which corresponds to the reducing steam back pressure trend. Despite the reducing draft, the increased mass flow rate dominates to improve tower performance. As shown in Equation 9, K_{to} and a_{e5} appear as a sum in the draft equation, thus their effect is considered together. Values of this sum that approach unity degrade tower performance, while values approaching zero improve tower performance. It is interesting to note that there exists an optimum ratio for (d_5/d_3) of 0.61. However, this effect is very small as it does not significantly affect the steam back pressure trends.

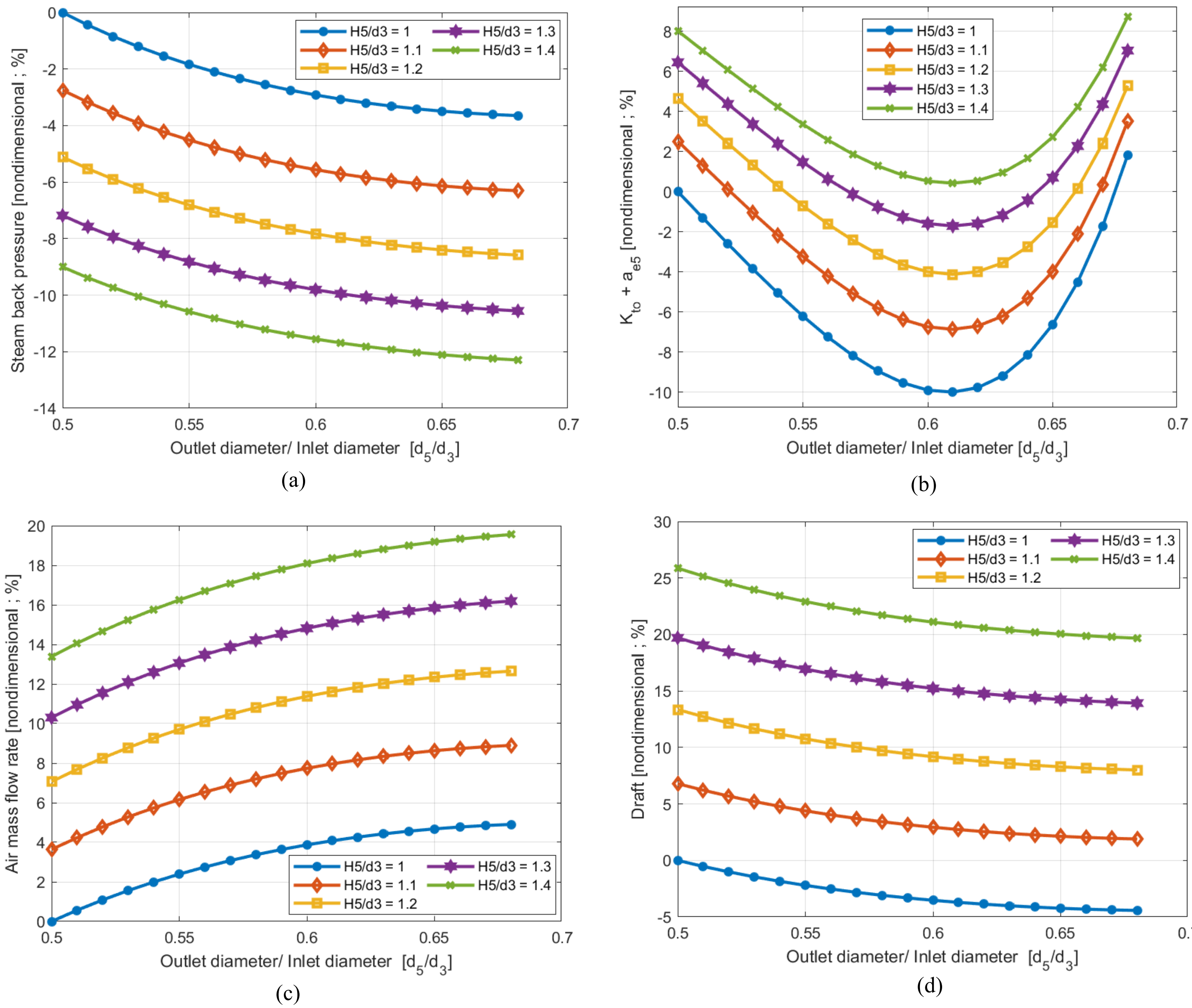


Figure 2: a) Steam back pressure (kPa) b) $K_{to}+a_{e5}$ (unitless) c) air mass flow rate (kg/s) and d) draft (Pa) performance

Results generated according to the second protocol are depicted in Figure 3, which show some very interesting trends. The model predicts best performance (i.e. lowest back pressure) for a tower with (d_3/H_4) ratio of 10, which is largely driven by higher mass flow rates. Towers with the shortest H_4 dimension have the largest effective steam-side area, which produces the lowest steam velocities and pressure drops, resulting in a more efficient cooling system. Despite steam velocities being lower for a tower with (d_3/H_4) ratio of 15, the higher mass flow rates for a tower with (d_3/H_4) of 10 dominate. It is clear that systems with the lowest inlet diameter (d_3) are most affected by increases in outlet diameter

(d_5). As d_5 is enlarged, the larger air-flow area dramatically boosts the achievable air mass flow rate. Increasing ratios of (d_3/H_4) are significantly more susceptible to cold inflow, as indicated by the densimetric Froude number. This is due to the mean outlet air velocity being higher for a (d_3/H_4) ratio of 5 compared to the velocity at higher ratios. A narrower tower experiences significantly lower pressure losses as the air enters the heat exchangers and turn upward into the tower compared to wider towers, which is captured by the tower loss coefficient of Equation 10 (which is only a function of the (d_3/H_4) ratio).

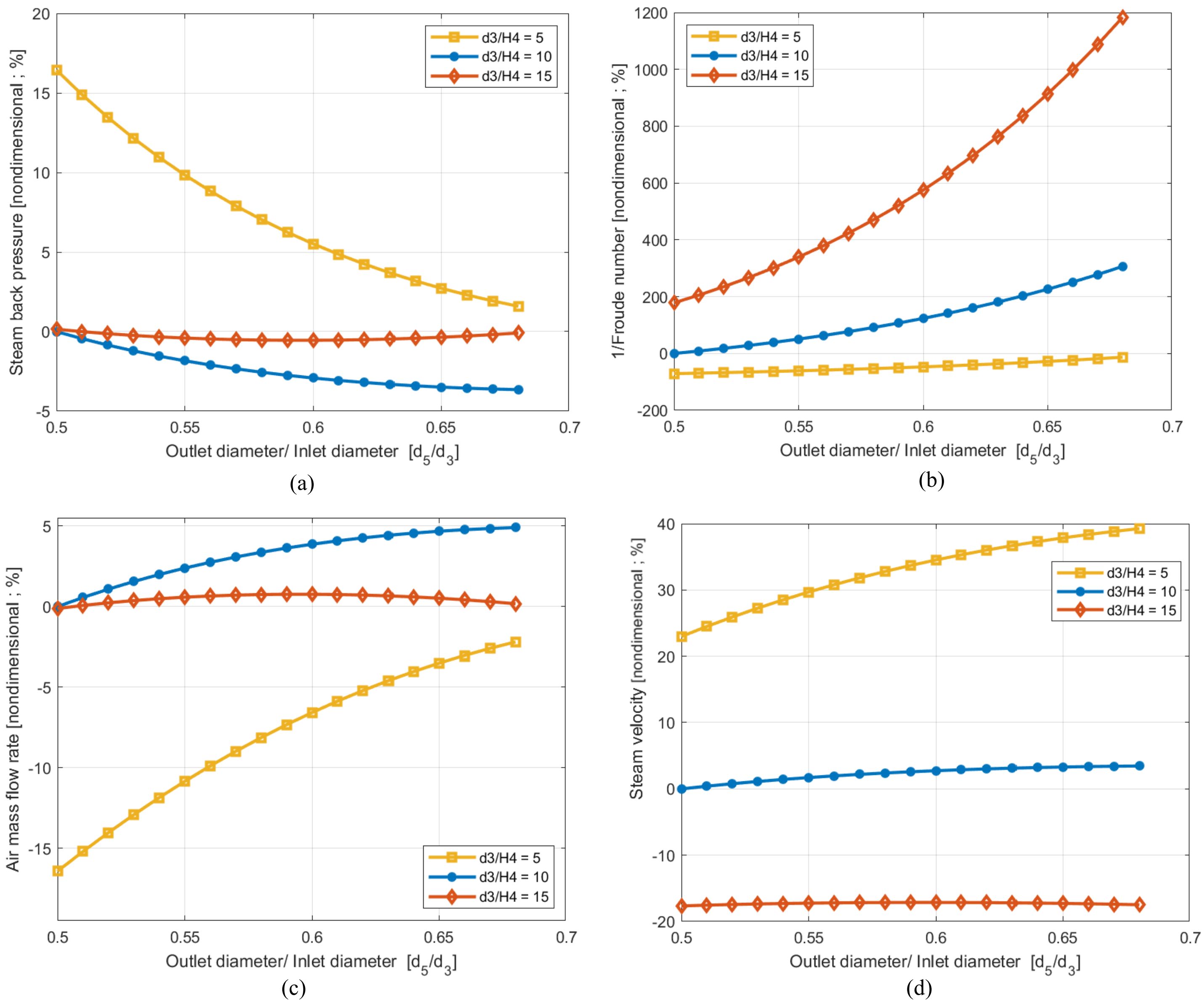


Figure 3: a) Steam back pressure (kPa) b) inverse densimetric Froude number (unitless) c) air mass flow rate (kg/s) and d) steam velocity (m/s) performance trends

5. Conclusion

This study performed a sensitivity analysis for a CSP scale NDDDCS under no wind conditions. Results show that increasing (H_5/d_3) at a constant heat load leads to lower steam back pressures. System performance is dominated by the achievable air mass flow rate, while lower inlet heights resulting in lower steam velocities also carry some benefit. The velocity distribution at the tower outlet is optimized at this scale for a (d_5/d_3) ratio of 0.61. Higher ratios of (d_3/H_4) are susceptible to cold inflow as a result of the lower exit air velocity present. The tower (inlet) loss is significantly lower lower ratios of (d_3/H_4), which is indicative of the negative role that air separation plays in this region. For cases where integrated central receiver / cooling tower solutions (as applied at Khi Solar One) aren't optimal or possible at a CSP plant, this study indicates the reduced size of a NDDDCS that could achieve the desired performance under no wind conditions.

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