Thermal Simulations for Optimization of Dry Transformers Cooling System

Andrea Cremasco, Paolo Di Barba, Bogdan Cranganu-Cretu, Wei Wu, and Andreas Blaszczyk

Abstract An efficient computational model based on principles of thermo-fluid dynamics is crucial for thermal design and optimization of transformers. In this paper we propose a Thermal/Pressure Network (TPN) model of a dry transformer encapsulated in enclosure with natural or forced cooling. The network model has been validated by Computational Fluid Dynamics (CFD) simulations with ANSYS Fluent and then applied for the computation of real transformers, comparing results to thermal measurements. Finally, the parameterized transformer TPN model has been utilized in an optimization loop in order to improve the cooling system. In this respect, the use of a gradient-free optimization algorithm under a multi-objective frame is recommended to avoid local minima and smooth the dependency on the initial guess.

1 Introduction

For air-insulated (dry) transformers, the heat generated in the windings is transferred via convection to the bulk air above and then dissipated to the ambient air through the ventilation system, see Fig. 1. For a numerical simulation of such complex phenomena a very resource demanding CFD analysis is required [1], therefore, designers of transformers typically create their own simplified calculation procedures

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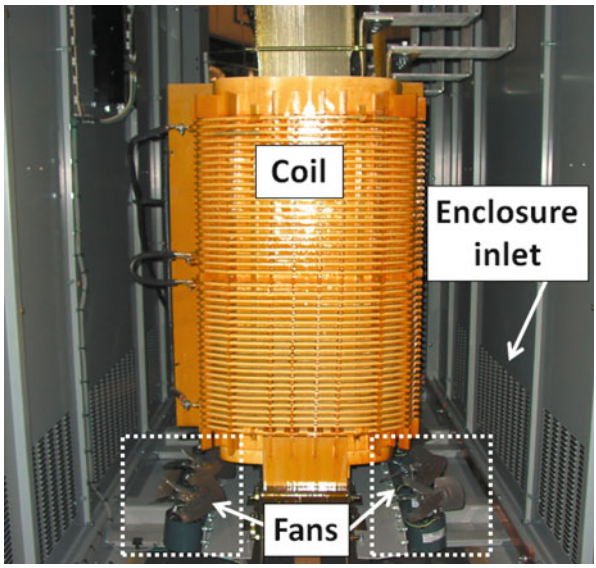


Fig. 1 Dry transformer with ventilation system including fan (which can be switched off) and the enclosure inlet/outlet openings, with grids and optional filters used to protect the transformers against designated conditions

based on empirical assessments and parameters of heat transfer phenomena that are valid for a specific transformer technology [2]. Such procedures are integrated into transformer design systems and used for optimization, thanks to very fast computation times.

In this paper we propose a new method for the thermal simulation of a dry transformer together with the cooling system. The new method is based on a coupled Pressure/Thermal Network (TPN) model, as described in [3]. The new method offers much better computational performance than the detailed CFD; since the TPN method is founded on thermo-fluid dynamics principles, it can be extended to all transformer technologies and cooling configurations including dry transformers as presented in [4].

The basic concept of the network approach is presented in Sect. 2. In this section we included an example for a CFD-based validation of a simple network element representing convection from a vertical wall.

The major new achievement reported in this paper is a network model for a dry-type transformer operated in an enclosure with ventilation openings. The transformer is cooled naturally or by means of fans installed inside the enclosure. The new model has been validated based on CFD computation for a simplified axial-symmetric transformer configuration. With the new network model we could reproduce with reasonable accuracy the fluid flow for different conditions: fans on/off, ventilation openings open/partially closed/closed. We applied the same TPN approach to the computation of real transformers in order to compare results with heat run test measurements. The result of CFD and experimental validations are presented in Sect. 3.

Finally, the fast speed of the network model calculation (less than a second for a typical design) made it possible to apply this method to design transformers in industrial design process and to optimize the cooling system based on multi-objective optimization [5], as shown in an example included in Sect. 4.

2 Network Concept and Network Elements

The TPN method is a lumped-parameter modelling approach based on substituting geometrical parts like windings, cooling ducts, enclosure walls, ventilation openings, fans, etc. by network elements in form of sources, resistors or sub-circuits representing thermo-fluid dynamic phenomena. The basic concepts of TPN, definition of network elements, coupling between the networks and the mathematical background of the solution method have been described in [3]. In Table 1 a short summary is presented.

In order to illustrate how the CFD validation of network elements has been performed, we present here a result for convection from the outer surface of the transformer coil to the bulk air. The height H of the cylindrical coil is variable in a typical range between 500 and 2000 mm. The heat flux $P/A = 150\,\mathrm{W/m^2}$ is dissipated through the cylindrical vertical wall, with radius $r = 350\,\mathrm{mm}$. The goal is calculation of the average temperature of the wall ϑ_{wall} assuming natural convection to the ambient air at temperature $\vartheta_{amb} = 20\,^{\circ}\mathrm{C}$ (radiation is not included).

Table 1 Characteristics of thermal and pressure networks and electrical analogy

Network type	Electric (analogy)	Thermal	Pressure
Quantities, units	Current I (A)	Power P (W)	Mass-flow rate $\dot{m}$ (kg/s)
	Voltage U (V)	Temperature $\Delta\vartheta$ (K)	Pressure Δp (Pa)
	Resistance R (Ω)	Thermal Res. R_t (K/W)	Flow res. S (1/(m · s))
Network principles	Current, voltage law	$\sum_i P_i = 0, \quad \sum_i \Delta\vartheta_i = 0$	$\sum_i \dot{m}_i = 0, \quad \sum_i \Delta p_i = 0$
	Ohm's law: $R = \mathrm{U}/I$	$R_t = \Delta\vartheta/P$	$S = \Delta p/\dot{m}$
Thermo-fluid dynamic principles for network elements evaluations[a, b, c]		Newton's law (convect.): $P = hA\Delta\vartheta$, Stefan-Boltzmann law (radiation), see [6]	Bernoulli's principle: Friction: $\Delta p = 1/2\,\xi\rho v^2$, Buoyancy: $\Delta p = gH_p(\rho_{ref} - \rho)$
Coupling equation		$P = \dot{m}c_p\Delta\vartheta$	

Symbols: h heat transfer coefficient, A heat transfer area, ξ friction factor, ρ fluid density, v fluid velocity (assumed uniform), g gravitation, H_p pressure height, c_p specific heat

[a] In the pressure network only relative pressure resulting from friction and buoyancy is included. Therefore, we can assume that the fluid properties are independent of Δp. The fluid density depends on static pressure according to ideal gas laws

[b] All fluid properties are temperature dependent, which results in non-linear resistors and sources of both pressure and thermal networks

[c] The flow resistance S depends for turbulent flows on velocity, which results in a strongly non-linear behavior of S, the thermal resistance R_t is also velocity dependent, in particular for convection in cooling ducts [4]

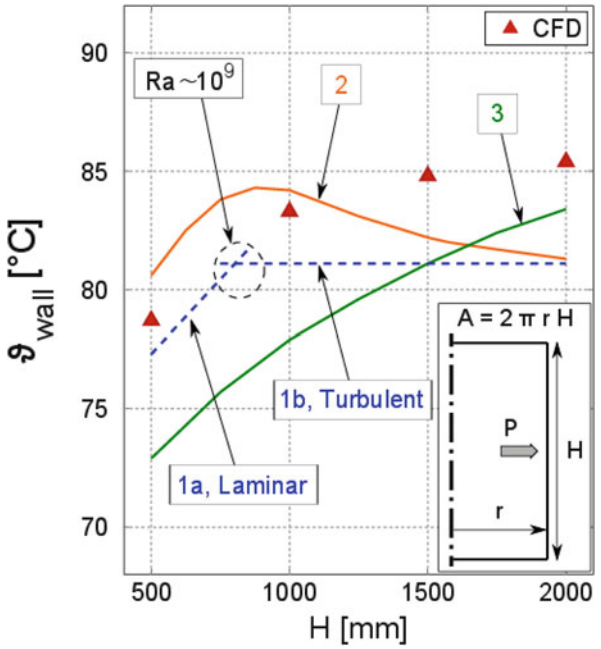


Fig. 2 CFD vs. network results for outer wall temperature of a transformer coil. Wall average temperature: $\vartheta_{wall} = \mathbf{R_t P} + \vartheta_{amb}$ with thermal resistance for convection $\mathbf{R_t} = 1/(\mathbf{h} \cdot \mathbf{A})$ and heat transfer coefficient $\mathbf{h} = (\mathbf{Nu} \cdot \mathbf{k_f})/\mathbf{l_{ch}}$, where k_f is thermal conductivity and $l_{ch} = H$ is characteristic length. The Nusselt number Nu is based on similarity theory, with the following correlations: $\mathbf{Nu} = \mathbf{c_1 Ra^{n1}}$ with $c_1 = 0.54$, $n_1 = 1/4$ for laminar flow, curve 1a (Rayleigh number $Ra < 10^9$); $c_1 = 0.1$, $n_1 = 1/3$ for turbulent flow, curve 1b ($Ra > 10^9$). Curves 2, 3 are based on correlations for constant temperature and constant heat flux models respectively, see Eq. 4.33–4.36 [8]

The results are presented in Fig. 2. The CFD solution is based on heat transfer coupled to Navier-Stokes equations, using $k\omega$-SST turbulence model [7]. The network result, including computation of convection resistor R_t as defined in Table 1, is based on thermodynamic correlations explained in Fig. 2. For all H variations, the difference between temperatures calculated by CFD and the TPN models is less than 5–6 °C, which is still acceptable for applying resistor R_t in the model of the cooling system (see Fig. 4 between HV winding and bypass-duct). Improvements and tuning of R_t can be a subject of future work.

3 Modelling of Cooling System

CFD Analysis and TPN Modelling For the CFD analysis we selected an equivalent axial-symmetric transformer model including a core leg, coil, fan and enclosure with bottom and top ventilation openings, see Fig. 3. The coil consists of a low voltage winding, LV, divided into two radially stacked segments, LV1 and LV2,

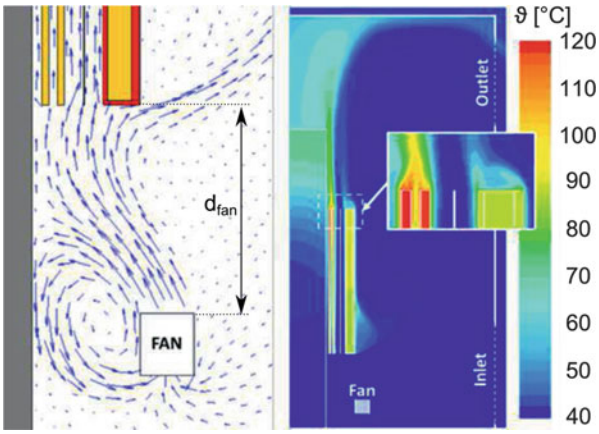


Fig. 3 *Left*: fluid flow recirculation of the fan. *Right*: temperature distribution

a barrier B and a high voltage winding HV casted in solid insulation; the region between the coil and the vertical wall of enclosure is called bypass-duct. The enclosure includes inlet and outlet ventilation openings at the bottom and the top respectively, whose friction was taken into account in CFD with pressure jump boundary conditions [7].

The cold air enters the enclosure from the inlet and flows through the cooling ducts between winding segments; for natural cooling (AN = air natural) the fluid is driven by buoyancy only, while for the forced cooling (AF = air forced) its major part is blown by fans. In both cases, there is air circulation from the bottom to the top of the coil, resulting in hot fluid flowing out from the enclosure through the outlet and taking heat away.

As the main extension of the standalone transformer model investigated in [4], we introduced the "Bypass Duct" as well as "Top" and "Bottom Fluids", see Fig. 4. Together with "Coil Ducts" these elements are responsible for controlling the temperature of the fluid according to the mass and power flow rates in each corresponding network branch (based on the coupling equation in Table 1). The fluid flow direction in the bypass-duct is reversible, see dashed lines in Fig. 4 and the $\dot{m}_{bypass}$ values in Fig. 5. Its direction depends on the performance of the fans and the ventilation grids. Due to reverse bypass flows and the recirculation of the hot air the temperature distribution inside the enclosure can be significantly influenced as shown in Fig. 5.

In the TPN the friction of the enclosure ventilation openings (called here vents) is modelled by a non-linear resistor whose characteristic is based on the equation $\Delta p_{grid} = 1/2\ \xi_{grid}\ \rho\ v^2$ in Table 1. The velocity v is calculated as $v = \dot{m}/(\rho A_{open})$, while A_{open} is the open surface area of the vents. ξ_{grid} is the friction factor of the vents, which depends on construction parameters such as the dimension and shape of the holes, the density of the grid and the presence of filters; these features are related to the Ingress Protection (IP) class of the enclosure [9]: for example dense grids with

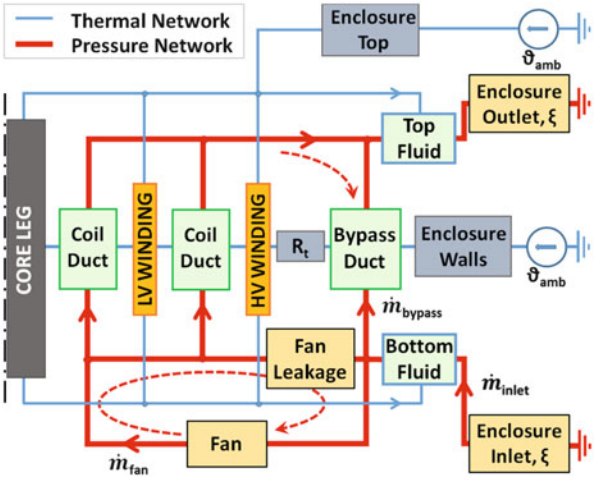
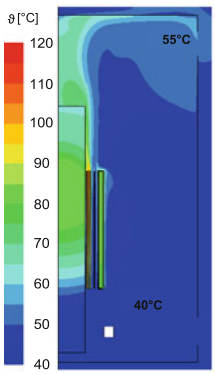
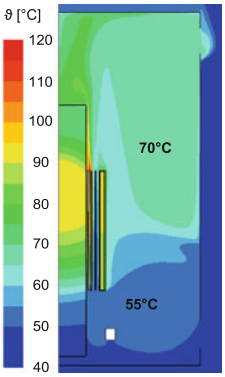


Fig. 4 Concept of the equivalent network model



$\xi^*_{grid} = 10$		
$\vartheta[°C]$	CFD	TPN
LV1	104.8	101.8
LV2	98.6	98.1
HV	81.2	80.0
Core	65.7	60.7
$\dot{m}[g/s]$	CFD	TPN
m_{inlet}	123.4	125.4
m_{bypass}	43.8	41.6
m_{duct1}	8.6	8.9
m_{duct2}	8.0	8.3
m_{duct3}	31.1	31.8
m_{duct4}	32.6	35.0



$\xi^*_{grid} = 200$		
$\vartheta[°C]$	CFD	TPN
LV1	114.3	109.7
LV2	108.5	106.4
HV	89.5	89.1
Core	72.4	67.7
$\dot{m}[g/s]$	CFD	TPN
m_{inlet}	51.2	48.2
m_{bypass}	-23.8	-31.6
m_{duct1}	7.9	8.4
m_{duct2}	7.4	7.9
m_{duct3}	29.0	30.2
m_{duct4}	30.7	33.3

Fig. 5 Temperature maps and results comparison between CFD and TPN model (AF)

filters provide higher IP class but reduce cooling because of the stronger friction. When comparing efficiency of different grids it is convenient to relate the pressure drop not to the open but to the total area A_{tot} of the vent (area that is occupied by the vent in the enclosure wall): $A_{open} = rA_{tot}$, with ratio $r < 1$. After applying this relation to calculation of the velocity and the pressure drop (see equations above) we define an equivalent friction factor $\xi^*_{grid} = \xi_{grid}/r^2$, which has been used for all computations in this paper. The value range for ξ^*_{grid} between 10 and 600 is typical for grids of transformer enclosures in a wide range of IP classes.

TPN Model Validation: CFD and Heat Run Tests In the Figs. 6, 7, 8 we show a comparison between CFD and TPN model results for the average temperature ϑ_{ave} of windings and enclosure walls as well as for the mass flow rate $\dot{m}$ through the enclosure vents and bypass-duct. All the results are referred to the same model

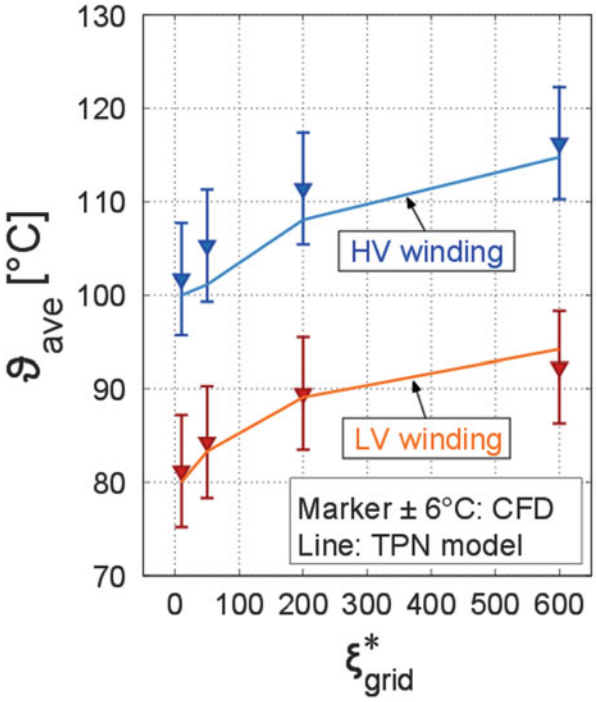


Fig. 6 LV and HV windings average temperature ϑ_{ave}

with load losses, varying only IP class to which different values of ξ^*_{grid} are related; radiation heat transfer has been included.

The temperature deviation is never beyond 6 °C and the CFD trends are followed by TPN, see Figs. 6 and 7. The mass flow rate deviation is always lower than 10 (g/s). The TPN model predicts the distribution of the flow inside the enclosure even when high ξ^*_{grid} limits the flow-rate of the outgoing fluid, causing a downward inversion of the flow in the bypass-duct (this corresponds to a negative value for $\dot{m}$, see Fig. 8).

We applied the TPN method to a real transformer tested with enclosure: the ϑ_{ave} of the windings was derived from electric resistance measurements after reaching the thermal steady state, see Table 2. The deviation from measurements falls into an applicable range of transformer designing.

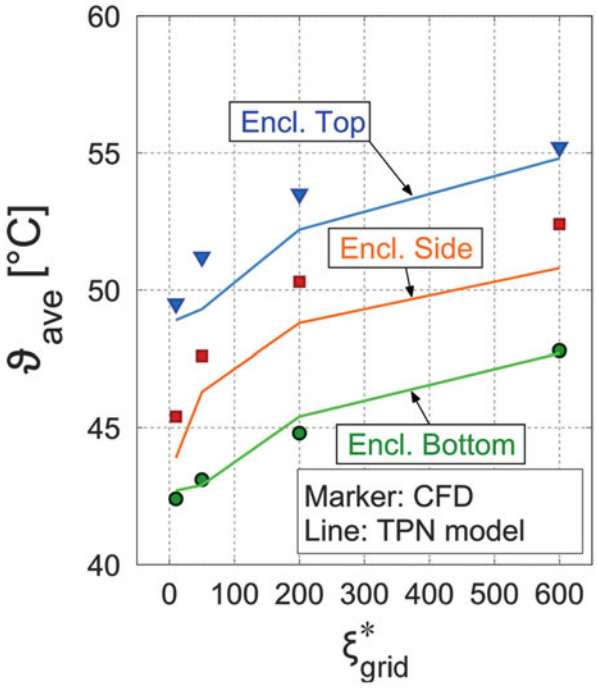


Fig. 7 Enclosure top, side and bottom walls ϑ_{ave}

4 Optimization of Cooling System

In this section we present a formulation of the multi-objective optimization problem applied to the network equivalent model of the dry transformer; the aim is to identify the Pareto front of the non dominated solutions, trading off three design criteria (objective functions). The objective functions to minimize and the design variables are described in Table 3.

In Fig. 9 the 2D projections of the 3D objective space are shown, with Pareto optimal solutions. Results have been obtained by means of the Non Dominated Sorting Algorithm NSGA-II [10]; finding the Pareto front lasted few hours on a standard processor for personal computing.

A posteriori, having identified the Pareto front, the designer can extract a single optimal solution taking into account extra preferences like e.g. the pressure vs. volume flow rate characteristics of a real fan.

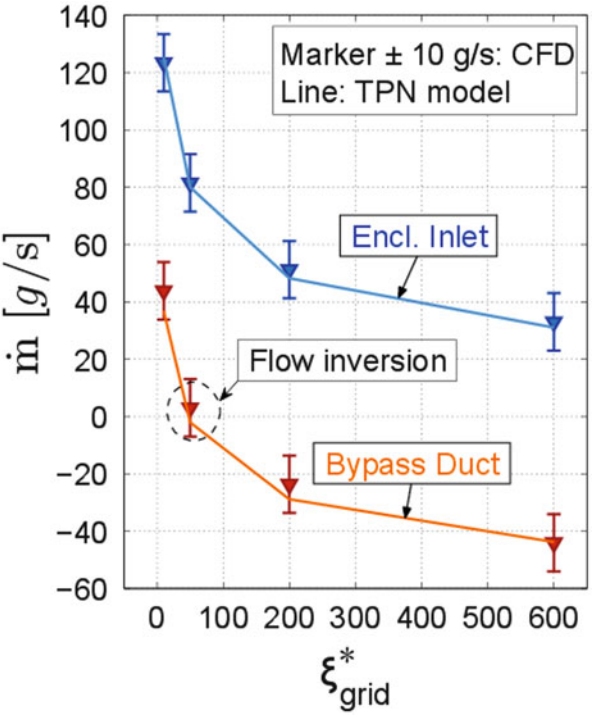


Fig. 8 Enclosure inlet and bypass-duct $\dot{m}$

Table 2 Temperature (°C) comparison between heat run test measurements and TPN model with natural (AN) and forced (AF) ventilation

Ventilation	LV winding			HV winding		
	Measured	TPN	Deviation	Measured	TPN	Deviation
AN	98.1	98.5	0.4	97.8	97.5	−0.3
AF	87.2	83.3	−3.9	95.8	93.5	−2.3

5 Conclusions. Next Steps

In this work we introduced the new model of equivalent Thermal/Pressure networks for dry transformer cooling systems. The CFD validation and comparison with heat run tests confirmed the applicability of the model to a wide range of enclosure Ingress Protection (IP) classes for natural and forced cooling. The new model has been integrated into a transformer design system (used in ABB) and will be a subject of tuning and statistical evaluations based on a large number of transformer designs. The presented application of finding the Pareto front from a multi-objective optimization will be considered as a possible extension of the transformer design system.

Table 3 Formulation of the optimization problem

Objective function to minimize		Description
$\vartheta_{ave} = \frac{\Sigma_k T_{wind,k} V_{wind,k}}{\Sigma_k V_{wind,k}}$ (°C)		Average temperature of the coil: the winding temperatures T_{wind} are weighted by the winding volumes V_{wind}
$\Delta p_{fan} = p_{fan,out} - p_{fan,in}$, (Pa)		Pressure jump provided by the fan [a]
$q_{rec,\%} = \left(1 - \frac{\dot{m}_{inlet}}{\dot{m}_{fan}}\right) \cdot 100$		Recirculation index: if $q_{rec,\%} = 0$, then $\dot{m}_{inlet} = \dot{m}_{fan}$ and there is no recirculation
Design variable	Bounds	Description
Q_{fan} (m^3/s)	(0.10, 0.35)	Fan volume flow rate. Note that there are two fans in parallel per coil, each one blowing the same Q_{fan}
d_{fan} (m)	(0.15, 0.40)	Axial distance of the fan out-take from the coil bottom, see Fig. 3
$k_v = \frac{A_{outlet}}{A_{inlet}}$	(0.7, 1.3)	Vent surface ratio, subject to the constraint on the enclosure design: $A_{tot} = A_{outlet} + A_{inlet} = 7\,m^2$. Vent surface is defined as $A_{inlet} = A_{tot} k_v/(1 + k_v)$, $A_{outlet} = A_{tot}/(1 + k_v)$

[a] The pressure jump Δp_{fan} provided by a fan blowing a certain volume flow rate Q_{fan} depends for example on the vent open surface or the distance d_{fan}. A real fan can supply higher Q_{fan} when lower Δp_{fan} is required, improving the cooling of the transformer

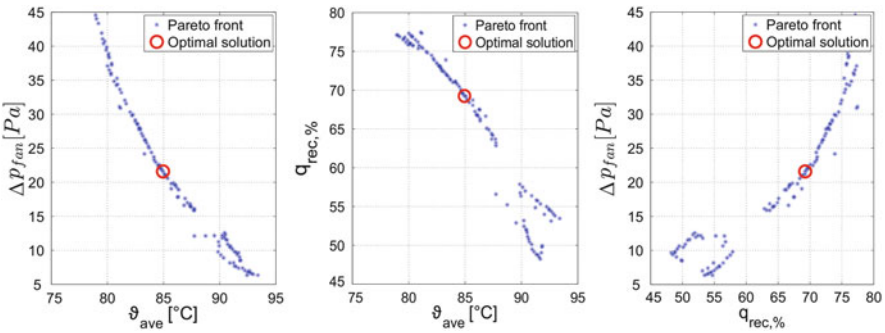


Fig. 9 2D projections of the 3D objective space. For the extracted optimal solution: $Q_{fan} = 0.25$ m^3/s, $d_{fan} = 0.21$ m, $k_v = 1.04$

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