Heat Output from Space Heating Radiator with Add-on-fan Blowers

P-O. Johansson*[1], J. Wollerstrand[2]
[1 & 2] Lund University, Department of Energy Sciences, Faculty of Engineering, Division of Efficient Energy Systems
*Corresponding author: Department of Energy Sciences, Lund University, P.O. Box 118, 22100 Lund, Sweden, per-olof.johansson@energy.lth.se

Abstract: In a hydronic space heating system the heat output arises from natural convection and radiation from the radiators. In order to increase the heat output, without replacing the radiators with larger ones, or increasing the temperature level, the radiator can be equipped with add-on-fan blowers. In the latter case, the heat output from the radiator will increase due to increased convection. This paper describes simulations of heat output from a standard panel radiator before and after it has been equipped with add-on-fan blower.

The radiator model is made in COMSOL, using a 2D-model and the multi physic mode. The model uses General Heat Transfer and two alternate Navier-Stokes tool boxes.

With the model, heat output at different temperature levels in the radiator, and at different fan speeds, can be calculated. The aim with this work is to derive new temperature programs for the radiator at different fan speeds.

Keywords: District heating, Waterborne heating system, Space heating temperature program

1. Introduction

District heating (DH) is common in many countries and it is the dominating heating system in e.g. Denmark, Finland and Sweden. DH consists of centralized heat production units, and a piping system distributing the heat by hot water circulation. Centralized heat production allows use of waste heat from e.g. industries and combined heat and power production (CHP). In Sweden more than 30 % of the DH is produced in CHP [1].

In order to increase production and distribution efficiency the temperature level in the DH network should be kept at a low level. During heating season, the DH supply temperature is described by the temperature demand in the space heating systems of the DH connected buildings. This work describes a method to reduce the temperature demand for a hydronic space heating system with radiators. By adding an add-on-fan blower at the bottom of each radiator the heat output will increase due to an increased convection.

A reduced DH supply temperature will increase the electric production in CHP stations and lead the way towards an increased share of electricity produced by non fossil fuels.

This paper describes the modeling add-on-fan blower attached to an radiator. The goal is to show to what extent such a device can promote the reduction of the temperature program for a hydronic heating system while the heat output is left unchanged.

2. Description of the add-on-fan blower

The simulations described in this paper are based on an add-on-fan blower application mounted below a panel radiator, see Figure 1 for a schematic picture.

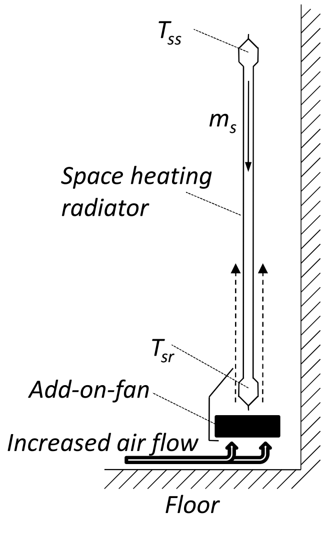

Figure 1. Principle of the add-on-fan blower mounted on a radiator.

The need for calculating and simulating the effect of an add-on-fan blower was discovered in a field study of a commercial add-on-fan blower application carried out by the authors of this paper. The add-on-fan blower in the study was provided by a Swedish company: A-energi AB (the product is called "fläktelement" in Swedish). The company describes the features of the add-on-fan blower as a possibility to reduce the temperature program without replacing the radiators, with the aim to reduce the electricity demand for buildings supplied with heat from heat pumps. Results from the field study are presented in [7].

3. Method

In this study the radiator is represented as a 2-D model. A study of the influence of tuning parameter and mesh size has been carried out. Based on the results from the study a model of the radiator and the fan has been developed. In the studies, temperatures (radiator supply and return) are adjusted so that the total heat output is the same as with a standard temperature program when only natural convection and radiation is present.

To derive new temperature program for the radiator, two different strategies has been investigated.

1. Mass-flow through the radiator is kept constant. This results in a reduced radiator supply and return temperature while the cooling of radiator water is unchanged.
2. Supply temperature to the radiator is constant resulting in reduced mass flow though the radiator. The radiator return temperature will then decrease.

3.1 Limitations and assumptions

There is no water flow in the radiator model. A linear temperature profile in the middle of the radiator is assumed instead. No heat losses from outer wall or the floor are taken into account.

4. Use of COMSOL Multiphysics

In the paper we use COMSOL Multiphysics to model the radiator using the General Heat Transfer and Navier-Stokes toolboxes for a 2-D model. Both Weakly Compressible Navier-Stokes toolbox and the Incompressible Navier-Stokes toolbox have been used. A screenshot of the model is shown in Figure 2.

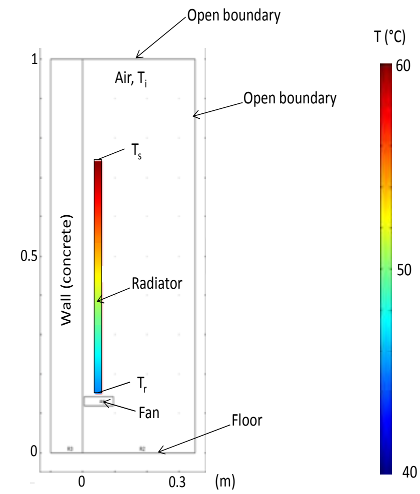

Figure 2. Picture of the 2-D COMSOL model.

In the model, the radiator is placed in a small part of a room, leaving the boundaries to the surrounding room open. The dimensions of the radiator are: height 0.59 m and width 0.02 m. The length of the radiator is 1 m. Heat output from radiation is on the left hand side surface to surface, with the concrete wall as corresponding surface, and on the right hand side, to an ambient temperature, T_i, of 21°C.

The supply temperature (the temperature at the top of the radiator), T_s, is fixed, as well as the return temperature (in the bottom of the radiator), T_r. The temperature profile along the radiator is assumed to be linear.

The fan is represented by a box where above natural buoyancy force an additional vertical force (F_{Fan}) is introduced. The dimensions of the fan in the model is: width 0.09 m, height 0.025 m and length 1 m.

5. Evaluation of model

In this section a comparison between theoretical calculations and simulation results is presented. The radiator is described as a vertical flat plate.

5.1 Heat output from a radiator

The heat output from the radiator arises from convection and radiation. The total heat transfer process from radiator water to the room through a radiator is summarized in equation 1 [2], [3].

$$Q = \dot{m}_s \cdot c_p (T_s - T_r) = k \cdot A \cdot \Delta\theta \quad (1)$$

where k is the heat transfer coefficient describing the convection from the water to the surrounding metal, conduction through the metal and convection from the outer surface of the radiator to the room according to equation 2.

$$\frac{1}{k} = \frac{1}{\alpha_{water-metal}} + \frac{\delta_{metal}}{\lambda_{metal}} + \frac{1}{\alpha_{conv} + \alpha_{rad}} \quad (2)$$

For a radiator application, the dominating parameters in this equation are the convection and radiation between the radiator and the room (α_{conv} and α_{rad}). The other terms, in this case, can be neglected. This results in a new simplified equation for energy output, see equation 3.

$$Q = Q_{rad} + Q_{conv} = \alpha_{conv} \cdot A_{conv} \cdot \Delta\theta + \alpha_{rad} \cdot A_{rad} \cdot \Delta\theta \quad (3)$$

When dealing with a radiator where the supply and return temperature are not equal, the $\Delta\theta$ is the logarithmic mean temperature difference described in equation 4. When calculating a plate with a constant temperature $\Delta\theta$ is the temperature difference between the plate (radiator) temperature and the surrounding temperature (T_i).

$$\Delta\theta = \frac{T_s - T_r}{\ln \frac{T_s - T_i}{T_r - T_i}} \quad (4)$$

5.2 Convection

Heat output from convection arises due to a temperature difference between the radiator surface and the surrounding air. The heat transfer coefficient describing this phenomena is a function of the Nusselt number (Nu), see equation 5.

$$\alpha_{conv} = Nu \cdot \lambda / h \quad (5)$$

Heat output from convection is divided into three different cases: natural, mixed and forced convection.

For natural convection, the Nu number is dependent on the Rayleigh number (Ra) which is a product of the Prandtl number (Pr) and the Grashof number (Gr). For air, Pr can be considered constant, Pr=0.71, and

$$Gr = g \cdot \beta \cdot \Delta\theta \cdot \frac{h^3}{\nu^2} \quad (6)$$
$$\beta = 1/T_{\inf} \approx 1/T_i$$

where g is the gravity force, β is the coefficient of expansion and ν is the kinematic viscosity of air.

Many empirical relations describing the Nu number are available. In this study a relation described by Churchill has been used for natural convection [4], see equation 7.

$$\overline{Nu}^{0.5} = 0.825 + \frac{0.387 \cdot Ra^{1/6}}{[1 + (0.492/\mathrm{Pr})^{9/16}]^{8/27}} \qquad Ra > 10^9 \quad (7)$$

In case of forced convection the Nu number is calculated by equations described by Holman [5], see equations 8 and 9.

$$\overline{Nu} = 0.664 \cdot \mathrm{Re}^{0.5} \cdot \mathrm{Pr}^{1/3} \qquad Re < 5 \cdot 10^5 \quad (8)$$

$$\overline{Nu} = Pr^{1/3} \cdot (0.037 \cdot \mathrm{Re}^{0.8} - 871) \qquad 5 \cdot 10^5 < \mathrm{Re} < 10^7 \quad (9)$$

where the Reynolds number, Re, is described as:

$$\mathrm{Re} = \frac{u \cdot L}{\nu} \quad (10)$$

The product of Gr/Re^2 describes the dominating type of convection. If Gr/Re^2>10, natural convection is dominating, if Gr/Re^2≈1, both natural and forced convection is of importance and if Gr/Re^2<<1, forced convection is dominating. When a mix of forced and natural convection occurs, the Nusselt number is calculated according to equation 11 [6].

$$\overline{Nu} = (\overline{Nu}^3_{forced} + \overline{Nu}^3_{natural})^{1/3} \quad (11)$$

5.3 Radiation

A simplified expression for heat output from radiation is described by Trüschel [3] according to equation 12.

$$Q_{rad} = \alpha_{rad} \cdot A_{rad} \cdot \Delta\theta \approx$$
$$\approx 4 \cdot \frac{\varepsilon_{rad} \cdot \sigma}{\varepsilon_{rad} + \frac{A_{rad}}{A_{radiator}} \cdot (1 - \varepsilon_{rad})} \cdot T_m^3 \cdot A_{rad} \cdot \Delta\theta \quad (12)$$

In equation 5 the temperature, T_m, is the mean temperature of the radiator surface and the surfaces in the room, see equation 13. For a panel radiator the $A_{rad}/A_{radiator}$=1 [3].

$$T_m = \overline{T_{radiator} + T_{room,surface}} \approx \frac{(T_{sf} + T_{sr})/2 + T_i}{2} \quad (13)$$

Since the Arad and emissivity, ε_{rad}, are constant for a specific radiator, the relation can be simplified to equation 14.

$$Q_{rad} \approx C \cdot T_m^3 \cdot \Delta\theta \tag{14}$$

5.4 Validation of model, natural convection

According to a calculation program provided by a panel radiator manufacturer, the total heat output from the radiator with a supply temperature of 60°C and a return temperature of 45°C is 330 W [10], which corresponds very well to simulations performed at the same temperature level, with an emissivity of ε_{rad}=0.95 and ε_{wall}=0.8.

In a study, [8], performed by van der Wijst, the heat output from a vertical plate surrounded by air is analyzed. The plate temperature is set to 40°C. Heat output at different maximum mesh sizes applied to the plate is tested for different rate of isotropic diffusion, see Figure 3.

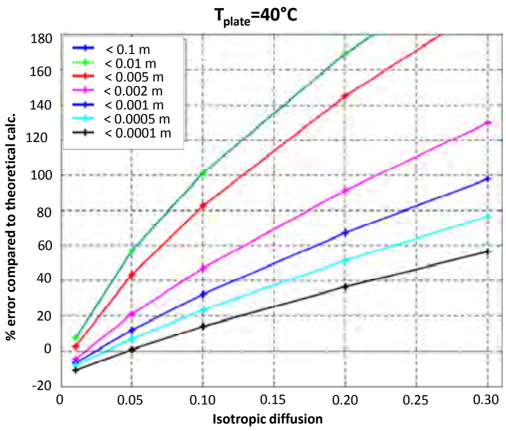

Figure 3. Result of variation of mesh size and isotropic diffusion [8].

In our model we used standard triangular mesh, with a maximum mesh size of 0.002 m on the radiator and 0.01 m in the room. This mesh size has resulted in good resemblance with theoretical calculations for both natural and forced convection by using theory from section 5.1.

5.5 Validation of model, forced convection

In order to validate the model against calculations, the influence of the tuning parameter isotropic diffusion has been in focus. The aim was to create a model giving proper results both with and without an additional fan blower. In this section, the influence of the isotropic diffusion is investigated for various forced air velocities. The simulations in this section are made with a simplified model without the concrete wall replaced with an open boundary and the floor replaced with an inlet at fixed air velocities. The radiator temperature is fixed to be 50°C (T_s=T_r=50°C) and heat output from radiation is turned off. See Figure 4 for a screenshot of the simplified model.

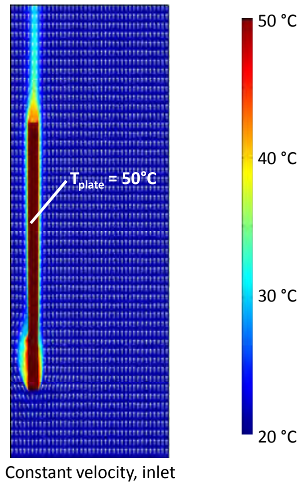

Figure 4. Screenshot of COMSOL model with constant air velocity.

In Figure 5 to Figure 7 the heat output is plotted as function of the isotropic diffusion for three different air velocities. Each figure includes results from theoretical calculations according to section 5.2 and results given by COMSOL built in equations. In case of COMSOL equations, a weight between the influence of natural and forced convection is used by using equation 11. Results from the simulations with both Weakly Compressible Navier-Stokes toolbox (chns) and Incompressible Navier-Stokes toolbox (ns) are presented.

The relative error of the simulation result compared to the results given by equations is also presented. Figure 8 shows the same type of plot, but for natural convection.

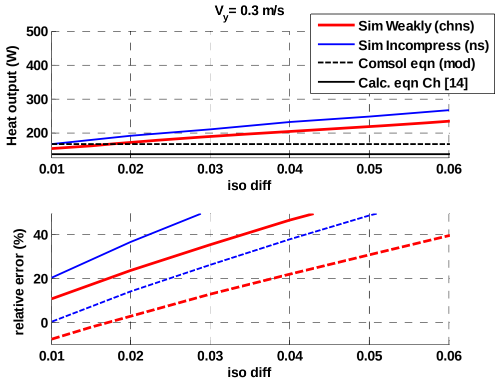

Figure 5. Heat output as a function of isotropic diffusion, forced convection, V_y=0.3m/s.

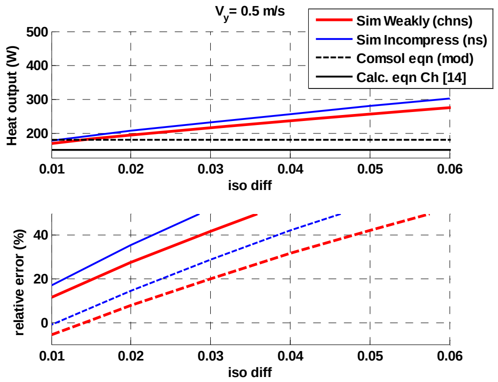

Figure 6. Heat output as a function of isotropic diffusion, forced convection, V_y=0.5m/s.

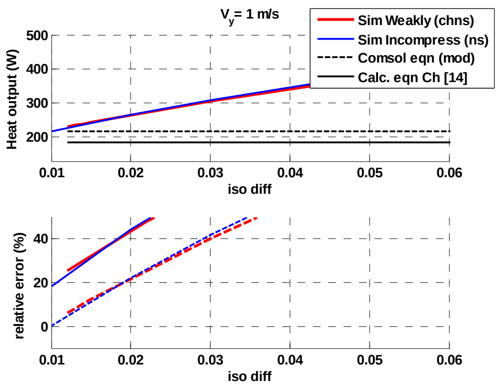

Figure 7. Heat output as a function of isotropic diffusion, forced convection, V_y=1m/s.

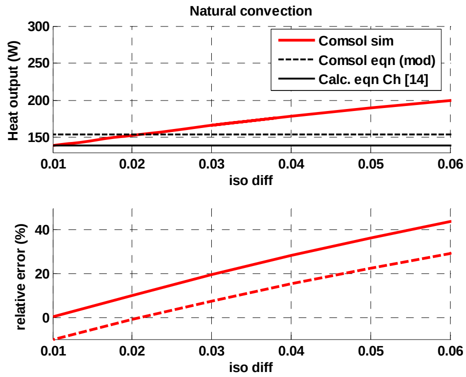

Figure 8. Heat output as a function of isotropic diffusion, natural convection.

As seen, when using a low value of isotropic diffusion, the error could be kept at an acceptable level within the air velocity range investigated. Results from the simulations in Figure 5 to Figure 7 shows that results when using the Incompressible Navier-Stokes toolbox results in slightly better correspondence to theoretical calculations than result achieved by using the Weakly Compressible Navier-Stokes toolbox, especially at air velocities higher than 0.3 m/s.

In further simulations when the fan is active, an isotropic diffusion of 0.01 and Incompressible Navier-Stokes toolbox is chosen. Corresponding simulations did not converge when using the Weakly Compressible Navier-Stokes toolbox for a vertical force in the fan higher than 15 N/m^3.For all calculations, the time dependent Direct UMFPAK solver is used with a simulation time of 259200 s (3*24 h). At that time heat output is stable for all calculations.

6. Results

New, simulated temperature programs for the radiator as a function of the relative space heating load are presented in Figure 9 and Figure 10.

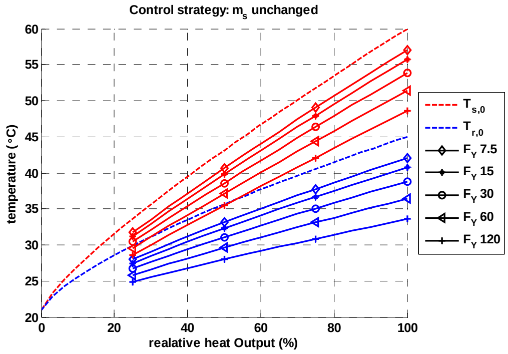

Figure 9. New temperature program for the radiator with a constant mass flow through the radiator. Red lines: supply temperature, blue lines: return temperature.

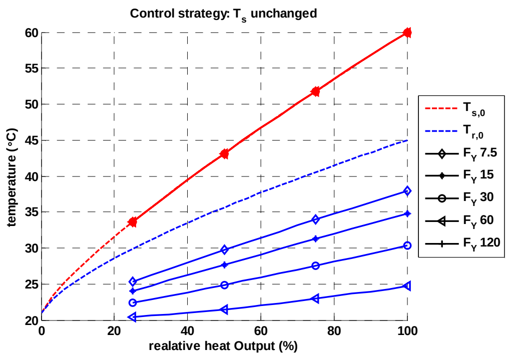

Figure 10. New return temperature for the radiator with an unchanged supply temperature program. Red lines: supply temperature, blue lines: return temperature.

The strategy resulting in constant mass flow through the radiator, see Figure 9, reduces both the supply and return temperature to the radiator with as much as 12°C at full load when high driving force is applied. When keeping the supply temperature program unchanged, and adjusting the mass flow through the radiator instead, the return temperature from the radiator is reduced even more, see Figure 10. However, one has to keep in mind that at very low mass flows through the radiator, the assumption regarding a linear temperature profile along the radiator is not valid any longer.

In Table 1 corresponding air flow through the fan for different driving forces is presented as well as the differential pressure over the fan and the corresponding work of the fan according to the relation: $P_{fan}= \Delta p \cdot V_{air}$.

Table 1: Air flow through the fan at different driving forces.

Vertical force F_y (N/m^3)	Air volume flow V_{air} (m^3/s)	Δp (N/m^2)	P_{fan} (W)
7.5	0.025	0.11	0.003
15	0.036	0.26	0.009
30	0.051	0.55	0.028
60	0.073	1.12	0.081
120	0.104	2.23	0.231

7. Conclusions/discussion

Results from this study show that by using add-on-fan blowers it is possible to reduce the space heating temperature program with several degrees C. The two simulated control strategies show that the return temperature from the space heating system can be reduced in both cases. If the mass flow through the radiator remains constant, the supply temperature is also reduced. Otherwise, with an unchanged radiator supply temperature program, the mass flow through the radiator can be reduced and the additional reduction of the return temperature from the radiator is obtained.

The simulation model shows good correspondence to theoretical calculations within the simulated range. The simulated additional heat output and possible new temperature program calculated according to the strategy with maintained flow through the radiator shows good correspondence to measurements from a field study presented in [7].

The advantage with an add-on-fan blower is that improvements can be achieved without, often costly, modifications of the heating system itself.

It is important to point out that, according to Holman, results from real applications can differ from theoretical calculations with as much as ±25% [5]. On the other hand, the calculations of the relative heat output change, which is in focus in this study, are much more reliable.

8. Nomenclature

8.1 Abbreviations

CHP Combined heat and power station
DH District heating

8.2 Variables

α	Heat transfer coefficient (W/m2.K)	Gr	Grashof number (-)
β	coefficient of expansion (K-1)	h	Height (m)
δ	Thickness (m)	k	Heat transfer coefficient (W/m^2·K)
λ	Conductivity (W/m.K)	L	Length (m)
ε	Emissisivity (-)	$\dot{m}$	mass flow (kg/s)
ν	Kinematic viscosity (m^2/s)	Nu	Nusselt number (-)
σ	Stephan-Boltzman constant	P	Electric power (W)
$\Delta\theta$	Logaritmic mean temperature difference (K)	Pr	Prandtl number (-)
A	Area (m^2)	Q	Heat output (W)
c_p	Heat capacity (J/kg·K)	Ra	Rayleigh number (-)
C	Constant	Re	Reynolds number (-)
g	Gravity force (N/s^2)	T	Temperature (ºC or K)
		V	Velocity (m/s) or volume flow (m^3/s)

7.3 Subscripts

0	Design condition (without fan)	rad	Radiation
i	Indoor (ambient)	rel	Relative
m	Mean	s	Supply
r	Return		

9. References

1. Swedish Energy Agency, Energy Indicators 2008, Theme: Renewable energy, 2008
2. J. A. Myhren, S. Holmberg, Design consideration with ventilation-radiators: Comparisons to traditional two-panel radiators, *Energy and buildings*, 41, p. 92-100, 2009
3. A. Trüschell, Värmesystem med luftvärmare och radiatorer, En analys av funktion och prestanda, *Licentiate Thesis*, Chalmers, Göteborg, 1999
4. S. W. Churchill, Correlating equations for laminar and turbulent free convection from a vertical plate, *Int. J. Heat Mass Transfer*, Vol. 18, p. 1323-1329, 1975
5. J. P. Holman, Heat transfer, *McGraw-Hill inc., 9th edition*, 2002
6. Discussion with Professor B. Sundén, April 2010
7. P-O. Johansson, J. Wollerstrand, Improved temperature performance of radiator heating system connected to district heating by using add-on-fan blowers, *Proceedings of The 12th International Symposium on District Heating and Cooling*, Tallinn, 2010
8. R.P.M.M. van der Wijst, Improved performance of hydronic heating system connected to district heating, *Report to be published, Eindhoven University of Technology,* 2010
9. http://a-energi.jetshop.se/, 20010-09-10
10. http://www.lenhovdaradiatorfabrik.se/display_sub.asp?apid=20, 2010-04-16, Downloaded spread sheet for calculating heat output.

10. Acknowledgements

This work is part of the Primary Energy Efficiency project of Nordic Energy Research.