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Interaction of Heat Load and Air Supply in CAV Systems.

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SYNOPSIS

By means of parametric analyses, the paper describes how the "constantness" of a Constant Air Volume system is affected by temperature differences resulting from heat load variations or otherwise. Several design related parameters are considered.

The paper starts with the background, then an outline of the (simulation based) approach, and how calculations were performed. Results are shown with respect to consequences for volume flow rates and for energy consumption.
The paper finishes with some conclusions, indicating there is a "flaw" in current ventilation standards, that it is hard to make real constant volume system, some preliminary conclusions with respect to energy consumption consequences, and that this type of problem really needs an integral (simulation) approach of building and systems.

1 INTRODUCTION

To minimize the energy consumption related to the distribution of air, a HVAC system could be dimensioned based on constant (and preferably as low as possible) air flows. In such a Constant Air Volume (CAV) design the fresh air flows to the various building zones are supposed to match the fresh air demands. In practice the latter is dictated by governing ventilation standards.
When designing a CAV system, it is common practice to assume constant air temperatures (ie fluid densities). In reality this is not so since the outdoor temperature varies and because the heating or cooling power of a CAV system is controlled by changing the supply air temperatures. Due to buoyancy forces and other density related effects, the air distribution to the various spaces will therefore vary with time.

In order to give some qualitative and quantitative information of the consequences - for flow rates and energy consumption - due to this "constant air temperatures" assumption, parametric studies were carried out based on simulations for a reference CAV HVAC system. These simulations were performed with a modelling environment, which supports analysis of coupled heat and fluid flow.

This paper now continues with a brief outline of the calculation method, a description of the reference CAV system and the results of the parametric studies. It finishes with some conclusions in terms of ventilation standards, flows, energy consumption, design decisions and future work.

2 CALCULATION METHOD

In earlier publications a full account has been given of the internal workings of the ESP^R simulation environment both with respect to energy simulation in general (Clarke 1985) and with respect to simultaneous heat and mass flow simulation (Clarke and Hensen 1991, Hensen 1991). An outline of the approach which is used could be: during each simulation time step, the mass transfer problem is constrained to the steady flow (possibly bi-directional) of an incompressible fluid along the connections which represent the building/ plant mass flow paths network when subjected to certain boundary conditions regarding (wind) pressures and/ or flows. The problem reduces therefore to the calculation of fluid flow through these connections with the internal nodes of the network representing certain unknown pressures. A solution is achieved by an iterative mass balance technique in which the unknown nodal pressures are adjusted until the mass residual of each internal node satisfies some user-specified criterion.

Each node is assigned a node reference height and a temperature (corresponding to a boundary condition, building zone temperature or plant component temperature). These are then used for the calculation of buoyancy driven flows (or stack effect) which are obviously of importance in the current context. Since the approach for buoyancy calculations has already been described in an earlier paper (Clarke and Hensen 1991), this will not be repeated here.

The in ESP^R incorporated flow simulation module *mfs* offers many different flow component types. In the current context, only the flow conduit (duct), flow inducer (fan) and common orifice flow (opening) types were employed. What follows is a brief description of these specific component types, in order to show how fluid densities are taken into account. For a full description of all available flow component types, the reader is referred elsewhere (Hensen 1991).

For fluid flow through a **conduit** (ie. a duct or a pipe) with (a) uniform cross-sectional area, (b) no pressure gain due to fan or pump, and (c) steady-state conditions, the sum of all friction and dynamic losses ΔP is found from:

$$\Delta P = fL\rho\bar{v}^2/2D_h + \Sigma C_i\rho\bar{v}^2/2 \quad (Pa) \tag{1}$$

where f is the friction factor (-), L is the conduit length (m), D_h is the hydraulic diameter (m), $\bar{v}$ is the average velocity (m/s), and C_i is the local loss factor due to fitting i (-).
The local loss factors represent dynamic losses resulting from flow disturbances caused by for example: entries, exits, elbows, bends, obstructions, etc. Numerical values for the local loss factors can be found in literature.

The friction factor depends on the type of flow, which can be characterized by the Reynolds-number: $\mathrm{Re} = \bar{v}\ D_h/\nu$ (–) where ν is the kinematic viscosity (m^2/s).
In ESP^R distinction is made between three regions: for $\mathrm{Re} \le 2300$ laminar flow is assumed, for $2300 < \mathrm{Re} < 3500$ a transition region is assumed, and for $\mathrm{Re} > 3500$ the flow is assumed to be turbulent. In the current context we only deal with the latter case. For turbulent flow the friction factor is calculated from an explicit approximation of the implicit Colebrook-White equation, which is sufficient accurate for most technical purposes:

$$f = 1/[2\cdot\log(5.74/\mathrm{Re}^{0.901} + 0.27\cdot k/D_h)]^2 \quad (–) \tag{2}$$

where k is the absolute wall material roughness (m).

The mass flow rate through eg a duct can now be calculated from a "known" pressure difference by:

$$\dot{m} = A\sqrt{\frac{2\rho\Delta P}{fL/D + \Sigma C_i}} \quad (kg/s) \tag{3}$$

where A is the cross-sectional area (m^2). Because, effectively, we have an implicit formulation for $\bar{v}$, calculation of $\dot{m}$ involves an iterative solution method (fixed point in this case).

Due to practical reasons (ie. lack of detailed data) **fan** performance modelling is usually based on an empirical approach. According to Wright and Hanby (1988), for system simulation studies an empirical model of fan performance is most appropriate. Fan performance is usually characterized by a curve such as shown in Figure 1, which relates the total *pressure rise* to the volume flow rate for a given fan/pump speed and fluid density.

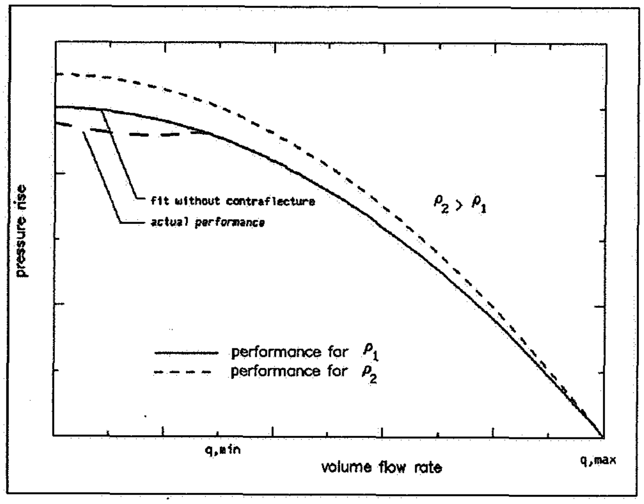

Figure 1 Schematic fan performance curve

As suggested by Figure 1, it is not uncommon for a performance curve to contain points of contraflecture, with up to three different flow rates possible at certain values of fan pressure. This causes difficulty in solving for the flow rate. In practice however, it is usually recommended that the fan operates in the region away from the contraflecture points. Therefore, the flow characteristic may be modeled with a performance curve that does not include the contraflecture as long as it is checked that the fan does indeed operate inside the valid region (if during the simulation the flow rate is outside this region, a warning is issued).
In ESP^R, fan performance is represented by a cubic polynomial:

$$\Delta P_{act} = \frac{\rho_{act}}{\rho_{norm}} \left\{ a_0 + a_1 \left[\frac{\dot{m}}{\rho_{act}} \right] + a_2 \left[\frac{\dot{m}}{\rho_{act}} \right]^2 + a_3 \left[\frac{\dot{m}}{\rho_{act}} \right]^3 \right\} \quad (Pa) \tag{4}$$

and

$$\dot{q}_{\min} \leq \frac{\dot{m}}{\rho_{act}} \leq \dot{q}_{\max} \quad (m^3/s) \tag{4a}$$

where ΔP is the total pressure rise across the component (Pa), act denotes the actual conditions, $norm$ denotes normalized conditions for which the a_i fit coefficients $(Pa/(m^3/s)^i)$, the $\dot{q}_{\min}$ lower validity, and the $\dot{q}_{\max}$ upper validity limit of the polynomial (m^3/s) have been derived.
The term ρ_{act}/ρ_{norm} follows from the so-called fan law (stating $\Delta P_{act} = \rho_{act}/\rho_{norm}\, \Delta P_{norm}$) and corrects for air densities which differ from "standard air" conditions (usually dry air, 101.325 kPa and 20 °C; ie $\rho_{norm} = 1.20\ kg/m^3$).
The equation above requires an iterative approach to determine the mass flow rate for a given pressure difference. In this case, a fail-safe combination is used of bisection method (slow but safe) and Newton-Raphson method (simple and fast).

A basic expression for turbulent flow through relatively large openings (e.g. a purposely provided vent or a restriction in a duct), is the common orifice flow equation. If expressed as mass flow rate this is given by:

$$\dot{m} = C_d A \sqrt{2\rho\Delta P} \quad (kg/s) \tag{5}$$

where C_d is the discharge factor (-), and A is the opening area (m^2).

In the equations above the fluid density and the fluid viscosity depend on the direction of flow, ie. the temperature of the sending node.

3 RESULTS

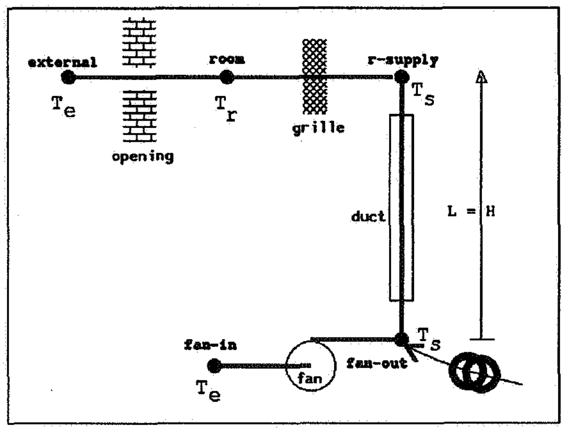

Figure 2 Schematic system lay-out

The present study started from a simplified - part of a - CAV system as schematically shown in Figure 2. The system may be used for fresh air supply and heating or cooling,† and comprises a fan (backward-curved blades) - with a performance characteristic quite like Figure 1 - operating at a fixed rotational speed, delivering maximum 360 m^3/h at zero pressure rise, respectively maximum 125 Pa pressure rise at zero flow (ie coefficients a_0 to a_3 in equation 3: 125.0, 0., -12500., 0.); a .125 m circular duct (wall roughness .15 mm) with a local loss factors sum of 5 (-); an inlet grille specified as an opening of .006 m^2, and an opening from the room to outside of .02 m^2. At the fan outlet, heat is injected or extracted - resulting in a certain supply air temperature - in order to heat or cool the room. There is no heat loss or gain in the duct. The windspeed is assumed to be zero; ie only system and stack effects are considered. Variables of interest here are height H, duct length L, outdoor air temperature T_e, and supply air temperature T_s. Obviously the latter is related to the building heat load; ie. in a CAV system the supply air temperature will have to increase when the heat load becomes larger.

3.1 Air Flow Rates

For this CAV system, simulations were performed in order to predict the influence of the various variables. The results with respect to flow rates are presented in the following. For clarity, only one variable is varied at a time; all other variables are kept at "base case" values, ie: $H = L = 6\ m$; $T_e = 6\ °C$ (close to average Dutch heating

† In heating or cooling mode the air flows would probably be larger than those employed in the present examples. However since these additional flows consist of recirculated air (entering the system at a "constant" temperature) these would only introduce a constant offset in the present calculations and are therefore left out of consideration.

season value); and T_s = 50 °C.

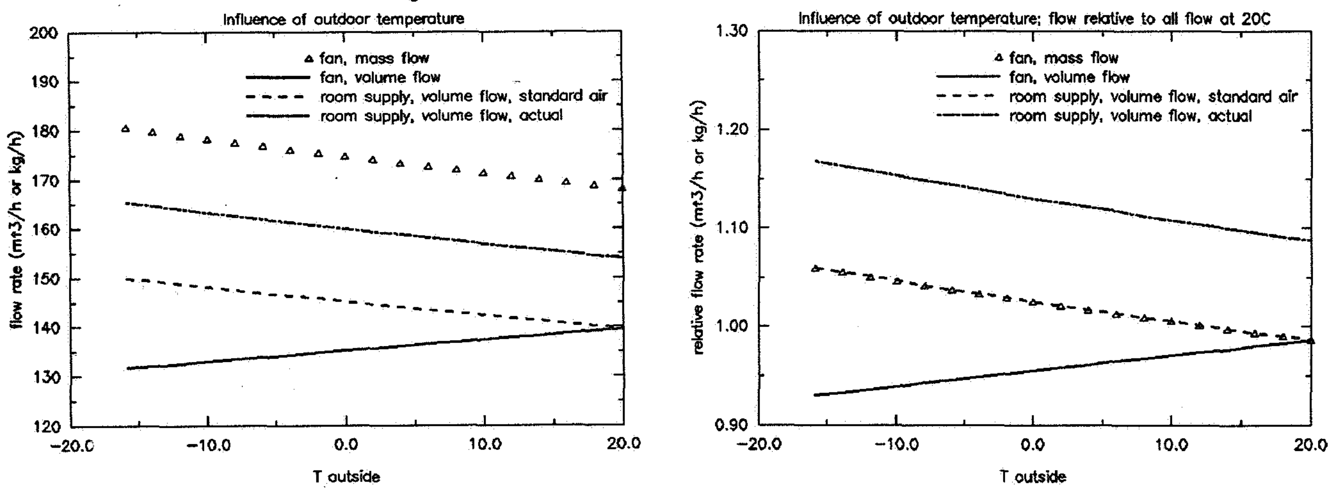

Figure 3 Absolute and relative influence of outdoor temperature;
for $H = L = 6\ m$ and T_s = 50 °C

Figure 3 shows the influence of outdoor temperature variations on mass and volume flow through the fan, and on both the actual and normalized (ie standard air) room supply flow rate. The right hand side figure also shows the flows but now relative to those in case all air temperatures would be 20 °C; ie resulting when density differences would not be taken into account.

It is clear that the flow through this system is affected, although the fan law states that volume flow rate through a fan is independent of density. The latter holds only if the density is constant. The fan volume flow rate variations in the current system are caused by density differences throughout the system. The heat input to the air supply (resulting in T_s = 50 °C) has two opposite effects: (1) a stack effect which has a positive influence on the fan volume flow which increases linear with increasing $T_s - T_e$; and (2) results in a higher volume flow in the duct (and thus velocity and pressure loss ΔP according to equation 1) relative to the situation where the air temperature and densities are constant. This has a negative influence on the fan volume flow rate, but this effect increases quadratic with increasing $T_s - T_e$. As shown in Figure 3 the net result for the current configuration is that fan volume flow decreases when the outside temperature drops.

In terms of room air supply this still results in higher flow rates when the outside temperature decreases (7% higher when T_e = -16 °C instead of 20 °C). Not surprisingly the room supply flow is ≈ 10% higher in reality (ie at 50 °C) than when expressed in terms of standard air (20 °C). It is surprising however that in most ventilation standards (eg ASHRAE 1989) there is no mentioning whatsoever that required flows are (not ?) expressed in terms of standardized conditions.

Figure 4 shows the absolute and relative (again to the all 20 °C case) influence of supply air temperature on the normalized room supply flow rate for various outdoor temperatures (ranging from heating to cooling conditions). Also shown is the case where $T_s = T_e$, ie as in mechanical ventilation. In that case the fan flow rate will be independent of outdoor temperature, which is obviously not true for the normalized supply flows (which are ≈ 293 / T_e * fan volume flow).

For a given outdoor temperature, the room supply flow decreases with increasing supply temperature. When the system is designed such that the flow rates match flow demands

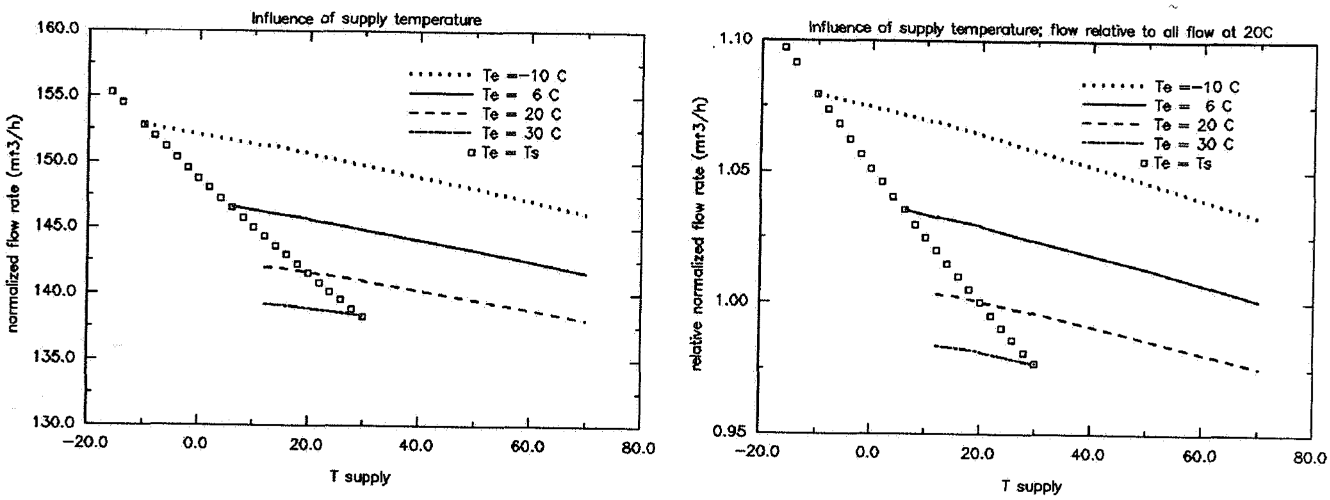

Figure 4 Absolute and relative influence of supply temperature; for $H = L = 6\ m$ and various outdoor temperatures T_e

- assuming all air temperatures are 20 $°C$ say - the actual flows will be up to ≈ 8% too high depending on outdoor and supply temperatures.‡ In heating conditions the situation "improves" (ie flows closer to design values) when the supply temperature increases; in cooling conditions when the supply temperature decreases.

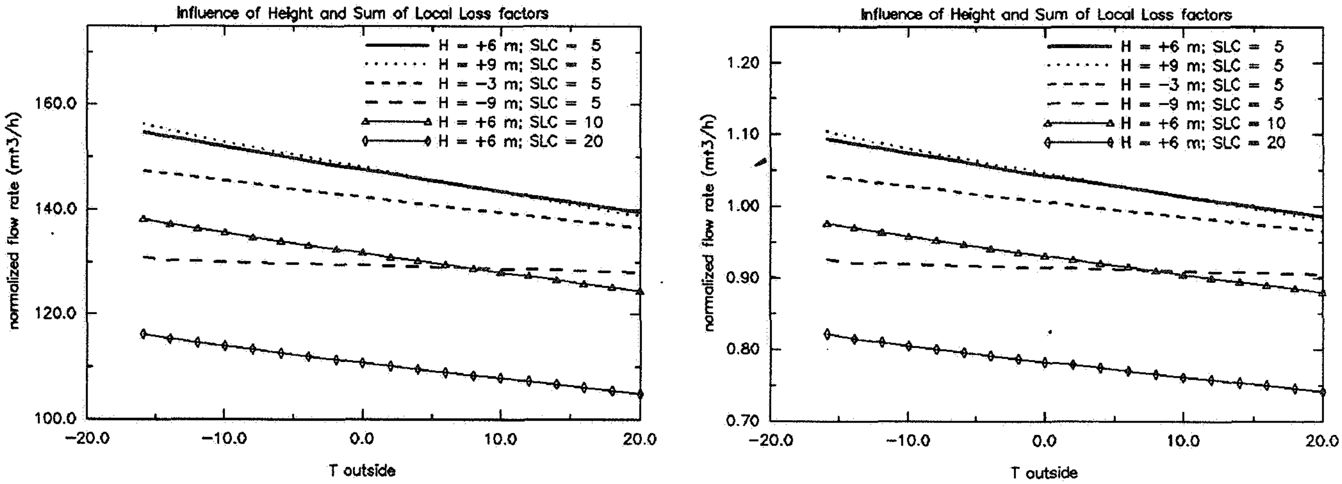

Figure 5 Absolute and relative influence of outdoor temperature for $T_s = 50\ °C$, various heights H (L = abs(H)) and various sums of local factors (denoted SLC)

Obviously the influence of density differences in the system, is affected by system parameters like height, duct length and dynamic losses due to fittings etc. Some indications are given in Figure 5 which shows the absolute and relative (to all 20 $°C$ case) influence of outdoor temperature assuming a fixed supply temperature and various

‡ Note that outdoor and supply temperature do not necessarily correlate. High temporary heat loads (and thus high supply temperatures) may also occur after a thermostat set back period. Also, if the heat input is on/off controlled (as opposed to modulating control), the total heat input period - and not the supply temperature - will correlate with outdoor temperature.

values for the parameters above. Note that the duct length is equal to the height difference. When plotted against changing supply temperature, comparable results would be found since the density difference is the governing parameter. A positive increase of height H has two opposite effects: (1) both duct length and pressure loss increase, but (2) stack pressure (negative for +ve height) decreases. For the given configuration and supply temperature, these two effects almost balance out. Since for negative heights (fan above the room) stack pressure increases, these two effects enhance each other in that case. This is clear from Figure 5, which also shows that the relative influence of outdoor temperature variation (ie the slope of the line) becomes smaller as height becomes more negative. This suggests that in the current configuration it is better to locate the fan and heat supply above the room. We may expect the opposite for cooling conditions (ie locate fan and heat "extractor" below the room). Obviously in case of system location on an intermediate level (servicing spaces above and below; as is common in high rise buildings) at least some of the serviced spaces will be affected all the time (depending on whether in cooling or heating mode the rooms below or above).
Figure 5 also shows the influence of local loss factors. Apart from the obvious decreasing flow rate, an increase in local losses also results in a lesser dependency on outside temperature. But of course, this also results in additional energy consumption of the fan for an equal flow.

3.2 Energy Consumption

In terms of energy consumption, it is obvious that fresh air supply in excess of demand will result in increased energy consumption in case of heating conditions and in case cooling is needed and outdoor temperature is above room temperature. Only when cooling is needed and with outdoor temperature below room temperature, fresh air supply in excess of demand is favourable.
As elaborated in the previous paragraph the actual supply flow rate at any time during the heating / cooling season depends on a number of inter-related factors like: outside temperature, supply temperature, system flow characteristics, system lay-out, etc. The supply temperature depends on the heat load which in turn depends on outside temperature, occupancy pattern, and system control.

It may be clear that in the current context, a detailed performance evaluation of a particular building and HVAC system configuration in terms of energy consumption can only be achieved through an integral building / systems (simulation) approach. At present, work is in progress to evaluate energy consumption related aspects of various CAV HVAC systems when used in domestic or small commercial buildings. Due to time and space constraints this work is not further described in the current paper.

To give some quantitative indication of the additional energy consumption due to excessive fresh air supply: in Dutch low-energy houses and small commercial buildings, approximately 50% of the fuel consumption for space heating is due to ventilation. So if the ventilation would be on average 10% higher than actual demands, the total fuel consumption would be ≈ 5% higher than strictly necessary. This may not seem much, but starting from a low-energy building it is very hard to make some additional savings and a feasible saving will usually be of the same order of magnitude (ie a few percent).

4 CONCLUSIONS

Since the volume flow of air is strongly affected by temperature it is at least surprising that in many ventilation standards there is no mentioning whatsoever that required flows are (not ?) - supposed to be - expressed in terms of standardized conditions. Here it was assumed that in fact "standard air conditions" are implied.

By parametric analyses, it was demonstrated that a real **Constant** Air Volume system is almost impossible to achieve in practice. During the majority of time, a CAV system will either provide too much fresh air which implies waist of energy, or too little fresh air which may have consequences for the indoor air quality.

The relative influence of temperature variations becomes less when the system is above the serviced rooms in case of heating mode. The opposite is true for a system in cooling mode. The relative influence of temperature variations also becomes less when the pressure loss in the (duct) system increases. This will increase fan power consumption however.

Energy conservation by eliminating excessive (ie sup-standard) fresh air supply caused by temperature effects in CAV heating and ventilating systems, may be expected to be in the order of 5% annually in case of low-energy domestic and small commercial buildings in The Netherlands.

Although the results are for an imaginary (but realistic) system the trends are expected to be valid for many CAV systems. The actual air supply rates and energy consumption consequences for a particular building configuration will be the result of many complicated interactions with opposite effects. This makes it extremely difficult - if not impossible - to create simplified design-aids for this purpose. Detailed building performance evaluation can only be achieved through an integral building / systems (simulation) approach.

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