Procedures to identify Energy Conservation Opportunities applied to HVAC system: example of VSD of chilled water pumps

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Julien Caillet[1], Philippe Rivière[1], Jérôme Adnot[1]

[1] Center for Energy and Processes, Mines Paristech, France

Corresponding email: julien.caillet@mines-paristech.fr

SUMMARY

A procedure to identify energy savings of variable speed drive (VSD) applied to chilled water network is proposed. The purpose is to identify on which type of building and climate this opportunity can be preferably examined among the list of Energy Conservation Opportunities (137) identified in the HARMONAC project [1]. First, a preliminary analysis of the likely energy savings is led using two methodologies. A simplified approach is proposed in order to create parametric benchmarks. Results of the parametric simulations for several representative buildings of the French stock can be used to select this ECO according to the approximate potential of energy savings. The effect of appliance loads and climatic condition are examined. These procedures, to identify ECOs on site, could be used to be integrated into inspection procedures across Europe in order to apply Article 9 of Energy Performance of Buildings Directive (EPBD).

INTRODUCTION

A control of HVAC equipment using variable speed drive (VSD) is an Energy Conservation Opportunity (ECO) which can reduce the energy consumption of buildings [1]. Some case studies demonstrate successfully the economical benefit of variable speed drive implementation [2] on various system components: chillers, fans, pumps. This paper aims at proposing a methodology to quantify the potential energy savings of retrofitting chilled water pumping systems with VSD in tertiary buildings.

VARIABLE FLOW IN CHILLED WATER NETWORK

Chilled water networks

The chilled water network is generally constituted by a primary network, where a chiller produces chilled water and a secondary network where terminal units, typically fan coil units (FCUs) or air handling units (AHUs) are used to maintain an adequate level of temperature of each air conditioned zone. This type of layout is common for multi-zone applications and control valves are used to control the building load [3]. The indoor temperature of a room is usually controlled by adjusting the water flow rate passing through the heat exchanger of the terminal units. There are two types of water circuit layout described in the following figures.

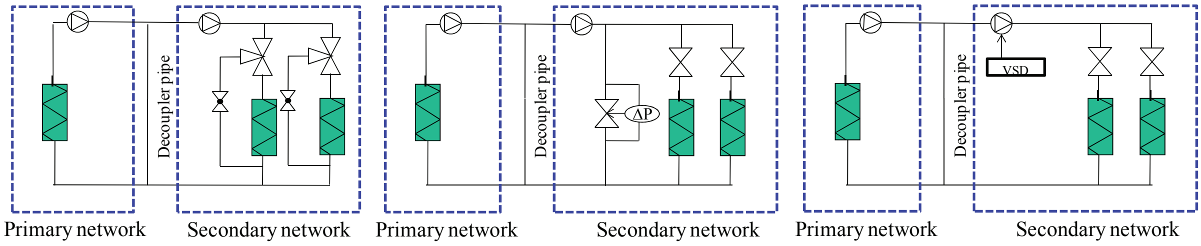

Figure 1 a) Three-way valve b) Two-way valve constant speed c) Variable secondary flow

In this article, the water network is a four pipes system with terminal three way valves (figure 1.a)) or two way valves (figure 1.b) and 1.c)). In figure 1.a), the heat transfer is controlled by a valve. When the valve is closed, the water flow rate surplus is bypassed at each heat exchanger level to maintain the same flow rate in the other branches of the secondary layout. In figure 1.b), an overall bypass is used to maintain a constant pressure difference in the secondary network. Thus, the water flow rate through the bypass pipe is equal to the difference of the water flow rate at nominal condition and the flow rate delivered to the heat exchangers. These two layouts use a secondary pumping system with a constant flow rate.

Three ways to control the flow rate in chilled water network

The basic principle of VSD is to control the pump speed according to the building thermal needs. By adapting the total flow rate (*q)* to the thermal load, the pump electricity demand can be cut. A typical retrofit of these chilled water networks at fixed speed on primary and secondary side is traditionally the general layout presented in figure 1 c) ([4], [5]). The power consumption of the secondary pump for a conventional constant-flow chilled water system is constant. Indeed, if the water network is correctly balanced, the chilled water pump always delivers the same nominal flow rate (Q_n) and the same pressure difference (PD). The electrical pump power P_{pump} is given in equation 1: $P_{pump} = \frac{P_{hydro}}{\eta_t} = \frac{q\ PD}{\eta_t}$ (1)

For all these layouts, the temperature sensor is placed locally in the room or directly in the air stream. A controller compares the air temperature set point and the temperature measurement, and actuates the valve's opening consequently. Centrally, flow rate control of the VSD water pump is based on the pressure difference. The flow rate is controlled either to maintain a constant PD value, a value proportional to the flow rate or a demand based set point. When the system flow decreases, if there is no bypass pipe, the motor follows the pump curve from point A (which represents the original condition of the system curve before valve closes) to point A' of figure 2 a) to 2 c). The VSD controller reacts to PD variation or to the flow rate variation according to the control strategy used. For a constant PD set point (figure 2a), the pump controller selects the new pump speed N2, the operating point is point B, where the pressure difference is kept constant to its nominal value *(*ΔP_n). For a proportional set point (figure 2 b), the pump regulator selects the new pump speed N2, the new operating point being point C, where *PD* is defined using the following equation:

$$PD = \frac{\Delta P_n}{Q_n}\frac{a-1}{a}\,q + \frac{\Delta P_n}{a} \quad (2)$$

By lowering the pressure differential set point as the cooling load drops, the pressure required for high load zones could be underestimated. Thus the value of the parameter "a" is to be kept higher than 2 [6], value kept in this study. For a demand based control (figure 2 c), a pressure set point is selected to fit the network curve. Ideally, this strategy enables to fit the required load without valve control. In other words, the operating point is point D, which belongs to the network curve at nominal conditions.

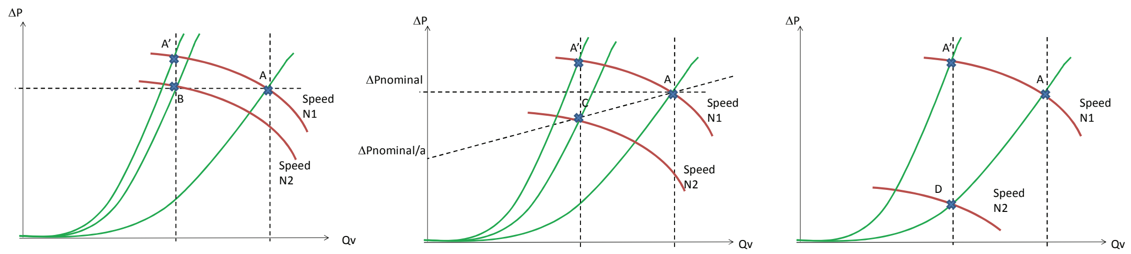

Figure 2 a) Constant PD b) Proportional PD c) Demand based control

Strategies 1 and 2 are the simplest to retrofit a chilled water network as the valves are kept. Only a pressure sensor has to be installed in the branch with the higher pressure drop. For the same flow rate reduction, strategy 3 appears as an ideal control. To approach practically strategy 3, the pressure signal can be reset by demand-control to keep at least one valve opened at or near 100 % as recommended in [7] or if the valve position is not available from the control system, a "trim-and-respond" algorithm can be used as in [8] or in [9] for a reverse return piping network. These practical strategies require a direct digital control (DDC) system.

Targeting some chilled water network

Technically, for a three way valve system retrofit, the valves will have to be changed to adapt the existing system. It can be noticed that this operation is a limiting factor of VSD application. Hence if the system is made of fan coil units, a replacement of all valves will be a limiting factor. For this reason, AHU systems, which have typically larger and fewer terminal units appear to be "better candidates" for VSD retrofit as compared to fan coil systems.

ENERGY SAVINGS COMPUTATION METHODOLOGY

Water network architecture and control

The chilled water flow rate on the primary side is maintained at a constant value. In order to maintain an adequate comfort level in the building, three types of control are used on the secondary side: a local controller to control directly the air temperature of the room and a central controller to control the distribution and the generation of the cooling media. The chilled water temperature is generally maintained at 7°C, hypothesis adopted hereafter.

For a thermal load profile on the secondary side, the thermal energy required to the primary side is unchanged, except that less cooling energy is required because of less heat released by the pump on the secondary water loop; it is neglected in this study. So the chiller consumption is unchanged. Thus, the energy savings are only pumping energy savings.

Method to compute energy savings

A methodology is developed to estimate energy savings obtained by adapting a constant speed pump with the 3 strategies of VSD control described previously. The general approach is given in figure 3.

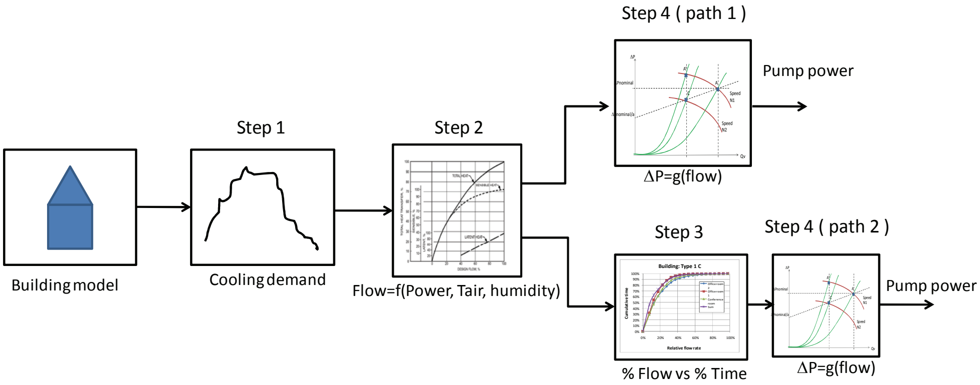

Figure 3 Simulation methodology

The sensible cooling load profiles of typical office buildings are computed (step 1) with TRNSYS [10]. This profile is used as an input of a simplified heat exchanger model [11] to compute the water flow rate required at terminal units hourly (step 2) ; the equation solver EES [12] is used to that purpose. A preliminary step is required to design the heat exchanger i.e. to identify the parameters of the heat exchanger model (heat transfer coefficient ($W/m^2/K$) and heat exchange area in m^2). For each hour, the global heat exchange coefficient is then computed as a function of the water flow rate. Manufacturer data of a model of FCU is used hereafter [13]. Then, the control strategy enables to determine the DP according to the flow rate for each time step (step 4 path 1). In order to study the feasibility to introduce standardized cumulated occurrence curves of reduced flow rates, a step 3 is added (before step 4 path 2). The reduction is made using 5 % length reduced flow intervals (Figures 6 and 7). In order to evaluate the pumping electrical consumption, the total efficiency of the pump is computed hour per hour (path 1) or using the percentage of running time of each reduced flow rate bin (path 2). The total efficiency of the pumping system is expressed as a function of the efficiency of the VSD [14], the efficiency of the electric motor and of the standard hydraulic efficiency [14]. The energy impact of the 3 strategies are shown as a function of the reduced flow rate on figure 4 a) for an AC motor and on figure 4 b) for a DC motor. The electrical power can be compared to [15] in which the strategy 2 is used.

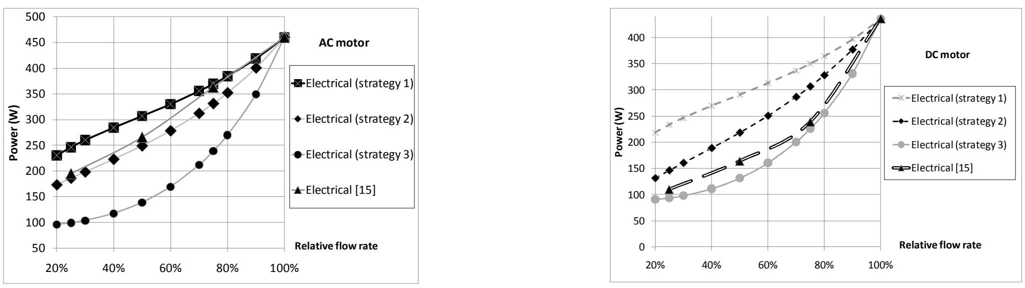

Figure 4 a) Electrical power of the pumping system versus flow (AC motor)

b) Electrical power of the pumping system versus flow (DC motor)

An asynchronous 1 phase motor with capacitor for the AC motor efficiency and a permanent magnet special electronical motor for the DC motor efficiency have been assumed [16]. A load correction factor was used for both motor efficiency models [14]. For an AC motor, the total efficiency model gives a maximum difference of -9%. For a DC motor, the maximum difference is +33%. For the DC motor the load correction factor seems not well adapted. The energy savings are estimated when there is a cooling demand, a minimum flow rate of 20 % of the nominal flow rate is assumed.

Flow computation: methodology 1 (uniform distribution of flow between zones)

For each simulation of building cooling load profile, the number of fan coil units required to cover the maximum building cooling load of each thermal zone is computed. The total cooling load at each time step is divided by the total cooling capacity of FCUs which is directly linked to the cooling capacity of a single FCU, i.e. it is assumed that the total cooling load is uniformly distributed between all the zones. Thus the cooling load per unit is directly used as an input of the heat exchanger model in step 2. The nominal flow rate (Q_n) of a single pump is evaluated to provide the nominal flow rate of all FCUs. This gives the baseline consumption of a constant speed pump which delivers this nominal flow rate at nominal DP.

Flow computation: methodology 2 (distribution of flow according to zone cooling loads)

As the simulation model is multi-zone, the impact of load splitting between all the zones is evaluated. The simulation methodology 2 is presented in figure 5. The design of terminal units is made zone by zone. Contrary to the calculation in methodology 1, the flow rate is evaluated using the cooling load of each thermal zone of the simulated building. The hourly flow rates of the different zones output of the heat exchanger model (step 2) are summed to evaluate the total flow rate (step 3) of the pump.

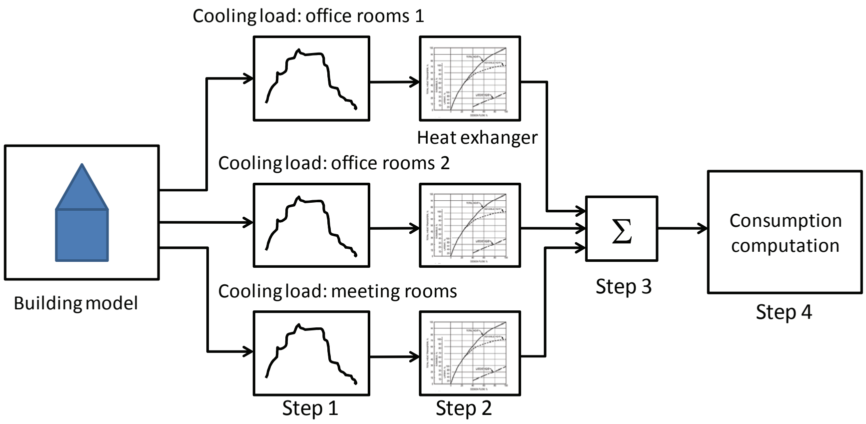

Figure 5 Simulation methodology 2

Buildings selected for simulations

Two representative European office buildings have been simulated. Building shape, envelope properties and profile of occupation are fully described in [17]. The two buildings are constituted of three conditioned zones (2 office rooms and 1 conference room), corridors and toilets. The type 1 C is constituted by 12 identical floors for a total surface of 15000 m² and an average occupation of 1000 people. The type 3 is constituted by 3 identical floors for a total surface of 1000 m² and an average occupation of 65 people. An average internal load of 15 W/m² is used for appliances. For each building type, a hypothesis of nominal DP at nominal flow rate is assumed.

SIMULATION RESULTS

Parametric analysis of simulation results of step 3 and methodology 1

A first pool of simulations was performed supposing the cooling load is uniformly distributed between all the zones. The cumulated occurrence curves at reduced flow rates are given in figure 7a) and 7b) for 4 meteorological files (Paris, Lisboa, Milano and Stockholm) and building types 1c and 3. In both cases, there is a difference of around 10 % of cumulative time between Stockholm climate and Milano climate for a relative flow rate of 20 %.

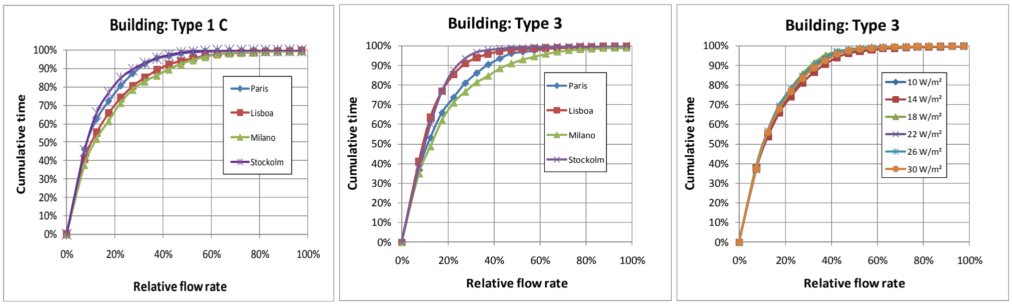

Figure 7 a) Type 1c b) Type 3 c) Parametric study: internal load

A parametric analysis of internal load (from 10 to 30 W/m²) in office rooms is shown in figure 7c). The expected impact is lower than the one of climate.

Analysis of simulation results of step 4 (path 1) and methodology 1

Savings are computed with this methodology for the 4 climates and the 3 strategies. Two scenarii are examined (scenario 1: the motor efficiency remains the same; scenario 2: the motor efficiency is improved by using a DC motor).

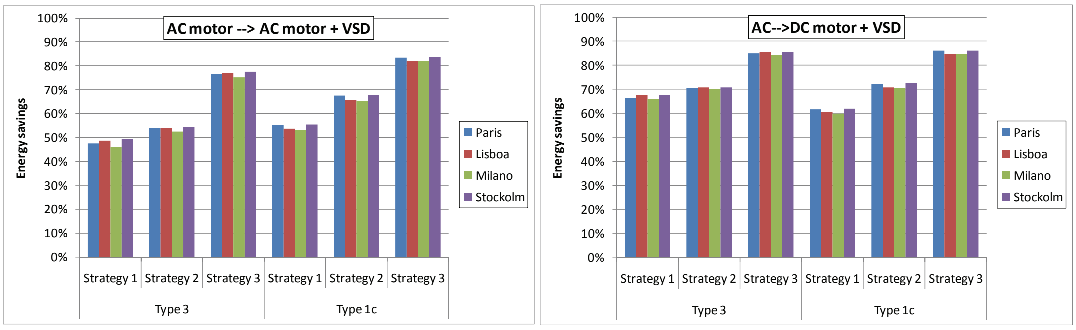

Figure 8 Energy savings of 4 climates and 3 strategies of control and 2 types of buildings

The different VSD control strategies lead to the highest variations in terms of gains as expected from the curves in figure 4. However, the two pressure controlled strategies (1 and 2) lead to comparable gains (between 45 and 65 %).

Energy savings are different for the two buildings. These differences arise from the pump system efficiency increase with their size. Indeed, the total efficiency of the pump is lower for type 3 (1000 m²) than for type 1c (15000 m²), respectively in the range of 0.2-0.4 and 0.25-0.6. It also appears that small pumps with DC motors can contribute significantly to the savings (up to 40 % over simple VSD adaptation of AC motors) for strategies 1 and 2. Finally, as opposed to what could be expected, the energy savings are very close for the four climates.

		Strategy 1		Strategy 2		Strategy 3		Nominal motor efficiency %
		Energy savings %	DC/AC gains %	Energy savings %	DC/AC gains %	Energy savings %	DC/AC gains %	
Type 3	AC/AC + VSD	47,6%	39,75%	53,9%	30,94%	76,6%	11,01%	54%
	AC/DC + VSD	66,5%		70,5%		85,1%		85%
Type 1c	AC/AC + VSD	55%	12,20%	67%	7,22%	84%	2,93%	78%
	AC/DC + VSD	62%		72%		86%		91%

Table Impact of VSD and motor type for Paris climatic conditions

Comparison of methodologies 1 and 2

The methodology 1 assumption of uniform flow rate distribution is evaluated with a simulation of the building type 1c for the Paris climate. The reduced flow rate occurrence curve obtained with the simplified methodology 1 is close to the one obtained with the methodology 2 that takes into account the different flows by zone (Figure 6). The bias is lower than 4.5 % and remains above 1 % for a reduced flow rate higher than 20 %.

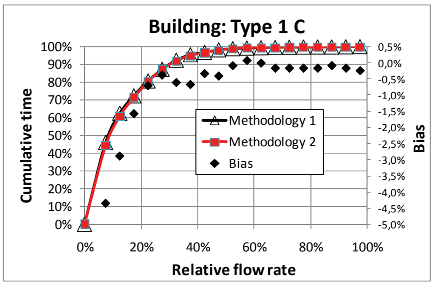

Figure 6 Comparison of the occurrences of reduced flow rates with methodologies 1 and 2

The "least accurate" methodology to evaluate the energy savings is the methodology 1 path 2 and the "most accurate" the methodology 2 path 1. The bias on energy savings introduced by "the least accurate" methodology is presented in table 1 for the 3 VSD control strategies; it is lower than 0.54 % in the worst case.

	Strategy 1		Strategy 2		Strategy 3	
	Energy savings %	Bias %	Energy savings %	Bias %	Energy savings %	Bias %
Methodology 2 (path 1)	54,8%	-0,5%	67,2%	-0,5%	83,3%	-0,5%
Methodology 1 (path 2)	55,1%		67,5%		83,7%	

Table 1 Methodology impact on energy saving estimates for 3 VSD control strategies

Field restrictions

Where some field restrictions could limit the reduction of water flow rate to a minimum as for instance specific zones with constant and high internal loads (e.g. computer rooms), the energy saving evaluation computed before is not available. Indeed the binned occurrence curves are modified. Some assumptions could be made to estimate the new energy savings. For example, a conservative approach for a constant PD strategy could enable to estimate the energy savings without further simulations.

DISCUSSION

A methodology based on a simulation tool is developed to estimate the energy savings of the use of VSD pumping system of the secondary chilled water network. The influence of different climates and building types is shown to have a very low impact on the potential gains, the main parameter affecting the gains being the total efficiency curve of the circulator. A simplified methodology based on binned occurrence curves of reduced flow rates for the whole water network flow rate is compared to the hourly method. The results show an excellent agreement between the two approaches. Additional simulations and parametric studies are still to be performed to verify the accuracy of the simplified methodology and its applicability to a wider set of building and water networks configuration. In that spirit to adapt the method to existing water networks, a second method is proposed to assess the savings when a minimum value of water flow rate is required in one of the thermal zones of the building. The preliminary results suggests that default curves of the cumulated occurrences of reduced flow rates could be used in order to produce estimates of potential gains of the conversion of constant flow secondary system to variable flow systems. Additional required information to produce these estimates can be obtained during a field visit (motor nominal power, nominal DP, nominal flow rate), e.g. during an inspection.

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