ENERGY EFFICIENCY IN HVAC HYDRAULIC SYSTEMS

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Abstract:

When things don't work out as expected, it is said that *"the devil is in the details"*, to be effective we need to *"put God in the details"*.

The ambitious objective of decarbonising the economy, which includes the objective of constructing and operating buildings with a net zero energy balance (NZEB), requires a focus on the details of the systems, without this effort, the fulfilment of those objectives is compromised.

When predicting building's energy consumption using the most advanced detailed dynamic building energy simulation software, the energy use in HVAC system pumps is very residual, typically well below 10% of the energy used in chillers. In existing buildings, the energy use in pumps very often far exceeds the 10% value. This situation has been verified by us in several existing buildings.

This article presents several causes of excessive use of energy in existing HVAC hydraulic systems. Solutions are also presented to optimize energy use in HVAC hydraulic systems, bringing the performance of real systems closer to the theoretically predicted.

Although the focus of the article is on chilled water systems, almost all of what is described is also applicable to hot water systems.

1. THEORETICAL ENERGY USE IN HVAC PUMPS

In a typical HVAC system the energy used in pumps should not exceed 10% of the energy used in chillers, as can be seen from the simplified quick calculation shown in the following table.

Table 1 – Energy use in pumps. Quick calculation.

description	symbol	value	units	formula/source
utilized cooling power capacity	U_c	1.000	W	
piping system losses (pump heat + pipe gain + leakage)	L_c	7,5%	%	admitted [1]
chiller cooling power capacity	CH_c	1.075	W	U_c x (1 + L_c)
water density	ρ	1.000	$kg.m^{-3}$	at typical HVAC temperatures
water specific heat	c_p	4,2	$kJ.kg^{-1}.K^{-1}$	at typical HVAC temperatures
water delta T	ΔT	5,0	K	current value
cooling per (l/s)	cap	21,0	kW	q . ρ . c_p . ΔT / 1.000 ; q = 1 l/s
flow per kW cooling	q	0,048	$l.s^{-1}$	1 / cap
pump design pressure	P	375	kPa	typical value (primary + secondary)
pump efficiency (global)	η	0,60	-	typical value
pump power per kW cooling	P_e	**30**	**W**	q . P / η
average yearly EER	EER_{avg}	3,50	-	admitted
chiller power per kW cooling	CH_e	307	W	CH_c / EER_{avg}
pump power relative to chiller power	p_e	**10%**	**%**	P_e / CH_e

In variable water flow systems, equipped with two-way control valves, variable speed pumps and adequate command and control algorithms, since required pump pressure is much lower than design in most of the operating hours, energy use in pumps should be less than the 10% value calculated above. The application of state-of-the-art pumps, with superior efficiency, and the optimization of piping networks, with less pressure drop, also lead to lower values of energy use in pumps.
In constant water flow systems, equipped with three-way control valves and constant speed pumps, energy use in pumps will greatly exceed 10% of the energy used in chillers, as energy use in pumps will be constant throughout the year, while energy use in chillers will be variable depending on the thermal load at each moment. Furthermore, given the normal oversizing of the secondary piping network in constant water flow systems, as it results from the sum of non-coincident loads, even under design conditions, the secondary water flow will be higher than that determined in the simplified calculation presented in the table above.
In energy monitoring of existing HVAC systems, exceeding or approaching the 10% limit should be treated as an important alert, a sign of anomalies in the design, dimensioning or operation of the system, and cause an intervention to determine and correct its causes.

2. REAL-LIFE ENERGY USE IN HVAC PUMPS.

All existing buildings that we have analysed on site have an energy use in HVAC pumps that exceeds the value of 10% of energy use in chillers. The table below shows data from some existing buildings, whose actual consumptions were recently observed by us, which show an energy use in pumps that varies between a minimum of 11% and a maximum of 56% of the energy used in chillers.

Table 2 – Energy use in pumps in existing buildings.

building	Chiller (kWh)	Pumps (kWh)	Pumps (%)	note
B1	415.039	103.410	25%	yearly consumption
B2	138.000	77.000	56%	June consumption
B3	352.000	39.400	11%	yearly consumption
B4	443	238	54%	27^{th} february consumption

We believe that our experience, naturally from a limited universe of buildings, is representative of the

reality of energy use in existing buildings, however, this assumption must be verified through the observation of energy consumption in a greater number of buildings.
To have efficient buildings, the problem of excessive use of energy in HVAC system pumps needs to be resolved.

3.CAUSES FOR EXISTING SYSTEMS POOR PERFORMANCE

3.1 Operation with low ΔT

Existing HVAC hydraulic systems normally operate with a ΔT significantly lower than design, this phenomenon is the subject of numerous published articles and is commonly called “low ΔT syndrome”. It is common to see systems designed to operate with a 5 K ΔT operating with a ΔT of less than 2 K.
For the same used thermal power, the flow is inversely proportional to the ΔT, thus the operation with a low ΔT requires the movement of a water flow higher than the theoretically predicted.
Low ΔT operation, in addition to the increased energy usage in pumps, due to the operation with an unnecessarily high water flow, also leads to a relevant increase in energy usage in chillers, especially because they must operate at lower evaporating temperatures, reducing its energy efficiency, but also because approximately 90% of the energy used in pumps is degraded into heat in the hydraulic system, adding to the chiller load. In pumps with immersed rotors, motor losses are also transmitted to the circulating water, so that, with this kind of pumps, all the energy used in pumps becomes a thermal load for the chillers.
Next we describe the causes of low ΔT operation, presented in descending order of importance, according to our experience. Typically, in existing hydraulic systems, several of the causes described hereafter occur simultaneously.
The most evident and most impactful cause of low ΔT operation is the use of three-way control valves, since this type of system operates with constant flow and variable ΔT. Systems equipped with three-way control valves must be converted to variable flow operating with two-way control valves.
Variable water flow systems, equipped with two-way control valves, theoretically should operate with a constant ΔT and variable water flow, however, for this to occur, terminal equipments, AHUs, fan coils, induction units, etc., would have to operate under ideal conditions. Several imperfections in the real-world HVAC systems lead to an operation with a smaller than design ΔT.
Perhaps the most frequent cause of operation with low ΔT is the choice by users of setpoints that are too low in cooling, or too high in heating, and unattainable by the installed HVAC equipment. In fact, the most frequent behaviour of system users, when given the possibility to locally adjust the temperature of the space, is to choose the extreme temperatures provided by the local user command interface, i.e., the lowest possible in cooling or the highest possible in heating. I remember a large retail surface, where, noticing that the room temperature setpoint chosen by the operator, on the wall command, was 15 °C, I asked him the reason for this adjustment, having been told that the setpoint was set to 15 °C because the command did not allow choosing a lower temperature.
When the temperature setpoint is unattainable by the installed HVAC equipment, the temperature controller will command the control valve to the 100% open position, keeping the valve in this fixed position until the setpoint is readjusted to a value compatible with the capacity of the installed HVAC equipment. Note that the reduction in ambient temperature will lead to a reduction in the temperature differential between air and water in the coil, reducing its thermal capacity and, consequently, the water ΔT, for a constant water flow. Furthermore, beyond a certain limit, the increase in water flow does not have relevant consequences in the thermal power supplied, so that any increase in flow constitutes a pure loss without any positive effect.
Clogged air filters; dirty coils; coils with deformed fins; poorly tensioned fan belts or obstructions in the duct networks, by reducing the air flow or increasing the thermal resistance in the water coil, reduce the heat transmission coefficient of the coil and, consequently, its thermal capacity, conducting to frequent operation with the control valve 100% open and lower than design ΔT.
In HVAC hydraulic systems in which there is no permanent need for cooling, variable flow chilled water networks, equipped with two-way control valves, may have branches equipped with end-of-line bypasses, whose function is to keep the circulating water at the correct temperature even during

periods of time when control valves are closed. End-of-line bypasses must be equipped with flow limiting valves selected for the minimum water flow necessary to compensate for piping thermal losses; normally the water flow required for this effect is very low. In many existing installations end-of-line bypasses do not have correctly sized flow limiting valves, or even no flow limiting valves at all, leading to a very high bypass flow and a low operating ΔT. The following image shows an end-of-line bypass of an existing HVAC system, where it can be seen that, in addition to the absence of any flow limiting device, the bypass was installed with an excessively large diameter.

Figure 1 – End of Line Bypass in existing HVAC system.

<u>Recommended solutions:</u>

- Replace all three-way control valves with two-way control valves;
- Do not allow users to directly choose room temperature setpoints, for example using +/- 1,5 K commands;
- Implement adequate maintenance routines;
- Application of pressure-independent control valves with ΔT control (this technology is already available from multiple manufacturers);
- Shut-off end-of-line bypasses on all network branches that have permanent use of chilled water;
- Install correctly sized flow limiters on end-of-line bypasses;
- Replace all belt-driven fans with high-efficiency fans, with directly coupled motors, and variable frequency drive.

3.2 Cooling plant schematics

In many existing HVAC systems, anomalies in the conceptual design of the chilled water plant schematics lead to operation with low ΔT. A design anomaly often found in existing HVAC systems is the use of a water tank functioning as a hydraulic decoupler between primary and secondary circuits.

For a water tank installed as a hydraulic decoupler not to impair the system's effectiveness and energy efficiency, it would have to operate with a perfect thermal stratification of the water inside it. The following image represents this ideal situation, for a plant with 733 kW of capacity.

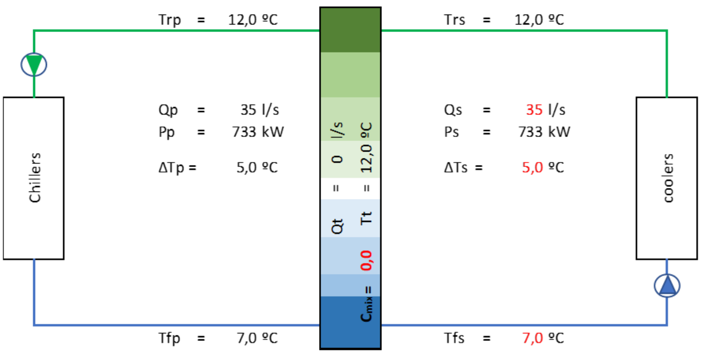

Figure 2 – Tank as a hydraulic decoupler. Perfect stratification.

It should be noted that, since very little mixing occurs in the perfectly stratified tank, only a small percentage of the tank volume contributes to the system's thermal inertia, this is very relevant since the tank is usually installed also with the intention to increase system's thermal inertia.
To have a perfect stratification, special care must be taken on tank design and dimensioning, namely on the following [2] [3]:

- Maximum speed in the intake flow pipes to minimize turbulence.
- Geometry of intake flow pipes to minimize vertical velocities.
- Height to diameter ratio.
- Minimum water ΔT to have an effective thermocline.
- Ratio between mixing forces and buoyancy forces, Richardson number.

Further research on this topic is needed to evaluate the effectiveness of the application of stratified tanks in chilled water systems, in which ΔT is usually 5ºC, or lower, and deliver guidance to tank design.
In all real-life applications, audited by us, with the decoupling tank installed, the decoupling tank operates perfectly mixed, resulting that the temperature of water supply to the secondary equals the temperature of the return water to the chillers.

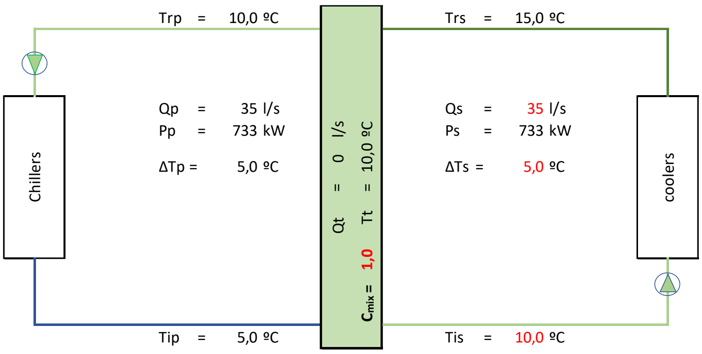

Figura 3 - Tank as a hydraulic decoupler. Perfect mix.

When the decoupling tank operates in a perfectly mixed configuration:

- If the system design considered a perfect stratification, plant effectiveness is compromised since it is not possible to deliver the design water flow temperature for the secondary.
- Chiller efficiency is very low, since chillers must produce chilled water at a much lower temperature than the one required in the secondary system.
- Energy use in pumps is very high since the delivery of chilled water at a higher than design temperature drives secondary terminal two-way control valves fully open to try to reach setpoints.

In existing systems with the above-mentioned issues, the removal of the decoupling tank, and its replacement by a simple decoupling bypass solves the problem and has a dramatic impact on system performance.
Many existing chilled water plants use manifolds with a "U" configuration instead of the correct "H" configuration. Chilled water plants with manifolds with a "U" configuration with the bypass on the secondary side, when secondary water flow is greater than primary water flow, deliver higher than design water temperature to the circuits located close to the bypass pipe, undermining the effectiveness and energy efficiency of these circuits. Chilled water plants with manifolds with a "U" configuration with the bypass on the primary side, when primary water flow is greater than secondary water flow, deliver lower temperature water flow to the chillers located close to the bypass pipe resulting in uneven loading of all chillers.
<u>Recommended solutions:</u>

- Preferably implement solutions with variable primary flow (VPF), without manifolds and equipped with a single pumping group for the primary and secondary.
- In systems with decoupled primary and secondary, use a simple bypass pipe as decoupler.

The decoupling bypass should have the minimum pipe diameter to have close to zero pressure drop for the maximum previewed bypass water flow. The bypass equalizes pressures between flow and return manifolds and allows for a stable pressure irrespective of pump operating state.

- If it is necessary to add thermal inertia to the hydraulic system, do so preferably by increasing the diameter of flow and return manifolds. In existing systems, install the buffer tank in the return line to the chillers or in the line going to the secondary, depending on the desired effect.
- Never install a tank as a hydraulic decoupler between primary and secondary. In existing systems where this solution is implemented, install a bypass line to the tank.
- Investigate the possibility of operating with temperatures and ΔT higher than the traditional ones, i.e., 7°C supply, 12°C return and 5 K ΔT. This option will have to be done before selecting the network terminal equipment, namely AHUs, fan coils, induction units, etc.
- Convert "U" type manifolds to "H" type manifolds.

3.3 Sizing, adjustment and control of pumps

A situation that is often found in real-life systems is incorrectly dimensioned pumps, either with insufficient or with excessive available pressure. When the pump is oversized, it is normal to find a static balancing valve downstream of the pump, dissipating the excess pressure produced and bringing the pump to a stable operating point, within its recommended operating curve. Even in pumps equipped with a variable speed drive, we often find a balancing valve downstream of the pump. It should be noted that this type of valve, even in the fully open position, has a relevant pressure drop.

In most existing variable flow systems observed by us, even when the pumps are equipped with a variable speed drive (VSD), they operate at constant speed, the VSD output being kept constant, normally at 50Hz. An effective control algorithm to readjust the pump rotation speed according to the needs of each moment is often not implemented.

In many systems there is a very high number of pumps, sometimes including tertiary pumps, making it difficult to effectively monitor and optimize their operation.

<u>Recommended solutions:</u>

- Ensure that the hydraulic network head loss is correctly calculated, for example by requiring the presentation of the respective detailed calculations in the design phase and their review by the contractor in the construction works preparation phase.
- Ensure that pumps are correctly selected, involving the respective manufacturer and its pump selection software.
- In all terminals, AHU, fan coils, induction units, etc., apply pressure-independent electronic control valves, with flow reading, indication of the actuator position and communication through protocol and use the valve position information to implement algorithm to automatically reset the pump pressure setpoint according to the critical zone reset principle.
- Ensure that the detailed design specifies an adequate pump speed readjustment algorithm and that this algorithm is correctly implemented on site.
- Replace static balancing valves whose only effective function is to shut-off the line with low pressure loss shut-off valves.

3.4 Incorrectly balanced systems

In most of the existing systems equipped with static balancing valves observed by us, these are fully open and there is no network balancing. The lack of network balance leads to a capacity deficit in some terminals and an excess in others, causing the latter to operate with low ΔT.

We have observed in many variable flow systems, equipped with two-way control valves and variable speed pumps, the existence of static balancing valves, thus, the balance of the network in partial load situations is not ensured.

<u>Recommended solutions:</u>

- In variable flow systems, install at all terminals electronic pressure independent control valves with the ability to communicate via protocol and correctly program the maximum flow limit for each terminal.
- In systems that require static balancing valves, ensure that the balancing procedure in

accordance with ANSI/ASHRAE 111 [4] is applied.

3.5 Control valves that are not water-tight

Most installed control valves are not water-tight in the closed position, allowing for some flow passage.

An extreme situation often observed is the installation of valves with very low maximum closing pressure, more frequent in larger diameters. It is common to find installed control valves whose maximum close-off pressure is significantly lower than the pressure provided by the pump, thus they cannot close the valve in many operating conditions. The circulation of a relevant water flow in coils whose control valve had a closing command is often observed; in four-pipe systems this anomaly is especially harmful.

Recommended solutions:

- Ensure that all control valves have a close-off pressure greater than the pressure of the pump.
- Preferably apply pressure independent control valves with Class A leakage, in accordance with standard EN 12266-1 [5].

3.6 Chillers operating with constant water flow

In most existing HVAC systems, chillers operate with a constant water flow, or only slightly variable, not taking advantage of the possibilities of current chiller technology.

Recommended solutions:

- Preferential implementation of solutions with variable primary flow (VPF), without manifolds and equipped with a single pumping group for the primary and the secondary.
- In systems with decoupled primary and secondary, implement variable water flow in the primary, within the limits indicated by the chiller manufacturer.

3.7 Cooling towers operating with constant water flow

In hydraulic systems serving open cooling towers, operating water flows are normally constant, not varying even in cases where all towers operate continuously and in parallel, cooling the chillers that are in operation at any given time. It should be noted that most open cooling towers available on the market allow water flow turndown up to 50% of its nominal value, in accordance with the requirements of section 6.5.5.4 of ANSI/ASHRAE 90.1 standard [6].

Recommended solutions:

- Select cooling towers that allow 50% water flow turndown.
- In systems with multiple towers and multiple chillers, always operate with all available towers, varying the rotation speed of the respective fans to control the temperature at the tower outlet, and vary the water flow in the towers up to the lower limit of 50% of the nominal flow depending on the flow requirements of the chillers in operation at any given time.

12. CONCLUSIONS

By designing, sizing, installing and operating HVAC systems with attention to the details described in this article, we believe it is possible to approximate the performance of real systems in operation to the predicted theoretical performance.

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