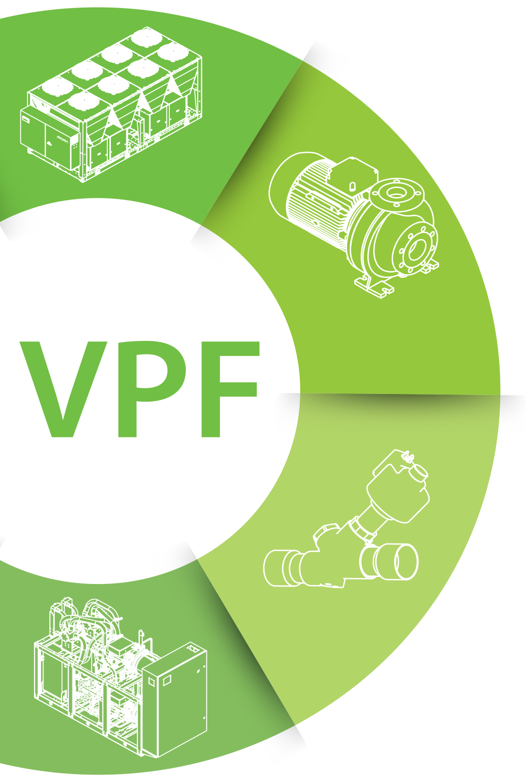
AERMEC
air conditioning

TECHNICAL FOCUS

VPF
VARIABLE PRIMARY FLOW



HYDRONIC SYSTEMS WITH VARIABLE PRIMARY FLOW

Closer attention towards reducing energy consumption by air conditioning systems, linked with the search for savings in plant operating costs and the need to reduce CO2 emissions, is encouraging builders of such plants and heating technology designers to examine new and alternatives solutions to the approaches used for decades that, while still valid, are no longer taken for granted. Within the scope of hydronic air conditioning, the innovations most often analysed in recent years include the variable primary flow systems (VPF); the development of these systems began in the USA a few years ago and was made possible thanks to technological improvements of variable speed pump units and the implementation of increasingly advanced control logic to manage modulating chillers.

Technical Sales Support Department

SUMMARY

INTRODUCTION

Obviously, in most plant installations, the extension of circuit sections dedicated to distribution (secondary circuit) is much larger than those located in the hot-cold system storage (primary circuit), so that much of the energy saved on pumping is generally already achieved by modulating water flow on the secondary circuit; the additional energy and cost savings associated with variable primary flow, albeit lower in percentage terms in relation to overall system energy costs, mean that the performance of already efficient plant can be improved, and is an energy efficiency measure that generally does not involve particular financial overheads.

This innovation in the way systems are designed directly involves manufacturers of conditioning machines, given the considerable problems that may arise in the management of one or more coolers carrying a variable water flow during operation.

We will examine the general state of the art, that is the most common plant solutions with chillers and heat pumps connected to the VPF schema; we will highlight the main expedients to be adopted when selecting machines, in system design and fine adjustment of the support system, and we will assess some of the potential economic and energy savings, with estimates based on a reference plant. Over and above general considerations and the example in question, it is important to bear in mind that, when adopting this new plant, each single case must be analysed individually, examining the critical aspects and developing necessary feasibility and cost effectiveness assessments based on the specific machines selected and the configuration and extension of the plant.

The modulation of the water flow developed by the pumps in these systems as the load varies (made possible by using two-way modulating valve on the heat exchange batteries on the UTA and/or 2-way ON/OFF valves on fan coil units) brings about, as regards full load pumping costs, a significant reduction. Mention need only be made in this regard of the fact that the power absorbed by the pump is proportional to the flow rate developed and the head supplied to the fluid treated:

$$P_a = \frac{Q \cdot h \cdot \rho}{\eta}$$

As is well-known, losses in an hydraulic system are related to the square of the flow rate of the fluid in transit:

$$\Delta p = k \cdot {Q_w}^2$$

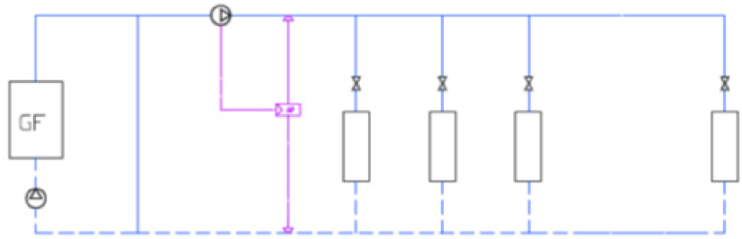
Key:

- Q Flow
- h Useful head
- ρ Density
- η Performance

Inasmuch, a 50% reduction of the rated value of flow rates helps reduce pressure losses in the shared sections of the circuit by about 25% of rated value; it is evident that, despite the penalty as regards pump performance in relation to its rated value, the reduction in pumping costs is far from negligible.

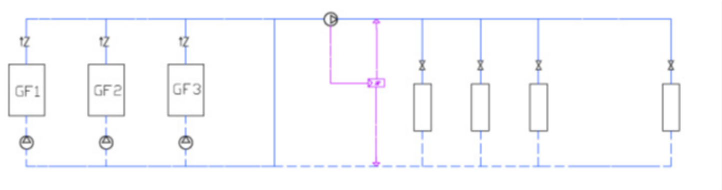
Chapter 1
HYDRONIC REFERENCE LAYOUTS

There follow the two most common plant layouts that have always been used for hydronic summer and/or winter air conditioning systems (the former with a single chiller, the second with several chillers in parallel).



Scheme 1-a: double ring system with single chiller.

As known, these double-loop schemes involve the existence of two circuits (the primary circuit, usually localized in the vicinity of the central heating and cooling, and the secondary circuit, to the service of the users.



Layout 1-b: Double loop system with several parallel chillers.

AERMEC MULTI-CHILLER SISTEM

Display pGD³

Electronic board pCO1

The main reason for this attention is associated with the behaviour of the refrigerant circuit during a transient with a variation in water flow; the groups can usually accept water flow rates with values in a fairly wide range (the maximum range generally corresponds to ΔT values of about 3 °C, and the minimum flow rate corresponds to ΔT values of around 10 °C for operation at full load); if the water flow is too low, this would create a laminar flow inside the heat exchanger, with a significant reduction of heat exchange coefficients; if the water flow rate is too high, this would cause vibrations and erosion of the heat exchanger's surfaces.
There are no problems if flow variations occur with the chiller turned off or if they occur when the chiller is on but in a sufficiently gradual way and always within the limits recommended by the manufacturer. The most critical situation arises when this takes place abruptly; this would bring about variations in cooling circuit operation that the thermostatic valve would tend to offset (but not instantaneously) and that might even causes harm to the compressors.

A sudden increase in water flow will instantly increase heat exchange on the evaporator and, during the transient, this means greater frictional overheating and a slight increase in evaporation temperature; this situation is not usually a risk for the compressors. A sudden drop in water flow, on the other hand, reduces the heat exchange to the evaporator, instantly reducing the overheating that could cancel itself; this situation is potentially hazardous for the compressors because it could cause a fluid return.

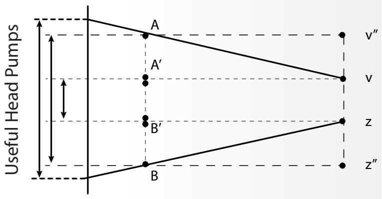
An electronic valve instead of the mechanical thermostatic more rapidly offsets any water flow disturbances but to be sufficiently precise and avoid oscillations it must also act for sufficiently long times, usually a few dozen seconds; in the case of plant involving flow variations on the chillers, this must be suitably foreseen, but the expedient is not itself sufficient to protect the chiller against every critical aspect.

In particular, additional measures will have to be envisaged to ensure the transit of a minimum water flow on every chiller in operation to avoid flow alarms or compromise heat exchange, and make sure that any flow variations on the machines are sufficiently gradual; in this regard, it is considered prudent to set an adjustment on modulating organs that ensures a maximum reduction in the water flow rate of to 10% of current value per minute.
The most widely used solution is to install a by-pass branch with a modulating valve in the immediate vicinity of the hot-cold plant to ensure the required minimum flow rate, as better illustrated in the following layouts.

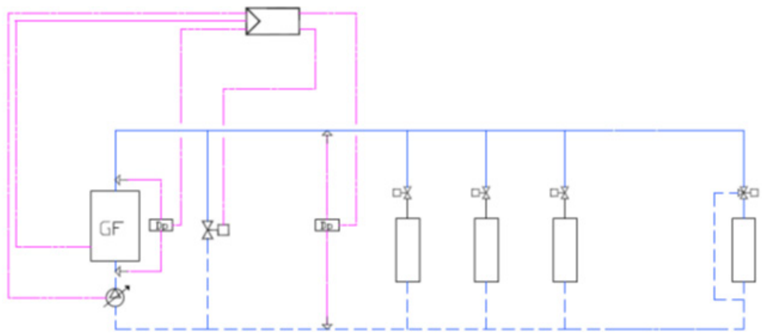
For this reason, management of variable flow systems requires new integrated management logics, often implemented by specific additional cards , where compressor capacity control is always obtained by adjustment of the delivery temperature, but specific system controllers are able to coordinate action on hydraulic circuit actuators (pumps, by-pass valves) with adjustment of chiller processors.

Thermostatic Valve Electronic

Performance profile of the pressures to changing the probe positions.



There follow a number of plant layouts envisaging variable flow on chillers and a brief description of their management logic.
The first layout (layout 2-a) refers to a single loop variable flow system with a single chiller.
System management is integrated into chiller processor functions or entrusted to an auxiliary controller normally supplied by the manufacturer and is able to interact with the water chiller, the pump unit and a modulating valve positioned on a suitably positioned by-pass; the flow of water sent to user points is modulated in relation to effective demand from the terminals through a differential pressure adjustment.
Pump speed is adjusted in such a way as to maintain a constant pressure differential between the delivery and return on the user circuit in two appropriately chosen points; the most appropriate choice of pressure probe position is based on considerations by the designer associated with the structure of the specific plant.
The chiller, which in turn modulates the cooling capacity delivered in relation to delivery temperature, nonetheless requires a minimum smooth flow value through the heat exchanger to be maintained (usually in the order of 50% of rated flow), whereby the system controller monitors this flow through a second differential pressure detector located in a position between upline and downline of the evaporator; on reaching the minimum permitted flow on the chiller matches the minimum pump speed, which is not further reduced; a further flow reduction requested on user points is achieved through the gradual opening of a two-way modulating valve located on a by-pass in the immediate vicinity of the chiller. An entirely similar logic manages the operation of a single loop variable flow system with multiple chillers in parallel (layout 2-b); as the flow rate requested by user points varies (which depends on the degree of opening of the two-way modulating valves and/or the number of two-way ON/OFF valves open, managed by specific local thermostats), the controller interacts with the inverter pump to modulate rpm and, maintaining operation with capacity control of both chillers, each carrying a reduced water flow; in this way, energy benefits are achieved in relation to capacity control operation of the chillers and the advantages of reducing pumping costs required even to pass through the machines and associated components, in traditional systems mounted on the primary circuit.
On reaching the minimum permitted flow rate on the chillers, as appropriately set on the controller through a minimum differential pressure value on the evaporators, the controller acts to manage the opening of the by-pass valve and gradually turn off chillers with the subsequent closure of the corresponding branch; the gradual shut-off and cut-out of the chillers helps modulate the overall capacity in relation to maintaining the minimum flow on those still in operation.



Scheme-2: a variable capacity single-ring system with single chiller.

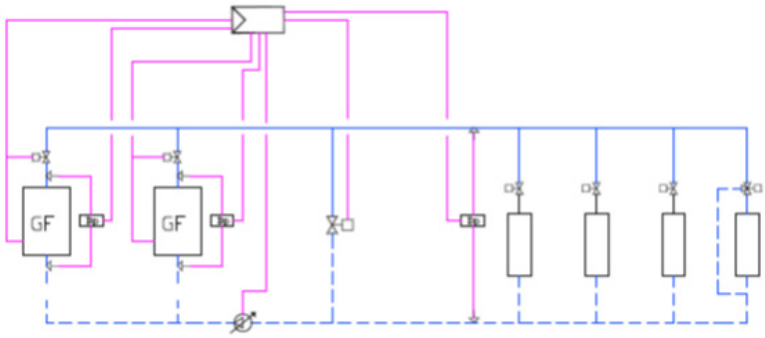


Diagram 2-b: variable flow single-ring system with more refrigeratori.

These solutions bring about energy benefits to be evaluated on an individual basis in relation to system architecture, the type of machine chosen and the seasonal and climatic load profiles of the place where the system is installed.
There follows a case study involving a single variable flow loop with a chiller equipped with screw compressors and tube bundle evaporator, in summer cooling operation in an office building, in order to provide an idea of the level of savings that can be achieved, and the cost effectiveness of the VPF solution compared to more conventional approaches.

Chapter 2
ANALYSIS OF A CASESTUDY

Example of a local office use

As already mentioned, in order to assess the potential economic and energy savings of a variable primary flow solution compared to more conventional approaches, we analysed summer air conditioning in an office building served by a dual-pipe system; the system, powered by a specific heating boiler during the winter season, is served by a chiller with screw compressors and tube bundle heat exchangers.
There follow the characteristics of the building in question:
Location: Roma
Outside design temperature in summer: 35°C / 95°F
Outside design relative humidity: 45%
Destination: office use
Plant activation: 5 days/week from 07.00 to19.00 in summer (May-September)
Floor area: (9688 sq.ft/floor - 900 mq/floor), 7 floors; Total = 67812 sq.ft. / 6300 mq
Type of system: primary air + fancoil
Peak cooling load: 713 kW / 202 Tons

NSM3202X°°E°°°00

Simulation performed using the Magellano selection program, available for download from the support site of Aermec.

Chiller: AERMEC NSM3202X°°E°°°00, selected under the following conditions:

Cooling Mode		
Cooling Capacity	kW / Ton	740,55 / 202
Cooling Power Input	kW	238,58
Cooling Current Input	A	389,15
E.E.R.	W/W	3,10
E.S.E.E.R.	W/W	4,07
Inlet air temperature dry bulb	°C / °F	35 / 95
Water temperature input	°C / °F	12,19 / 54
Δt	°C / °F	5,19 / 9,4
Outlet water temperature	°C / °F	7,00 / 44,6
Ethylene glycol	%	0
Water flow	l/h / gpm	122763 / 85,4
Pressure drop	kPa / psi	23,89 / 3,46

The rated data for the hydraulic circuit, referring to full load operation, are as follows:

Solution1(conventional plant):

Primary circuit:

Waer flow	mc/h	123,8
	gpm	546
Pressure drop	kPa	48
	psi	6,96

Secondary circuit:

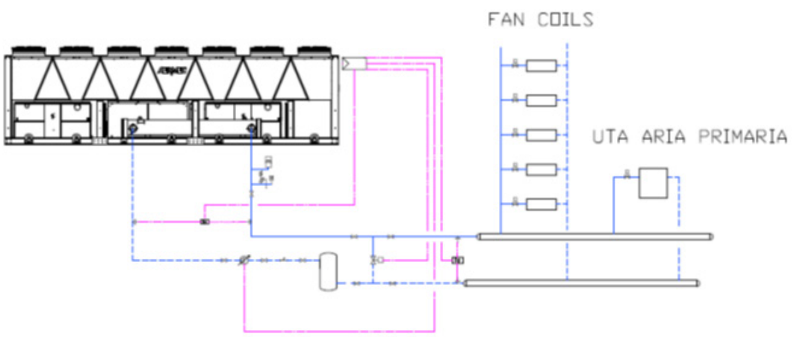
laden water flow	mc/h	123,8
	gpm	546
Pressure drop	kPa	154
	psi	22,33

Solution 2 (VPF system):

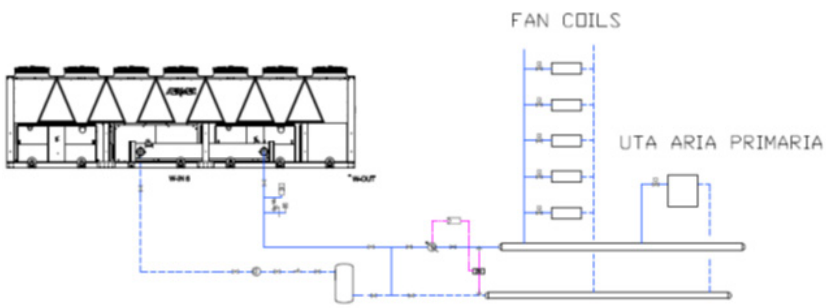
laden water flow	mc/h	123,8
	gpm	546
Pressure drop	kPa	194
	psi	28,13

The air conditioning plant is a primary air type with fan coils. It therefore envisages a processing unit for fresh air (neutral air) and area by area control of temperature conditions through fan coils installed in individual premises served.
Functional block diagrams of the plant are provided for a conventional solution and the VPF solution.

Plant operating diagram - VPF solution

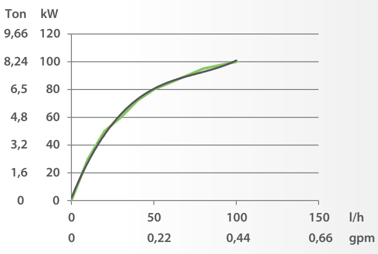


Plant operating diagram with conventional double-loop circuit

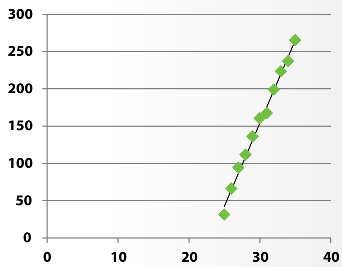


The modulation of the cooling capacity on the UTA battery (with delivery temperature control at a fixed point) is via a two-way modulating valve; adjustment of environment terminals takes place by means of ON-OFF cycles on ventilation with a two-way ON–OFF valve open as requested by the local thermostat.
The water flow to user points remains proportional to effective demand through a constant differential pressure adjustment between the delivery manifold and the return manifold that acts on secondary circuit pump in the conventional solution and on the single pump unit for the VPF solution.

PERFORMANCE CAPACITY / POWER BATTERY MODULAT (UTA)



PERFORMANCE CAPACITY / POWER ON-OFF (FANCOIL)

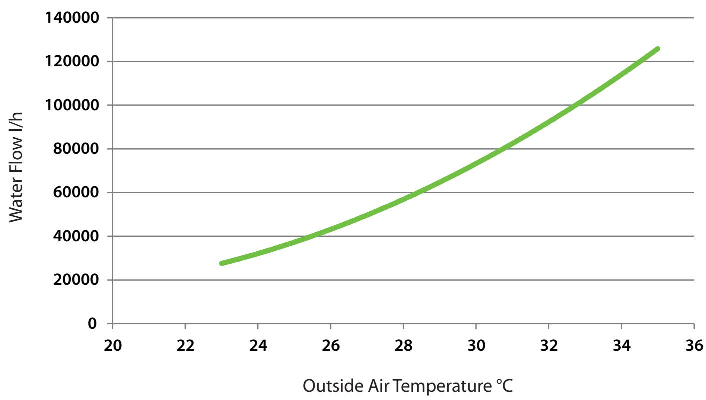


BUILDINGS ENERGY CONSUMPTION

The calculation of expected energy consumption by the chiller referred to summer was made by considering the cooling load associated with ventilation (covered by UTA) and the one associated with transmission, radiation and internal loads (covered by fan coils) in relation to the outside temperature. To calculate the electrical absorption of the air conditioning system in summer, we therefore used the following relationship between outdoor temperature, cooling power required and the water flow sent to terminals (UTA + Fan coils) as shown below:

Utilities Water Flow		
External Temperature °C	**Cooling Capacity kW**	**Water Flow l/h**
35	714	122693
34	662	117413
33	623	106227
32	575	92010
31	519	78497
30	488	72715
29	439	63997
28	390	56654
27	349	50823
26	297	44613
25	237	38102
24	182	31254
23	158	27084

WATER FLOW CIRCUIT SYSTEM



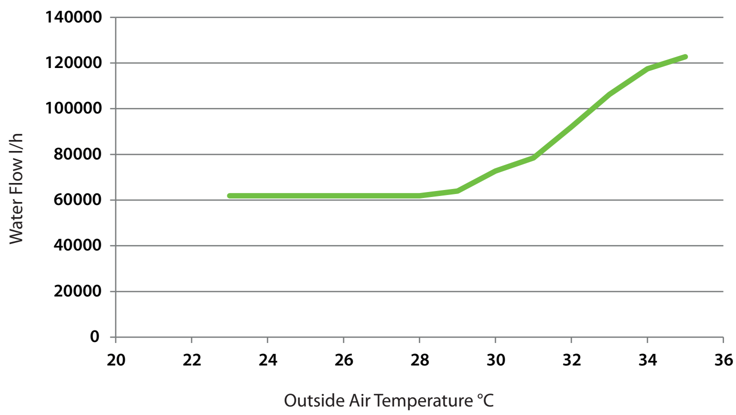
Average water flow rate in the user circuit in relation to outside temperature.

UTA: air handling units of the NCD number used to make the renewal of air required by the presence of people.

FCZ: fan coil used to meet the sensible and latent load in the various offices.

The water flow rate sent to user points and thereby defined refers to circulation on the secondary circuit or, for VPF, to the flow sent to user side manifolds; in the latter case, however, the minimum water flow rate processed by the chiller is equal to 50% of the rated value, so that the relationship between outdoor temperature – load and chiller water flow rate is as shown below:

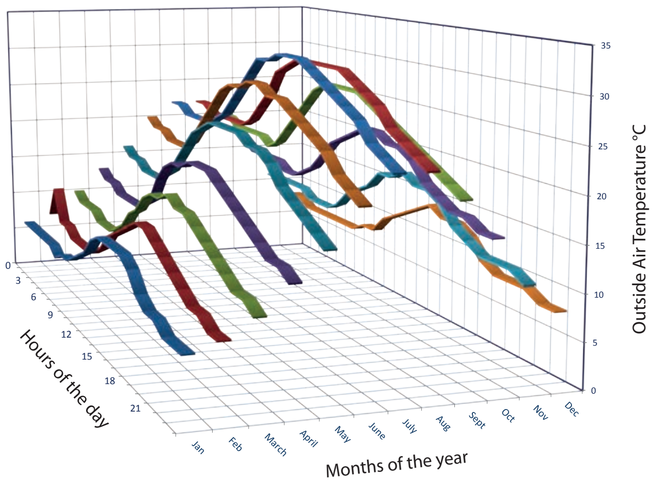
Chiller Water Flow (min 50%)		
External Temperature °C	**Cooling Capacity kW**	**Water Flow l/h**
35	714	122693
34	662	117413
33	623	106227
32	575	92010
31	519	78497
30	488	72715
29	439	63997
28	390	61911
27	349	61911
26	297	61911
25	237	61911
24	182	61911
23	158	61911



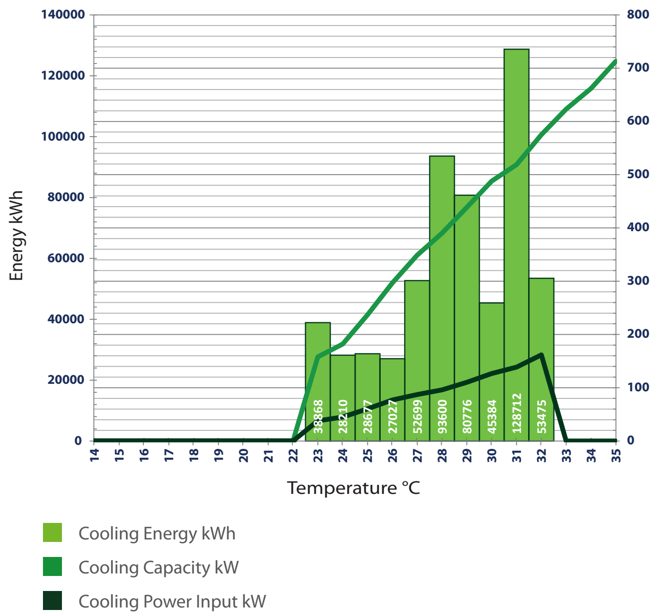
Pump VPF average water flow according to the outside temperature.

The frequency of occurrence of each temperature value was determined throughout the summer, with reference to the location (Rome) and peak times of effective use of the system. For each temperature value, we calculated power absorption in relation to chiller capacity control using the AERMEC ACES program; it must be noted that this value does not consider absorption by pump units which are external to the chiller unit; these values were calculated separately.

Time monthly average Profile of temperature
City: Rome

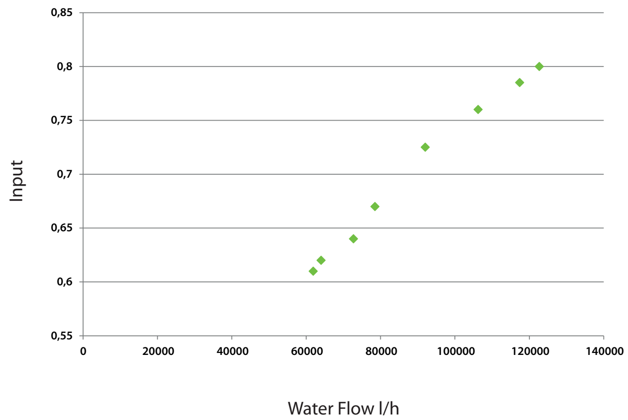


Analyzes chiller performance: NSM3202E



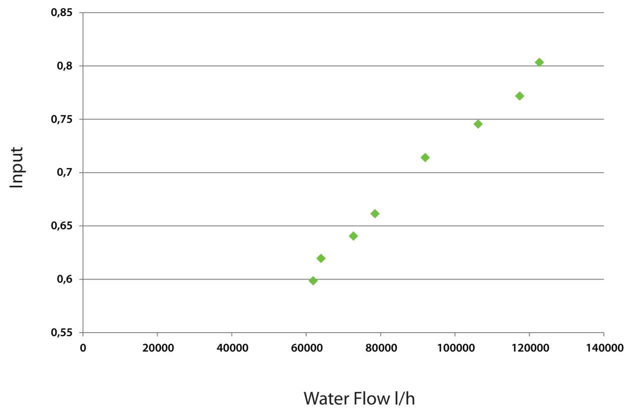
The ACES program is a tool developed by AERMEC to calculate the seasonal consumption and efficiency of chillers (single and parallel chillers alike); it is used for advanced energy analysis which in turn helps optimise the choice of chillers and the management strategy for parallel chillers to achieve maximum system energy efficiency. The same care should be given to calculations of pumping costs since it is here that economic and energy benefits between conventional and VPF solutions emerge. Calculating the effective absorption values for pumping units under various conditions takes flow values into account – head and performance deterioration on varying flow rate in relation to the rated value (for the secondary circuit pump in a conventional system and for the single circuit pump in the VPF system); obviously, the primary circuit pump in conventional plant always works at the rated point.

Secondary Pump Performance
Traditional System



Inverter Pump

Single-Pump Performance
VPF System

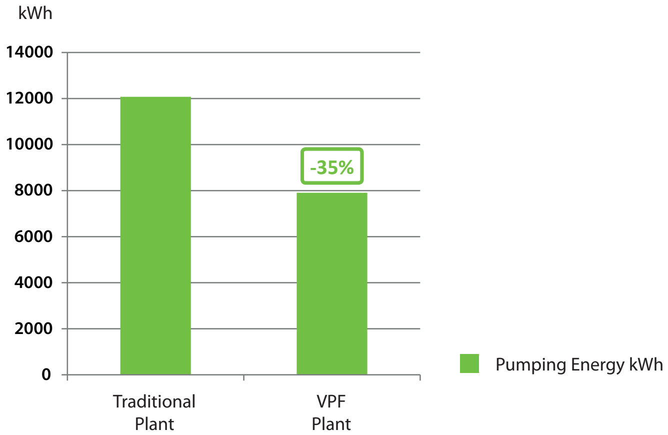


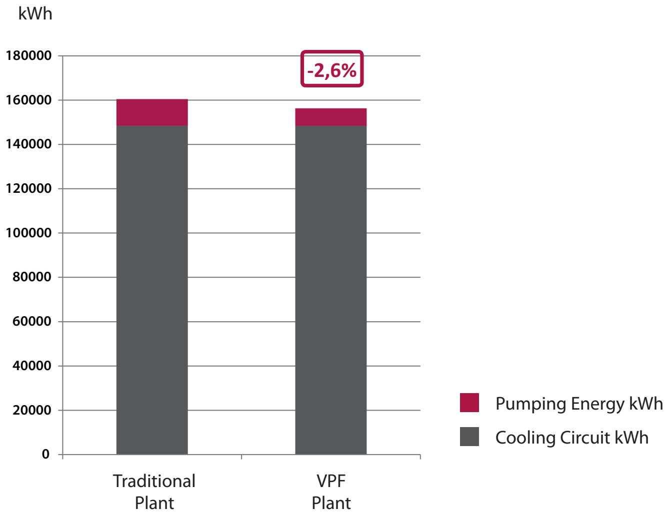
(1) The higher cost of the VPF solution takes following components into consideration: 2 flow measurement devices on the heat exchangers, by-pass with modulating valve, additional controller for the VPF system.
(2) Cost of electricity 0,18 € / kWh
(3) Annual interest rate 4%, inflation rate 2%, 15-year plant lifespan

There follow the calculation results referred only to pumping costs and the overall energy absorption (chillers + pump units) and the outcome of the comparison between the two systems.

			Traditional System	**VPF System**
Absorption Pumps	(kWh)		12070	7907
Absorption Chiller	(kWh)		148335	148335
Total Absorption	(kWh)		160405	156262
Maggiore costo VPF / tradizionale	(€)	(1)	1500	
Greater cost VPF / Traditional	(€/year)	(2)	749	
Pay back period	(years)		2 years	
VAN differential VPF / Traditional	(€)	(3)	7963	

PUMPING COSTS COMPARISON BETWEEN A TRADITIONAL PLANT AND A PLANT VPF





Chapter 3
CONCLUSIONS

Energy savings on pumping are not insignificant between the two cases; in percentage terms of total consumption, they are not particularly high values, as was to be expected considering the fact that the incidence of pumping costs for the primary circuit alone are certainly not the first consumption item. Nonetheless, this is a measurement of relatively low-cost energy efficiency, whereas the higher costs for components in the VPF system (controller, modulating valve on the by-pass, water flow meters on the evaporators) have irrelevant proportions for medium-high power systems. This measurement therefore has fairly short return times (usually around 3 years) and is an investment with a significant positive net effective value over the useful lifespan of the plant, so that it is a convenient choice not only from an environmental point of view.

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