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# MODIFIED WATER LOOP HEAT PUMP SYSTEM FOR A HOSPITAL WITH COMPLEX HVAC SYSTEMS

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Preliminary notes
When heat pumps are used in refurbishment of complex HVAC systems in buildings, high ratio of produced thermal energy and consumed electric energy must be achieved, while operating conditions of existing installations which are usually not renewed have to be maintained. The results of energy efficiency study for hospital complex situated in Rovinj (Croatia) have been presented in the paper. Possible heat pump systems suitable for renewal of the existing thermal energy supply system have been presented and analyzed. New concept has been proposed. Existing boiler room and distribution pipelines, including radiators and hot water boilers in buildings would remain unchanged as the backup system. VRF heat pump systems with water loop as the heat source and sink would be added. Eventual lack or surplus of thermal energy in water loop would be exchanged with sea water or a first stage heat pump. The reconstruction enables achievement of higher comfort level with still acceptable investment in energetic infrastructure. Numerical simulation has resulted in significant improvements in energy consumption compared with the case of systems renewal using the present concept.

**Keywords:** *efficiency, heat pump, renewal, simulation, water loop system*

## Modificirani sustav dizalica topline s vodenim krugom za bolnicu sa složenim termotehničkim sustavima

Prethodno priopćenje
Kod primjene dizalica topline pri obnovi složenih termotehničkih sustava građevina potrebno je osigurati visok omjer proizvedene topline i utrošene pogonske energije uz poštovanje uvjeta rada postojećih instalacija za distribuciju topline koji se često ne obnavljaju. U radu su prikazani rezultati analize provedene energetske studije za kompleks bolnice u Rovinju (Hrvatska). Prikazani su i analizirani mogući sustavi s dizalicama topline prikladni za obnovu postojećeg sustava opskrbe toplinom. Predložena je nova koncepcija sustava. Postojeći sustav kotlovnice, distribucijskih cjevovoda, radijatora i bojlera za potrošnu vodu u zgradama ne bi se obnavljao, već bi poslužio kao rezerva. Sustav bi se nadogradio VRF dizalicama topline s vodenim krugom kao toplinskim izvorom ili ponorom. Eventualni manjak ili višak toplinske energije u cirkulacijskom sustavu s vodom izmijenio bi se s morskom vodom ili s dizalicom topline prvog stupnja. Rekonstrukcijom bi se dosegao značajno viši stupanj komfora uz prihvatljiva ulaganja u energetsku infrastrukturu. Numeričkom simulacijom su utvrđene znatne uštede energije u odnosu na slučaj obnove uz zadržavanje dosadašnjeg koncepta.

**Ključne riječi:** *dizalica topline, obnova, simulacija, sustav s vodenim krugom, učinkovitost*

## 1 Introduction

Energy efficiency promotion and adoption of renewable energy sources are techniques used in modern society to deal with problems of high energy cost, large energy consumption and greenhouse gases emissions.

Heat pump applications are capable to make improvement in all mentioned areas simultaneously.

Directive of the European parliament and council on the promotion of the use of energy from renewable sources [1] recognizes efficient heat pumps with high seasonal performance factor (*SPF*) as systems for utilization of renewable energy. *SPF* is the ratio of produced useful energy and consumed electric energy and should be greater than

$$SPF > \frac{1{,}15}{\eta}, \tag{1}$$

where $\eta$ represents ratio of total electric energy production and primary energy consumption for its production and should be calculated as average value for EU, according to data from Eurostat [2]. This request ensures that useful energy is greater than primary energy consumed for electric energy used by heat pump.

Hospitals are complex facilities, often built and renewed through a long period of time. In most cases their thermal systems comprise water vapour production (heating, sterilization etc.), hot water production at different temperature levels (radiator heating, domestic hot water heating, fan coils, surface embedded heating systems etc.) and chilled water production (air – conditioning and cooling purposes).

Application of heat pumps in such systems is a challenge for a designer, as it can be done in many different ways. Efficiency of such systems depends not only on the choice of equipment, but on system behaviour in dynamic changes during heating and cooling due to boundary conditions imposed by the state of the heat distribution subsystem and surroundings. Medium temperature heat pump systems suitable for implementation into existing heating systems are cascade heat pumps with direct or indirect heat exchange between cascade stages, and a two stage heat pump modified for efficient operation in such systems. Water loop heat pump systems are also suitable for refurbishment.

The case study presented in this paper is the hospital complex in Rovinj (Croatia). The present energy system in the hospital is no longer acceptable due to high energy consumption and inadequate performance. It has been thoroughly analyzed in a detailed energy study financed by UNDP [3]. The important suggestion to the owner, given in the study [3] was adequate insulation of all buildings in order to make possible cost efficient installation of new energy system based on renewable energy source.

A new water loop heat pump system (WLHP) has been proposed as a feasible solution. It could be easily implemented besides the existing system which can remain unchanged and serve as the backup system. The heat source shall be the sea water. Heat pumps with variable refrigerant flow (VRF) have been chosen, due to the demand for as lowest as possible intervention into the existing heat distribution systems. Such heat pumps will ensure optimal comfort (including summer cooling which has not been available yet) and lowest energy consumption. High efficiency of WLHP system and chosen VRF heat pumps results in the payback period (evaluated only for heating) shorter than the system lifetime.

## 2
## Heat pumps within hydronic heat distribution system

Heat pump is a device that transfers thermal energy from a source to a sink that is at a higher temperature level than the source. Thus, heat pumps move thermal energy in a direction which is opposite to the direction of spontaneous heat flow. The heat pump "mixes" energy flow with low energy taken from the heat source with energy flow of high energy from electric network in order to accomplish the desired transfer of thermal energy from source to sink. A simplified heat pump functional diagram is shown in Fig. 1 where refrigerant thermal states within the system are labelled with numbers 1 to 4 and correspond to those presented in Fig. 2.

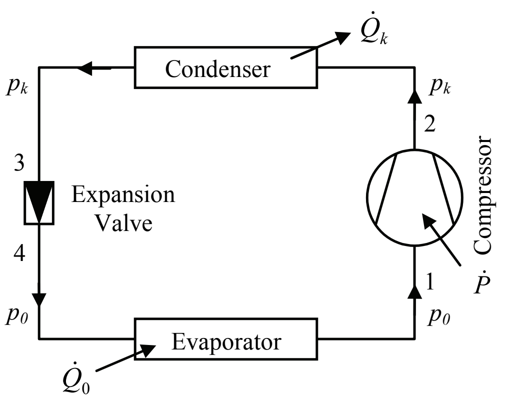


**Figure 1** Heat pump functional scheme

Refrigerant passes through change of phases which are presented on *p-h* and *T-s* diagrams in Fig. 2. Using those diagrams for a certain refrigerant one can determine enthalpies, specific work and power. Specific work for compression $w$, specific heating flux $q_1$, specific cooling flux $q_2$, coefficient of performance for heating $COP_1$ and coefficient of performance for cooling $COP_2$ can be calculated as follows:

$$w = h_2 - h_1,\ \text{kJ/kg} \tag{2}$$

$$q_\text{k} = h_2 - h_3,\text{kJ/kg} \tag{3}$$

$$q_0 = h_1 - h_4 = h_1 - h_3,\ \text{kJ/kg} \tag{4}$$

$$COP_1 = \frac{q_\text{k}}{w} = \frac{h_2 - h_3}{h_2 - h_1}, \tag{5}$$

$$COP_2 = \frac{q_0}{w} = \frac{h_1 - h_3}{h_2 - h_1}. \tag{6}$$

Refrigerant mass flow can be determined from the ratio of heating flux and specific heating flux:

$$\dot{M} = \frac{\dot{Q}_\text{k}}{q_\text{k}} = \frac{\dot{Q}_0}{q_0},\ \text{kg/s}. \tag{7}$$

Required power for isentropic compression can be calculated from refrigerant mass flow and specific work for compression:

$$\dot{P}_\text{is} = \dot{M} \cdot w,\ \text{kW}. \tag{8}$$

Effective power is calculated as

$$\dot{P}_\text{e} = \dot{P}_\text{is} \cdot \eta_\text{is},\ \text{kW}, \tag{9}$$

where isentropic efficiency $\eta_\text{is}$ values usually vary between 0,8 to 0,9 for a single stage compression. Expressions (5) and (6) describe theoretical *COP*, while real *COP* should be calculated taking into consideration effective power from (9) and system partial load as well.

Such a heat pump, with a single stage compression can, depending on the chosen refrigerant and compressor type, achieve condenser outlet water temperatures up to 55 °C, which is often not high enough for heat supply in existing heat distribution systems with radiators or heaters in air handling units designed for boiler heated systems. Nevertheless, it is possible to achieve higher condenser outlet water temperatures than 55 °C, but in that case refrigerant temperature at the end of compression phase can reach over 140 °C, what is upper temperature limit before compressor lubricants can lose their guaranteed properties.

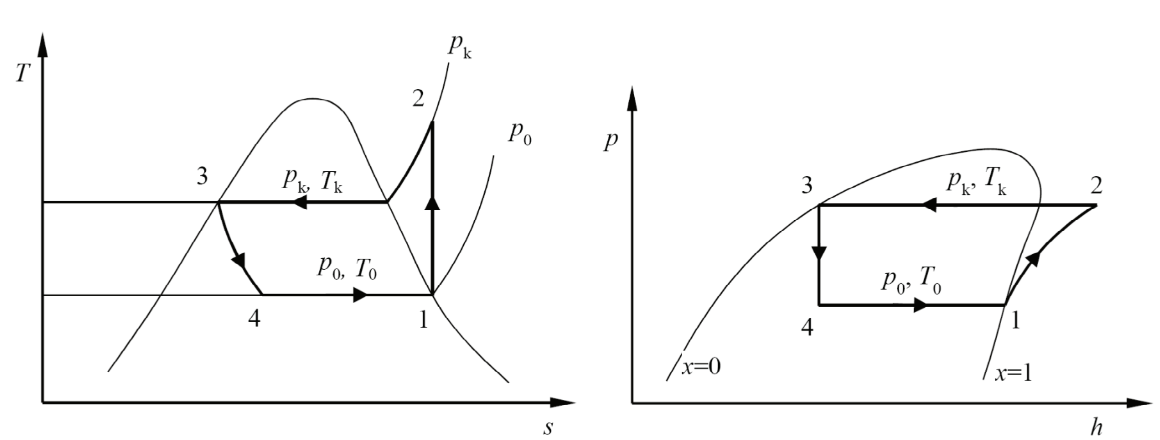


**Figure 2** *T-s* and *p-h* diagrams for a heat pump process with single stage compression and throttling

A comparison of compression ratios, coefficient of performance for heating and temperatures at the end of compression phase for usual refrigerants R134A (large capacity chillers) and R410A (low capacity chillers, split units, VRF units) is performed. Values of $COP_1$ are useful for evaluation and comparison of different processes, as well as evaluation of influence of chosen refrigerant on process characteristics and refrigerant performance within the process. This analysis is performed for condensation temperatures varying from 50 °C to 80 °C and evaporation temperatures in range from –20 °C to +20 °C. Higher COP values are achievable with R410A than with R134A for processes compared at the same condensing and evaporation temperatures. It is important to note that $COP_1$ is not a fixed value and it changes in time, depending on heat source and heat sink temperatures. Integration in time of variable values of $COP_1$ results in average *SPF* while integration of $COP_2$ for the entire system results in *SEER* (seasonal energy efficiency ratio often used for cooling).

Increase of the difference between evaporation temperature $T_2$ and condensation temperature $T_1$ results in increased pressure difference. That causes increased work for compression, while coefficient of performance for heating is descending. Higher compression ratio has also the consequence of higher temperature $T$ at the compressor outlet. Higher temperatures of R410A at the compressor outlet are not a problem for a single stage process, as the lubricant temperature limit is not exceeded, but higher pressures in the entire system limit the application of R410A to chillers of lower capacity (approximately up to 350 kW) while R134A is used for chillers with higher capacities.

When higher temperatures are necessary in a hydronic heat distribution system, cascade (a) or two – stage (b) heat pumps presented in Fig. 5 can be used. Heat pumps can also be arranged as a kind of cascade heat pump (Fig. 5c). In that case water loop is used for thermal energy transfer between evaporator of higher and condenser of lower heat pump stage. The choice of refrigerant makes possible efficient and technically acceptable operation of each heat pump in such a cascade.

All heat pump systems presented in Fig. 5 can reach condensing temperatures of 80 ÷ 90 °C without the risk of lubrication oil overheating and thus can be used for DHW heating and other medium temperature heating applications.

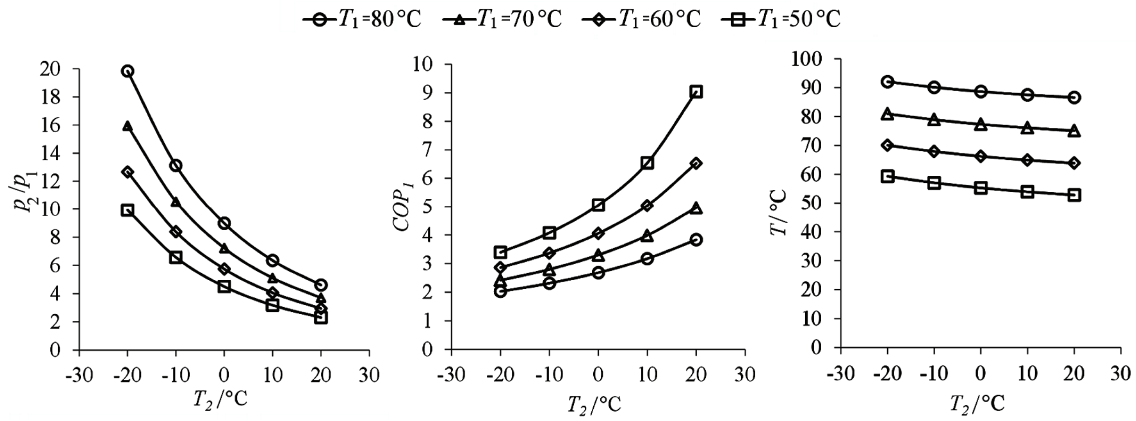


**Figure 3** Compression ratios, coefficient of performance for heating $COP_1$ and temperatures at the end of compression phase for different temperatures of evaporation and condensation - single stage process (refrigerant R134A)

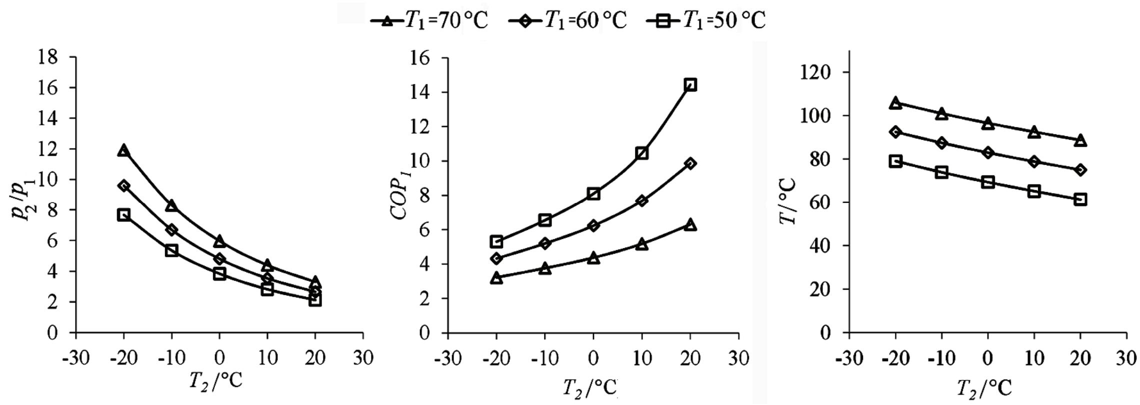


**Figure 4** Compression ratios, coefficient of performance for heating $COP_1$ and temperatures at the end of compression phase for different temperatures of evaporation and condensation – single stage process (refrigerant R410A)

Two stage systems can achieve high temperature with lower compressor outlet temperature but also lower $COP_1$. Examples of features for such a process are given in Figs. 6 and 7 for the heat pump presented in Fig. 5b. The choice of ammonia R717, instead of R410A has been done because ammonia is usual refrigerant for such systems. Another reason is that refrigerant R410A is not suitable for medium temperature applications.

Heat pump in Fig. 5b can be used for energy efficient cooling which does not demand high condensing temperatures in the case when the flash tank 6 is replaced with a condenser of sufficient storage volume and special design. One possible application is presented in Fig. 8.

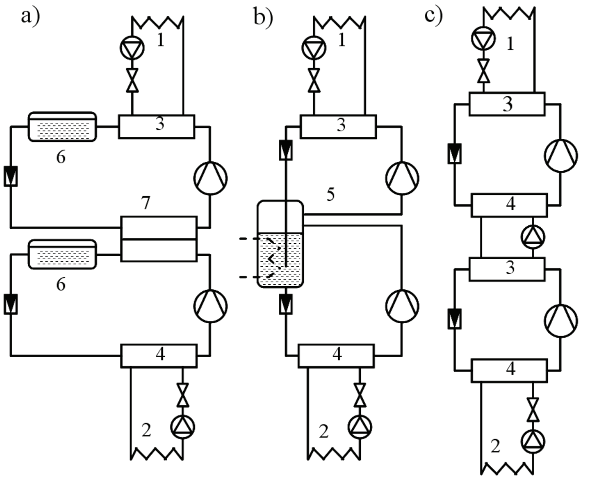


**Figure 5** Medium temperature heat pumps
(1 - Heat sink, 2 - Heat source, 3 - Condenser, 4 - Evaporator, 5 - Flash tank, 6 - Receiver, 7 - Condenser/evaporator)

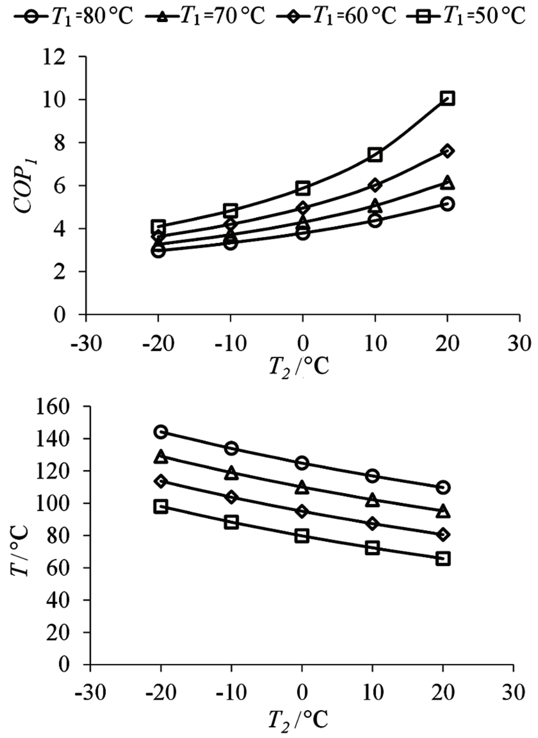


**Figure 6** Coefficient of heating $COP_1$ and temperatures at the end of compression phase for different temperatures of evaporation and condensation – two stage process (refrigerant R717)

During the operation in the cooling mode of the system presented in Fig. 8, the high stage compressor is shut off. The heat of condensation is removed from the first stage condenser (flash tank) and utilized either for domestic hot water heating in the tank DHW-1 (up to approximately 50 °C), or rejected to the sea. When higher temperatures of domestic hot water are necessary, high stage compressor starts up, and heating of DHW-2 tank is possible (up to approximately 70 °C, which prevents appearance of legionella). In winter season such a heat pump can replace the conventional boiler, and that is particularly suitable in refurbishment of buildings with complex heat distribution systems which are not subject to changes during the refurbishment. Another advantage of the system presented in Fig. 8 is that at higher outdoor temperatures $\vartheta_a$ the necessary supply water temperature $\vartheta_D$ decreases (Fig. 9) thus making possible heating with only the first stage of the heat pump which operates with higher $COP_1$ than in the case with both stages on. When the consumption of energy is analyzed it is obvious from the degree – day curve on the bottom of Fig. 9 that major part of necessary thermal energy for heating the buildings will be produced in the first stage mode of operation with high $COP_1$ ($Q_{HP,I}$) and only minor part in operation of both stages with lower $COP_1$ ($Q_{HP,II}$).

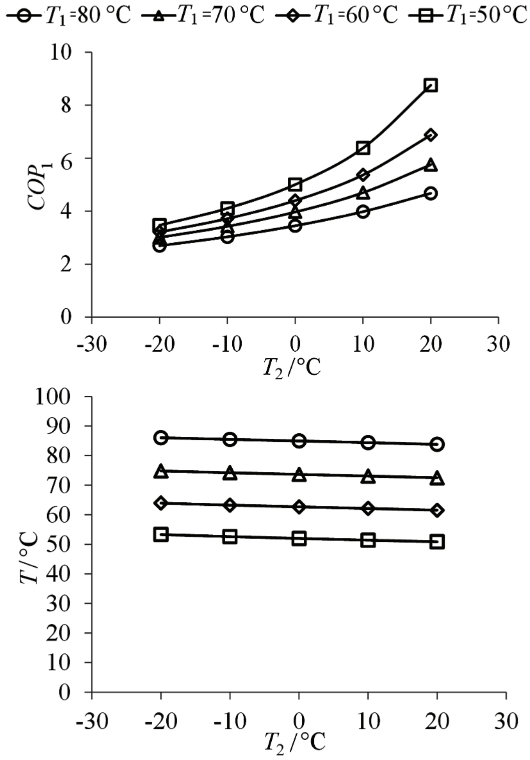


**Figure 7** Coefficient of heating $COP_1$ and temperatures at the end of compression phase for different temperatures of evaporation and condensation – two stage process (refrigerant R134A)

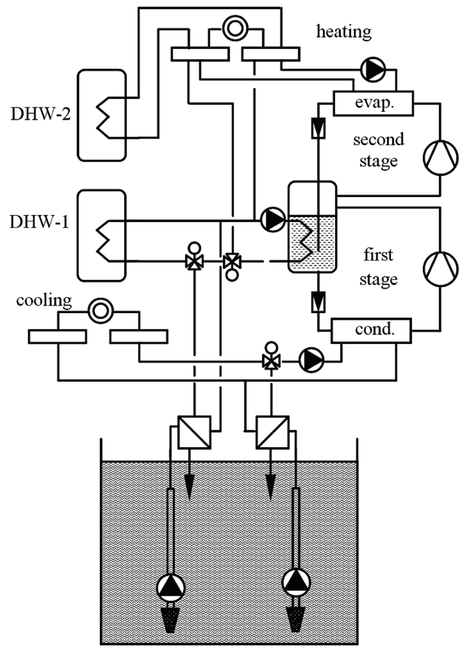


**Figure 8** System with two stage heat pump

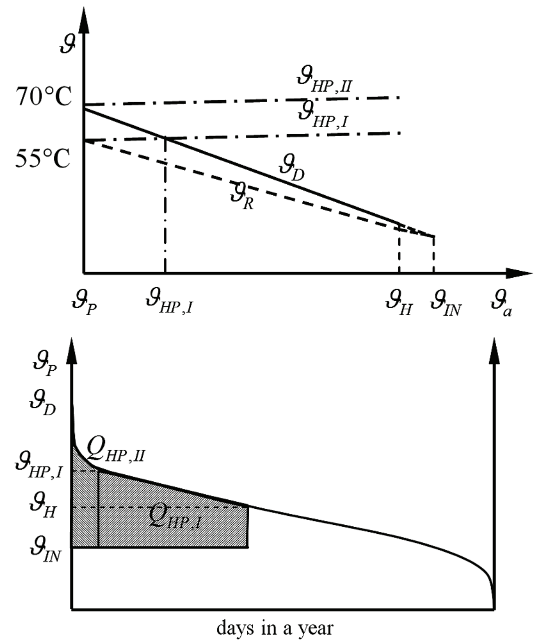


**Figure 9** Temperatures and heat consumption during heating with heat pump stages HP-I and HP-II

## 3 Water loop heat pump system

The water loop heat pump (WLHP) system consists of closed piping system with circulation of water which is used as heat source or heat sink for heat pumps. Those heat pumps can be of different types, such as domestic hot water (DHW) heaters, water to water heat pumps, water source VRF systems etc. All heat pumps can be reversible in refrigerant circuit thus making possible heating or cooling. In that case some of heat pumps provide heating and other heat pumps provide cooling. Thermal energy is interchanged and recuperated within water loop and there is no need for additional heating or cooling of such system. Typical WLHP system as described in [4] is presented in Fig. 10.

If heat pumps mainly provide heating, water loop temperature decreases and additional heat source must be included (e.g. conventional boiler). On the other hand, intensive cooling of buildings will increase the water loop temperature, what can be utilized by heat pumps intended for DHW. In case that water loop temperature increases, further thermal energy can be dissipated in environment (e.g. cooling towers).

Such systems are built for large buildings or groups of buildings. WLHP systems are typically installed in edifices with dislocated core and perimeter zones or commercial buildings, sometimes with cold stores.

Besides high efficiency and high level of heat recuperation, one of major advantages of WLHP systems is the possibility of gradual construction. The main water loop pipelines are built at the beginning of the system construction, whilst heat pumps could be attached according to consumer's needs till the specified system capacity is covered.

The efficiency of WLHP system could be improved by the use of variable refrigerant flow (VRF) heat pump units which are connected to a water loop. A VRF system is a refrigerant system that varies the refrigerant flow rate with the help of the variable speed compressor and the electronic expansion valves to match the space cooling load in order to maintain the zone air temperature at the set value. VRF system consists of "outdoor" unit with several compressors and internal units customized for relevant spaces. Such systems are capable to achieve high seasonal heating and cooling factors and relatively high investment is feasible with low energy consumption, which is mainly due to the refrigerant mass flow control as explained in [5].

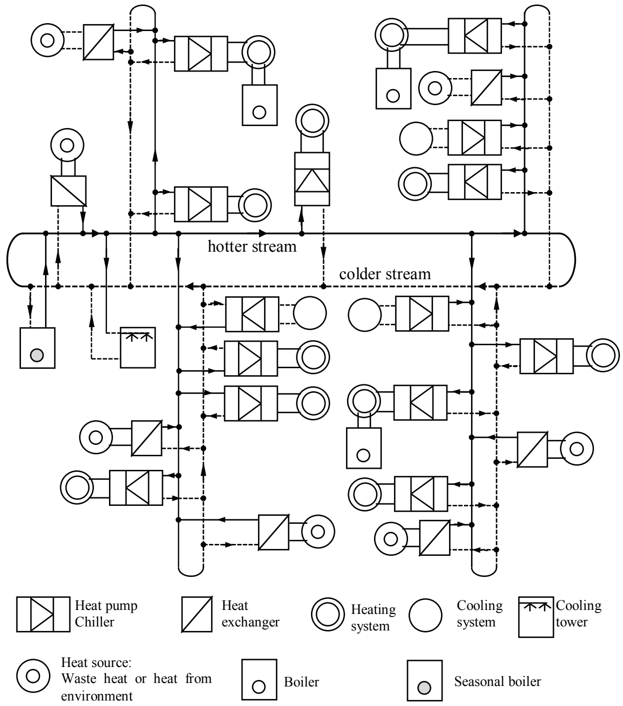


**Figure 10** Example of a WLHP system [4]

VRF systems use narrow pipelines for distribution of heating or cooling energy by refrigerants (usually R410A) which have significantly smaller diameter in comparison with existing pipes in heating network. That makes them suitable for setup and fitting into existing buildings. Central VRF units chosen in the presented case use water loop thermal energy as a heat source during winter season or as a heat sink during summer season, but air – source units are common as well.

## 4 Analyzed building complex and proposed reconstruction

Analyzed building complex is the orthopedic and rehabilitation hospital in Rovinj. The hospital consists of 17 separate buildings which are spread on an area of approximately 200 000 m$^2$. Total area of all 17 buildings is around 10 100 m$^2$. The hospital accommodates approximately 250 to 350 patients and 130 employees.

Thermal energy is supplied from the boiler room with two hot water fuel fired boilers (total power of 3 480 kW) and two steam boilers (total power of 1 775 kW). Installed heating power of the systems is 2 100 kW, where 660 kW is intended for DHW and the rest of 1 440 kW is used for buildings heating. Steam boilers provide steam for laundry (373 kW) and for kitchen (347 kW). Water steam is also used for the heating of these two buildings including another building for accommodation (160 kW).

Several recommendations for building and thermal system improvements have been given in [3] with intention to decrease current energy consumption.
It has been suggested to perform recommendations which deal with building improvement and insulation in order to reach lower thermal plant capacity and its price accordingly.

Modification of an ordinary WLHP system proposed in this article (presented in Fig. 11) comprises the use of the sea water as a primary heat source or heat sink instead of conventional boiler and cooling tower presented in Fig. 10. Heat pumps are built in the system in a "cascade" arrangement similar to one presented in Fig. 5 c), using central heat pump for additional heating when the sea water cannot supply sufficient heat to the loop. Thus, savings in consumption of fossil fuels and electricity could be achieved, especially when sea water temperature is adequate. VRF heat pumps are attached to the water loop.

Advantages in the use of sea water as the heat source are observed in achievement of better energy efficiency due to higher sea temperature during the winter and lower sea temperature during the summer compared to the air as the heat source. Better reliability and smaller environmental impact are also the consequence of utilization of the sea water as the heat source.

A minimal temperature of 10 °C of the water loop is maintained during the winter season. Heating of water loop is provided by heat exchanger with sea water and by heat pump HP-1 via HX-HP heat exchanger, as shown in Fig. 11. Return water in the water loop passes through HX-SEA-LOOP. In the winter operation case, when the water loop temperature after the heat exchanger is below 10 °C, the water loop is additionally heated by HP-1.

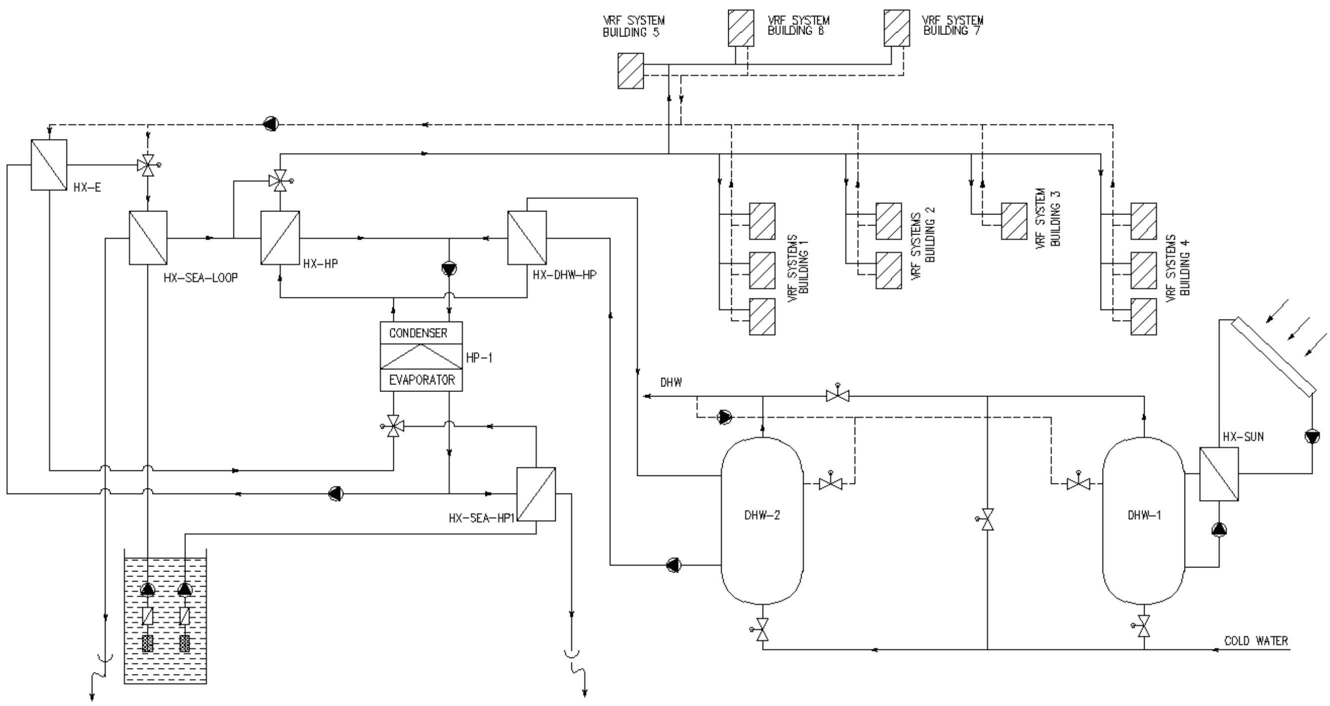


**Figure 11** Proposed water loop heat pump system

Maximal design temperature of the water loop during summer season is 30 °C. VRF units attached to WLHP system release condenser heat into water and additional cooling of the water loop is needed. Maximal sea water temperature never exceeds 25 °C and the water loop cooling can be provided only by the sea water exchanger HX-SEA.

There is also a possibility of DHW heating by the heat pump HP-1. In that case water loop is cooled in HX-E and the DHW is heated over HX-DHW-HP. Thus the condenser heat released from VRF units is used as an excellent heat source for the heat pump HP-1. Due to relatively high temperatures at the evaporator, high heat pump efficiency can be achieved.

A field of solar thermal collectors makes part of the proposed system. The collector field area has been optimized for simultaneous operation with a heat pump. The system has to be carefully designed, providing low temperature in DHW-1 tank, which makes possible operation of solar collectors with high thermal efficiency.

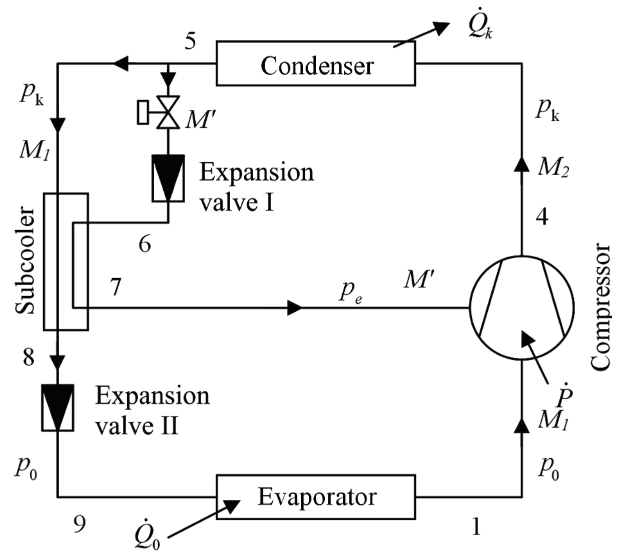


**Figure 12** Schematic diagram of economizer process

In proposed WLHP system a heat pump operating by the economizer process can be used. Such process ensures noticeable savings in work for compression and provides higher temperatures in condenser without exceeding allowed temperatures at compressor outlet.

Schematic diagram of economizer process is shown in Fig. 12, while changes of refrigerants state are shown in *p-h* and *T-s* diagrams in Fig. 13.

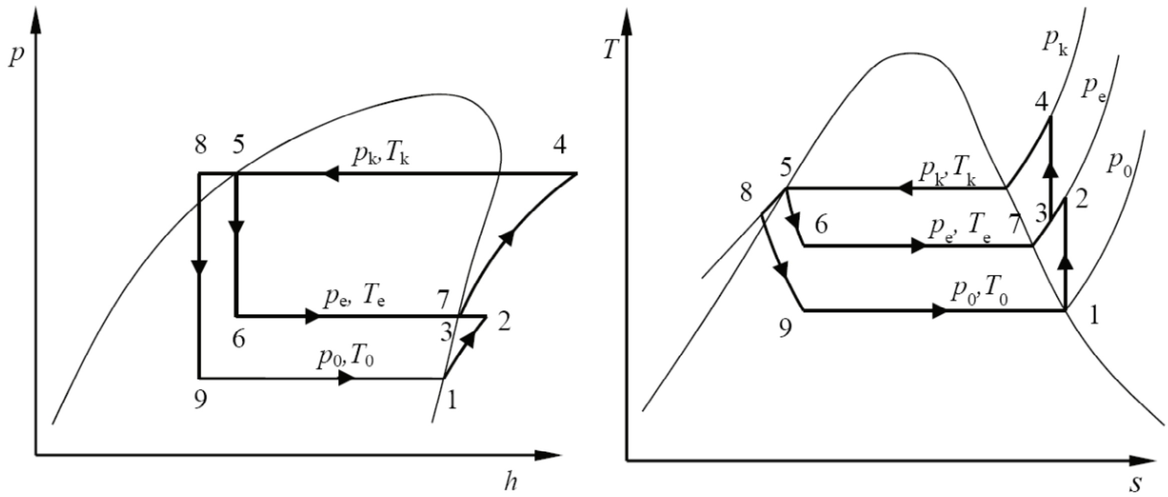


**Figure 13** *p-h* and *T-s* diagrams of economizer process

Heat exchanged in condenser is defined by:

$$\dot{Q}_k = \dot{M}_2(h_4 - h_5), \text{ kW}. \tag{10}$$

Mass flow of refrigerant through the condenser is a sum of mass flows $\dot{M}_1$ and $\dot{M}'$

$$\dot{M}_2 = \dot{M}_1 + \dot{M}' = \frac{Q_k}{q_k} = \frac{Q_k}{h_4 - h_5}, \text{ kg/s}. \tag{11}$$

Ratio of mass flows $\dot{M}_1$ and $\dot{M}'$ can be calculated from heat balance in evaporator-subcooler:

$$\frac{\dot{M}'}{\dot{M}_1} = \frac{h_7 - h_6}{h_8 - h_5}. \tag{12}$$

After vapour injection, saturated vapour at state 7 and superheated vapour at state 2 are mixed in compressor and that results with vapour at state 3 which is compressed till pressure $p_k$ (state 4). Hence, power for isentropic compression is:

$$\dot{P}_{is} = \dot{M}_1(h_2 - h_1) + \dot{M}_2(h_4 - h_3), \text{ kW}. \tag{13}$$

Coefficients of performance for heating and cooling are the following:

$$COP_1 = \frac{\dot{Q}_k}{\dot{P}_{is}}, \tag{14}$$

$$COP_2 = \frac{\dot{Q}_0}{\dot{P}_{is}}. \tag{15}$$

A comparison of compression ratios, coefficient of performance for heating and temperatures at the end of compression phase of the above described process for refrigerants R134A and R410A is performed. This analysis is performed for condensation temperatures from 50 °C to 80 °C and for evaporation temperatures in the range from –30 °C to +30 °C.

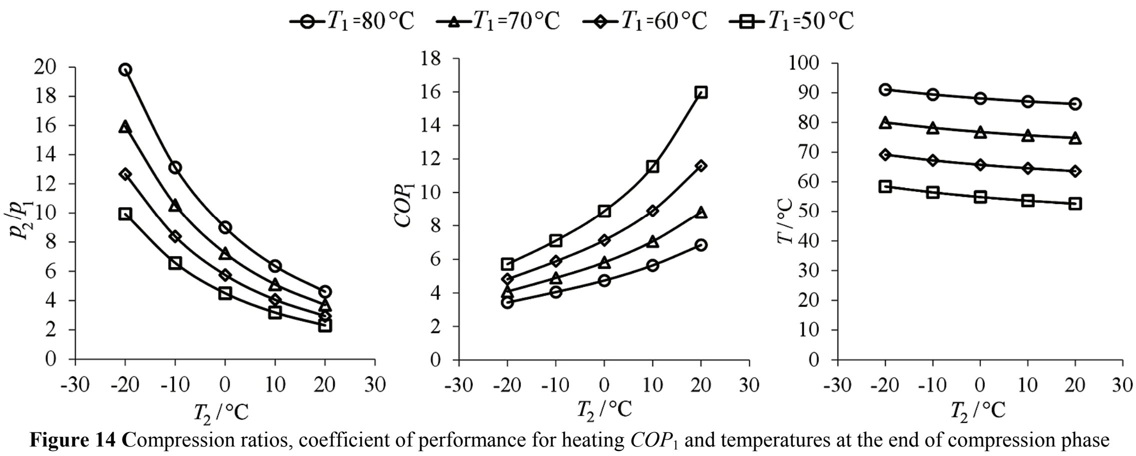


**Figure 14** Compression ratios, coefficient of performance for heating $COP_1$ and temperatures at the end of compression phase for different temperatures of evaporation and condensation - economizer process (refrigerant R134A)

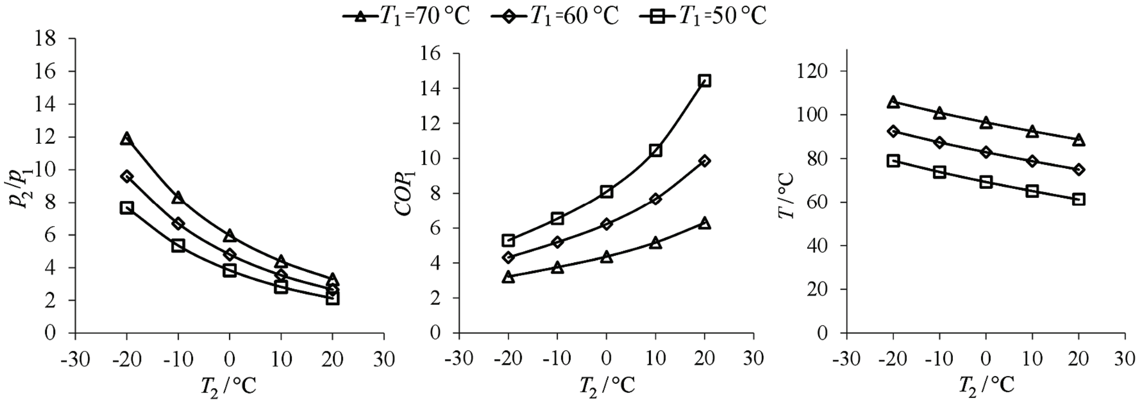


**Figure 15** Compression ratios, coefficient of performance for heating $COP_1$ and temperatures at the end of compression phase for different temperatures of evaporation and condensation - economizer process (refrigerant R410A)

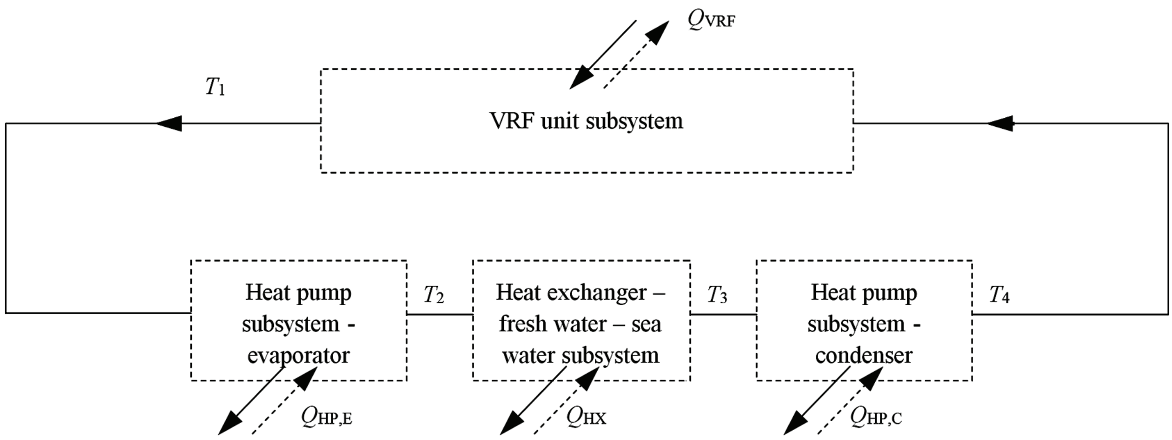


**Figure 16** Subsystems in unsteady simulation of proposed WLHP system

Heat pump operating by economizer cycle can achieve high water outlet temperatures (up to 60 °C) with still high coefficient of performance ($COP_1$).

## 5 Analysis of WLHP system dynamic behaviour

To provide precise results for energy consumption of VRF units and entire WLHP system a detailed dynamic analysis has been performed. Parameters like water temperature in specific parts of water loop, inlet and outlet temperatures from heat exchangers, exchanged thermal energy and consumption of electric energy were calculated with time step of one minute for the whole reference year. Input data for this dynamic simulation of WLHP system are air temperatures and solar radiation for considered location, sea water temperature, heating or cooling demand calculated on one hour basis for each building and properties of main system components. Such dynamic system model is suitable for detailed calculation of energy consumption and its optimization.

In order to perform the dynamic calculation of system energy flows, whole system has been divided into main control volumes which represent main system components where significant energy conversions appear. Following that presumption, the system is divided in four control volumes or subsystems: VRF unit subsystem, fresh water – sea water heat exchanger subsystem, water loop subsystem and heat pump subsystem, as shown in Fig. 16.

Calculation of each subsystem temperature includes mass and specific heat of media for thermal energy transfer (water) as well as mass and specific heat of pipelines and fittings. With this approach a more realistic temperature change can be obtained taking into account energy and time for heating or cooling of entire system.

Temperature of each control volume is calculated on the basis of temperature from previous time step including inlet and outlet temperature of the water which flows through control volume with predefined flow rate. For example, VRF unit subsystem temperature is calculated by the following expression:

$$T_1 = \frac{T_4 \cdot \dot{G} \cdot c_\mathrm{w} \cdot \Delta t + \dot{Q}_\mathrm{HP,C} \cdot \Delta t + T_1^0 \cdot (\dot{G} \cdot c_\mathrm{w} + \dot{G}_\mathrm{s} \cdot c_\mathrm{s})}{(\dot{G} \cdot c_\mathrm{w} + \dot{G}_\mathrm{s} \cdot c_\mathrm{s}) + \dot{G} \cdot c_\mathrm{w} \cdot \Delta t}, \quad (16)$$

where:
$T_1$ – VRF unit subsystem outlet temperature, °C
$T_1^0$ – temperature on heat pump condenser outlet from previous time step, °C
$T_4$ – temperature on the VRF unit subsystem inlet, °C
$\dot{G}$ – water mass flow, kg/s
$G_\mathrm{w}$ – mass of water in subsystem installation, kg
$G_\mathrm{s}$ – mass of steel in subsystem installation, kg
$c_\mathrm{w}$ – specific heat of water, kJ/kg K
$c_\mathrm{s}$ – specific heat of steel, kJ/kg K
$\dot{Q}_\mathrm{HP,C}$ – exchanged heat flux in heat pump condenser, W
$\Delta t$ – calculation time step, s.

Calculation of temperature in all subsystems has been performed using the above described procedure, except for the HX-SEA-LOOP heat exchanger subsystem. The heat exchanger subsystem includes input parameters which depend on the characteristic of the chosen heat exchanger (heat exchanger area, both streams flow rates) and the sea water temperature. According to these data and the water loop inlet temperature (Fig. 17), the exchanged heat and water loop temperature on the heat exchanger outlet are calculated.

Counter flow heat exchanger can be described by the following equation [6]:

$$\frac{T_{\mathrm{m,i}}-T_{\mathrm{m,o}}}{T_{\mathrm{m,i}}-T_2}=\frac{1-\mathrm{e}^{-\left(1-\frac{W_1}{W_2}\right)\frac{kA}{W_1}}}{1-\frac{W_1}{W_2}\mathrm{e}^{-\left(1-\frac{W_1}{W_2}\right)\frac{kA}{W_1}}}=\Phi_{\mathrm{HX}}, \tag{17}$$

which can be written also as:

$$\frac{T_3-T_2}{T_{\mathrm{m,i}}-T_2}=\frac{W_1}{W_2}\Phi_{\mathrm{HX}}. \tag{18}$$

From the last expression and with the known values of $W_1$, $W_2$, $T_2$ and $T_{\mathrm{m,i}}$, outlet temperature of the subsystem can be calculated by the following expression:

$$T_3=T_2+\frac{W_1}{W_2}\Phi_{\mathrm{HX}}\cdot(T_{\mathrm{m,i}}-T_2). \tag{19}$$

Considering $W_1 = W_2$, follows:

$$\Phi_{\mathrm{HX}}=\frac{1}{1+\frac{W_1}{kA}}. \tag{20}$$

Finally, heat flux exchanged in heat exchanger is:

$$\dot{Q}_{\mathrm{HX}}=W_2\cdot(T_3-T_2)=W_1\cdot(T_{\mathrm{m,i}}-T_{\mathrm{m,o}}). \tag{21}$$

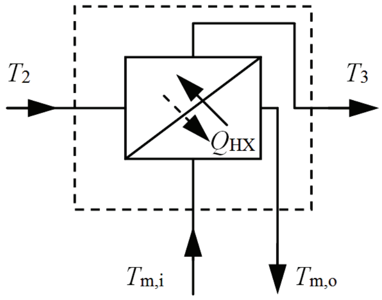


**Figure 17** Schematic diagram of the fresh water – sea water heat exchanger subsystem

After the calculation for the whole referent year in time steps of one minute is performed, temperatures for all subsystems are found. For illustration, Figs. 18 and 19 show temperature change and exchanged thermal energy in all subsystems for several typical days in winter and summer season. Monthly sums of exchanged thermal energies for some subsystems are shown in Fig. 20. It is obvious that almost all the heat rejected from VRF units during the summer season can be used as the heat source for the heat pump HP-1 which serves for heating up the domestic hot water. In winter season the majority of necessary heat for VRF heat pumps' operation is extracted from the sea, due to the low temperature of the water loop which has been chosen to be at least 10 °C and it is higher only in cases when sea water temperatures and the design characteristic of the heat exchanger HX-SEA-LOOP enable higher values. Energy balances which present shares of the used electric energy and renewable energy from the sea are presented in Fig. 21. In entire energy consumption for DHW heating, electric energy for heat pump operation presents only 14 %, while in the energy balance for buildings' heating, electric energy share is 22 % in which consumption of HP-1 for additional heating of the loop makes 3 % and electric energy consumption of VRF systems makes 19 %.

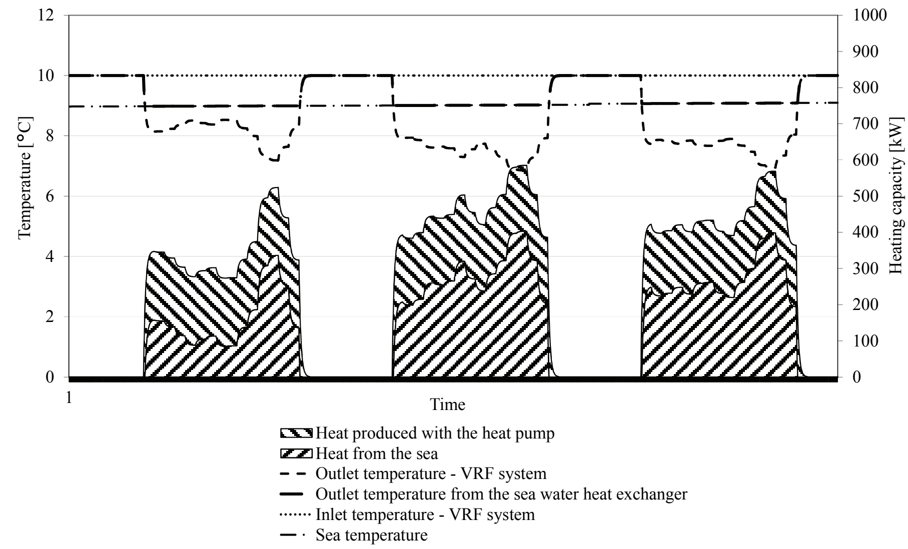


**Figure 18** Temperature change and exchanged heat in subsystems during several typical days in winter season

The presented analysis did not deal with determination of optimal lowest temperature level of the loop. With increased lowest loop temperature during the winter the electric energy consumption of VRF systems would decrease, but the share of renewable sea water energy would also decrease thus forcing HP-1 to operate longer. The decision about the lowest temperature of the loop has been made based on the author's experience in design of similar heat pump systems using the sea water as the heat source.

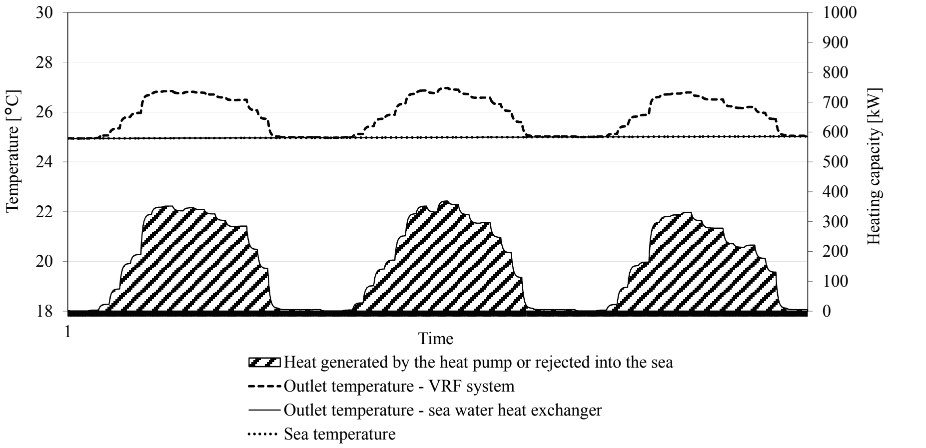


**Figure 19** Temperature change and exchanged heat in subsystems during several typical days in summer season

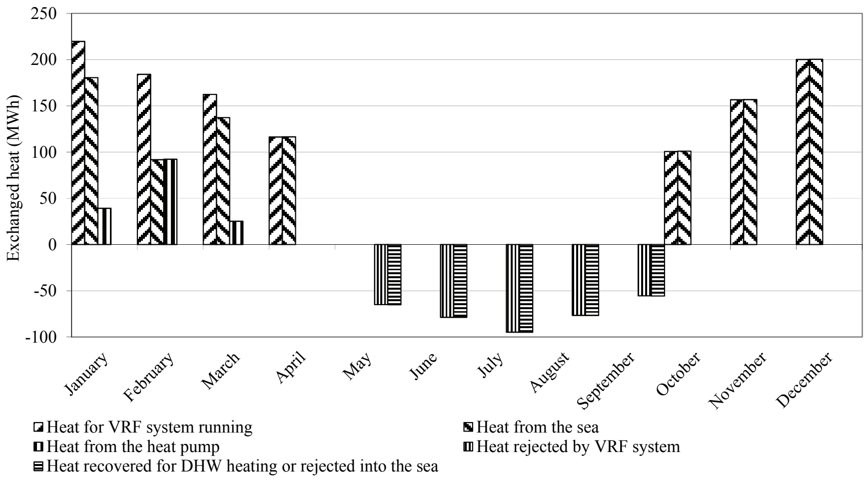


**Figure 20** Monthly exchanged thermal energy in subsystems

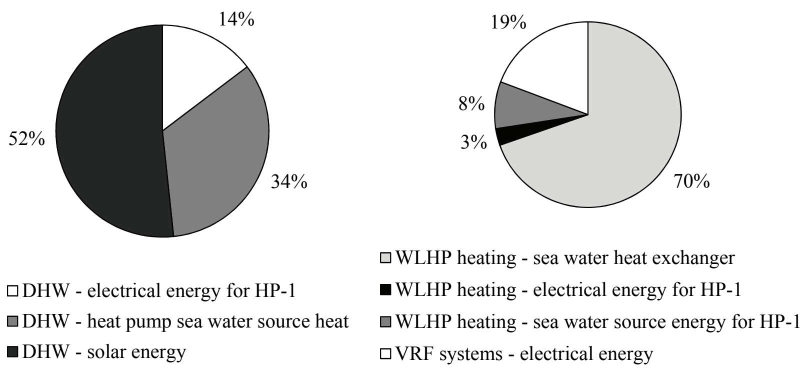


**Figure 21** Total heating energy balance

## 6 Conclusion

The paper presents heat pump systems which can replace conventional boilers, and which are suitable for efficient production of thermal energy for refurbished buildings with complex HVAC systems in the case when heat distribution systems are not subject to changes. Several medium temperature heat pump systems suitable for implementation into the existing hydronic heat distribution system have been presented. Those are cascade heat pumps with direct or indirect heat exchange between cascade stages, and modified two stage heat pump. Water loop heat pump systems have also been presented as suitable systems for refurbishment.

The case study of WLHP system application for a hospital on the Adriatic Sea has been presented. The main reason for the choice of a water loop heat pump system was to avoid the high cost of the total existing system replacement in a short period of time. Refurbishment is intended to be done gradually, without expensive changes in existing heat distribution systems. Numerical all year round simulation of entire heat supply system behaviour resulted in energy consumption data which represent good basis for decision about feasible implementation of such a system. Costs have been evaluated as well in [3], but have not been presented in the paper. Cost benefit analysis points out advantages of the proposed WLHP system. Further research will be directed to simulations of real system necessary for design, after user's decision on the level of refurbishment. Prerequisite for such a simulation is the choice of equipment suitable for particular buildings and entire system in the chosen level of complexity.

## 7 References

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