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EVALUATION OF THE EFFECT OF USING A DRAINBACK SOLAR THERMAL INSTALLATION TO SUPPORT THE HEATING OF A PUBLIC BUILDING

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Abstract: *In the report, an analysis of the potential effect of using a drain-back type solar thermal installation to support the heating of a public building was carried out. For this purpose, a numerical modelling of the heat removal coefficient and the heat energy obtained from the installation were caried out. It was made comparison on an annual base of useful heat from drain-back solar thermal installation and pressured one. The effect on an annual basis has been evaluated using the f-method for both types of thermal installations, when using identical flat selective collectors.*

Keywords: *Efficiency, Effectiveness, GPS, Seismic Protection Methods, Model*

INTRODUCTION

The main method of protecting solar thermal installations from freezing during the winter months is the use of a mixture of glycol and water. The report discusses a drain-back-type solar thermal installation, in which only water is used as a heat carrier. Water as a heat carrier has several advantages over mixtures of water with glycol, the main ones of which are: higher specific heat capacity, lower viscosity, low cost, and less volumetric expansion, in the case of stagnation temperatures there is no danger of chemical decomposition and change of Ph level, etc. In addition to advantages, the use of water in the installation has its disadvantages, which are related to the type of used solar thermal collectors, pipes outside the building must be installed with some minimum slope, larger diameter of pipes, increased corrosion if steel pipes are used, the risk of pump cavitation, higher electricity consumption from the circulation pump in the collector loop, etc. (Botpaev, R.; Louvet, Y.; Perers, Bengt; Furbo, Simon; Vajen, K.)

In this report, an analysis from the energy point of view of the effect of using a drain-back solar thermal installation versus a classical one with a glycol heat carrier has been conducted.

EXPOSITION

Description of the object of study

The object of study is a public building with a heating system supplied by the district network. The building is three-storey with partial heat insulation. The average monthly heating load of the building was estimated by the degree-day method, which is not the subject of this report. On the south slope of the roof of the building, it is planned to build a solar thermal installation to support heating. The slope of the roof is 25° and the solar thermal collectors will be installed directly on it. The orientation of the slope is 34° in the southeast direction. The building is located at 43.5° north latitude on the territory of the town. Ruse. The solar thermal installation consists of 120 flat selective collectors, each with an area of 2.5m2, divided into six groups of 20 pcs. Each. The individual groups have their pumping unit and are connected to a water heat accumulator with a total volume of 24m^3. Each group of 20pcs. solar thermal collectors are connected to a heat accumulator with a volume of 4m3. The report analyses the thermal energy yield of the solar thermal installation thus described and of a similar one executed as drainage. The pipe and ID schemes of these two installations are presented in Fig. 1.

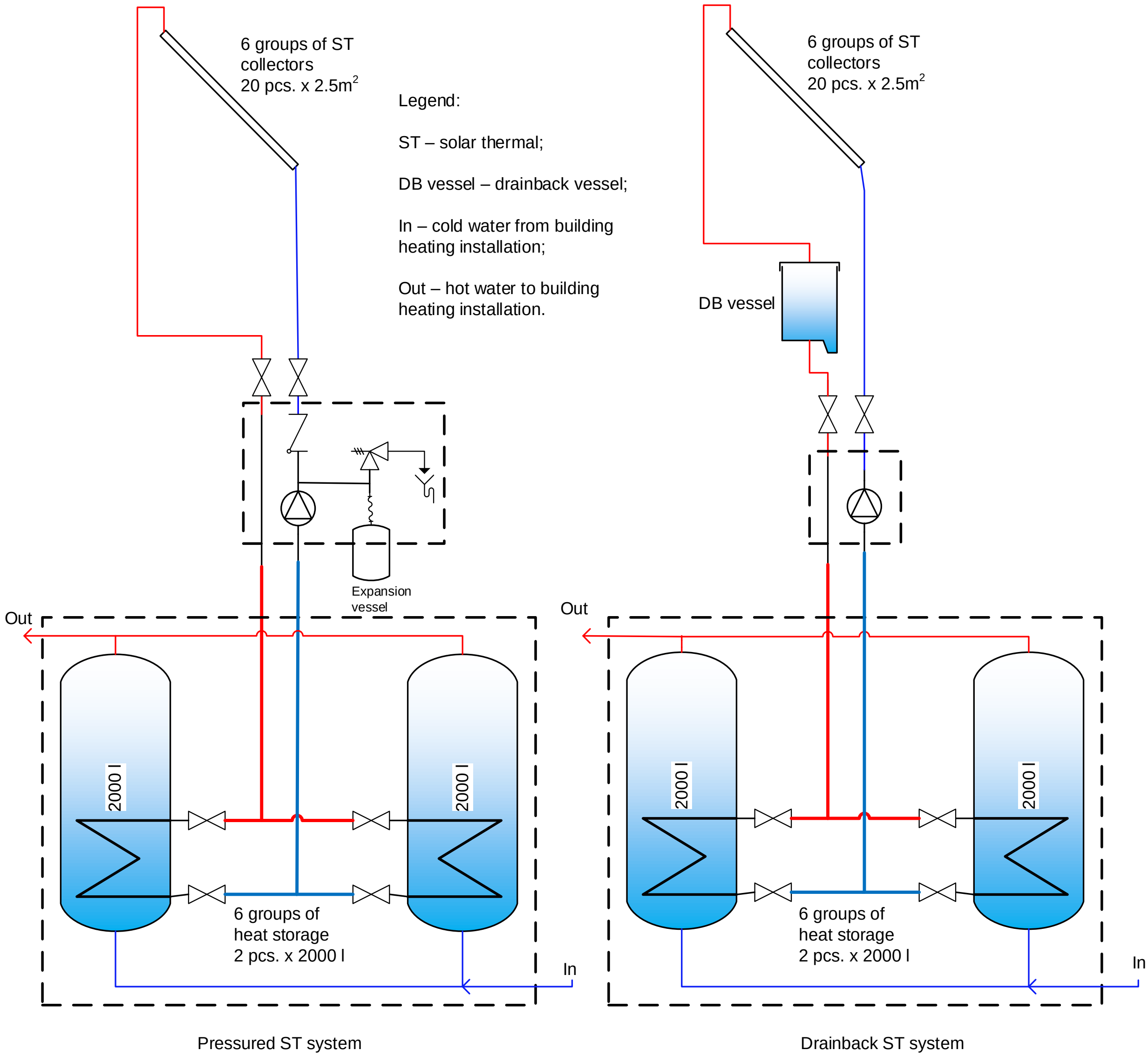

Fig. 1. P&ID diagram of pressured and drain-back solar thermal installations

The flat plate solar thermal collectors used in the installation have the following characteristics: Gross area 2.51m^2; Aperture area 2,29m^2; $\tau\alpha$=0.791; overall heat loss coefficient U_L=4.176 W/m^{2}K; Tinox coated absorber with absorption coefficient α-0,95 and emission coefficient ε - 0,05, absorber thickness 0,5mm aluminium, diameter of absorber tubes 8x0,5mm with a length of 1883mm each, distance between absorber tubes 76,5mm; 15 pcs tubes in the absorber; collector tubes of the absorber ф22x0,7mm; water content of the absorber 1,9 l.

The operating parameters of the solar thermal installation are as follows: the specific mass flow rate of a heat transfer medium through the solar thermal collectors for both types of solar systems is 15g/s.m^2; propylene glycol-water solution with a concentration of in the solar thermal collector loop 40% for the pressurized installation; in the drainback installation, water is used as a heat carrier in the solar collector loop; The heat accumulators are filled with water and the flow rate of heat carrier through the heat exchanger coils in them is equal to the flow rate of the heat carrier from the heating system of the building. The efficiency of heat exchanger in the heat accumulator is 45%; efficiency of the heat exchanger between the heat accumulator and the building installation is 100%; heat losses from the pipes connecting the heat storage and the solar thermal collectors 3% of the complete heat losses of the collectors; specific storage volume for both types of installations 85,836 l/m^2; full heat loss coefficient from the buffers 5.9W/°C; minimum temperature of the water in the buffers is +60°C; average air temperature in the buffer room is 10°C plus the average daily ambient temperature for the corresponding month;

The thermophysical properties of water and a 40% solution of propylene glycol and water are presented in Table 1.

Table 1 Thermal properties of different heat carriers used in solar thermal installations.

Property	Value	Measure
Specific heat capacity, water - Cp_db	4181	J/(kg.K)
Specific heat capacity, propylene glycol solution and water - Cp	3802	J/(kg.K)
Thermal conductivity, water - λ_db	0.6229	W/(m.K)
Thermal conductivity, propylene glycol and water solution - λ	0.4198	W/(m.K)
Kinematic viscosity, water – ν_db	5.515×10^{-7}	m^2/s
Kinematic viscosity, dilution into propylene glycol and water – ν	1.602×10^{-6}	m^2/s
Density, water – ρ_db	988.2	kg/m^3
Density, propylene glycol and water solution – ρ	1013	kg/m^3
Prandtl, water – Pr_db	3.543	-
Prandtl, propylene glycol and water solution - Pr	14.7	-

Calculations were carried out for one tube of the solar thermal collector, using a specific mass flow rate $\dot{m} = 2.289 \mathrm{x} 10^{-3}$ kg/s.

The Reynolds value is calculated by the following relationship for the two heat carriers used.

$$Re = \frac{4.\dot{m}}{\pi.\mu.d_i} \qquad (1)$$

$\dot{m} = 2.289 \mathrm{x} 10^{-3}$ kg/s - the mass flow rate through one tube from the manifold, μ – the dynamic viscosity of the heat carrier, di – the inner diameter of the tube.

The Nusselt value is obtained according to the following relationship according to the literature source (Duffie, J.A., Beckman, W.A. *Solar Engineering of Thermal Processes*. 2006 Third ed. Wiley).

$$N_u = N_{u\infty} + \frac{a \cdot [Re.Pr.d_i \,/\, L]^m}{1 + b(Re.Pr.d_i \,/\, L)^n} \qquad (2)$$

$N_{u\infty} = 4.4$, Pr value is from table 1 depending on the used heat carrier, L – length of collector tube, a=0.00236, b=0.00857, m=1.66 and n=1.13

The convective heat transfer coefficient in the collector tubes was determined by the following relationship.

$$\alpha_{fi} = \frac{Nu.\lambda_{fi}}{d_i} \qquad (3)$$

λ_{fi} – is the thermal conductivity of the heat carrier used according to Table 1

The results of applying equations from (1) to (3) incl. are presented in Table 2

Table 2 Heat transfer in solar collector tubes for two different heat carriers

Property	Value	Measure
Reynolds, water - Re_db	761.9	-
Reynolds, propylene glycol solution and water - Re	256.6	-
Nusselt, water - Nu_db	4.227	-
Nusselt, propylene glycol solution and water - Nu	4.404	-
Coefficient of convective heat transfer, water - α_db	386.8	$W/(m^2.K)$
Coefficient of convective heat transfer, propylene glycol and water solution - α	304.9	$W/(m^2.K)$

The efficiency of one fin of the collector absorber depends on its geometry and is determined by the following dependence according to literature sources (Kalogirou, S., (2009*). Solar energy engineering: processes and systems,* Elsevier) and (Duffie, J.A., Beckman, W.A. *Solar Engineering of Thermal Processes*. 2006 Third ed. Wiley)

$$F = \frac{tanh[m(W - D) / 2]}{m(W - D) / 2} \tag{4}$$

W=76.5mm – the distance between the absorber tubes, D=8mm – the outer diameter of the absorber tubes, m – parameter taking into the account properties of the material from which the absorbing surface is made and the heat loss of the collector to the environment, which also depend on its geometry. The parameter m is defined by the following relationship.

$$m = \sqrt{\frac{U_L}{\lambda \cdot \delta}} \tag{5}$$

U_L=4.176 W/m^{2}K – is the overall heat loss coefficient of the solar thermal collector. λ – conductivity of the absorber material. In this case, the absorber is made of aluminium with λ=237.3 W/m.K, δ – the thickness of the absorber δ=0.5mm.

The efficiency of the entire solar thermal collector absorber was calculated according to the following dependence according to literature sources.

$$F' = \frac{1 / U_L}{W\left[\frac{1}{U_L[D + (W - D) \cdot F]} + \frac{1}{C_b} + \frac{1}{\pi \cdot di \cdot \alpha_{fi}}\right]} \tag{6}$$

$1/C_b$ — the thermal resistance of the connection between the absorbing surface of the collector and the pipes. In this case, this value is negligible. d_i – the inner diameter of the absorber tubes, a_{fi} – the coefficient of convective heat transfer between the tubes and the heat carrier in them according to Table 2.

Accordingly applying dependence (6), for the collector efficiency in the drainage solar thermal installation F'_db=0.9568 is obtained, and for the collector in the classical thermal installation with a heat carrier propylene glycol solution and water a value of F'=0.943 is obtained.

It is of interest to determine the heat removal factor from the collector for the two types of installations operating with different heat carriers. For this purpose, a well-known dependence (7) on literature sources is used.

$$F_R = \frac{\dot{m} \cdot C_p}{Ac \cdot U_L} \cdot \left[1 - e^{\left(-\frac{A_C U_L F'}{\dot{m} C_P}\right)}\right] \tag{7}$$

In the equation (7) the value of the mass flow rate is $\dot{m}$ is for the overall solar thermal collector not for one pipe in it.

The mass flow rate for both cases is the same in equation (7) there are different values for Cp and F’ depending on used heat carrier in the installation.

The heat removal factor for the drain-back solar thermal installation according to equation (7) is F_R=0.9243 and for the installation with propylene glycol solution, the value is F_R=0.9084.

Fig. 2 presents in graphic form the amount of solar radiation for one day on a horizontal surface H_{bar} [MJ/m^2.day], in the plane of the solar collectors $H_{bar,\ T}$ [MJ/m^2.day] and absorbed by the solar collectors $S_{bar,\ T}$ [MJ/m^2.day] by months. Fig. 3 and Fig. 4 are also presented respectively the daily average outdoor air temperatures and degree-days for heating for the settlement.

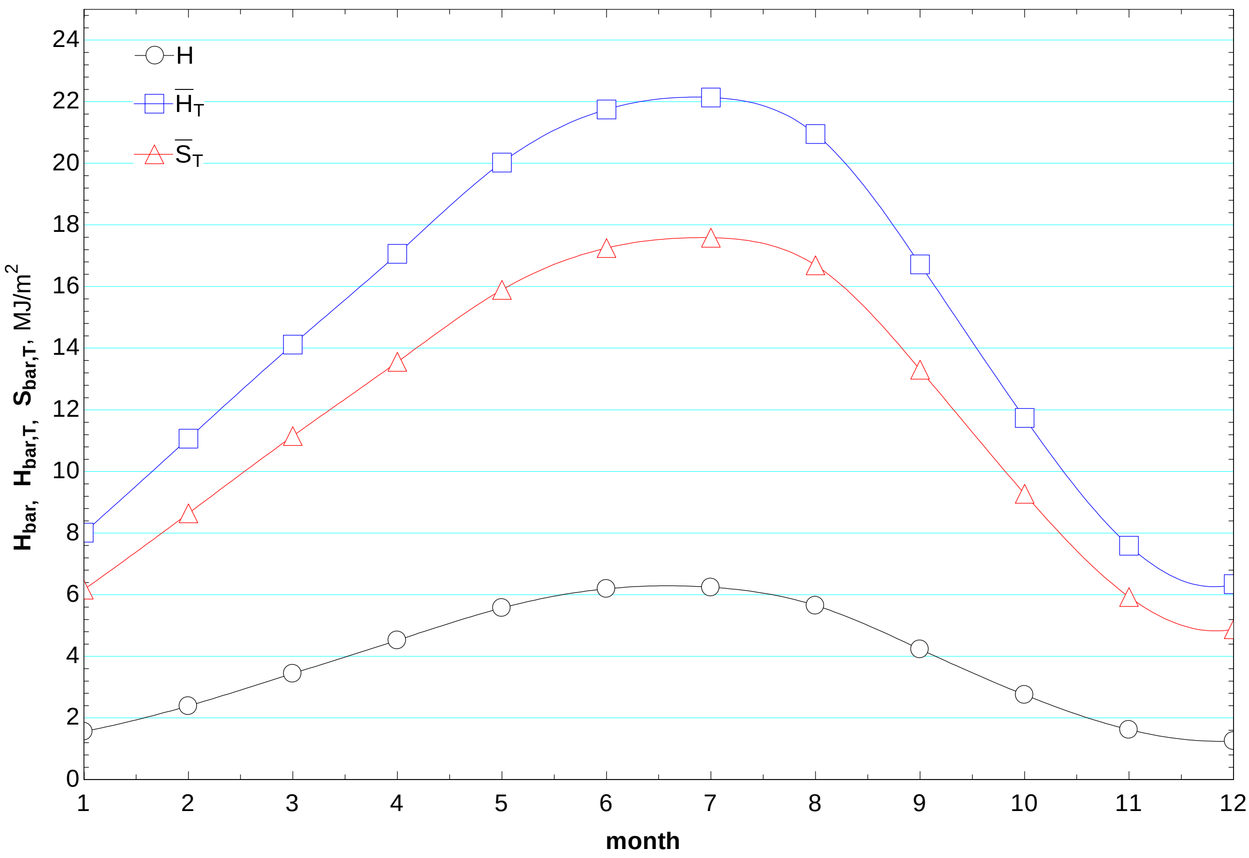

Fig. 2. Daily solar energy on a horizontal surface, on the surface of solar thermal installation and absorbed energy

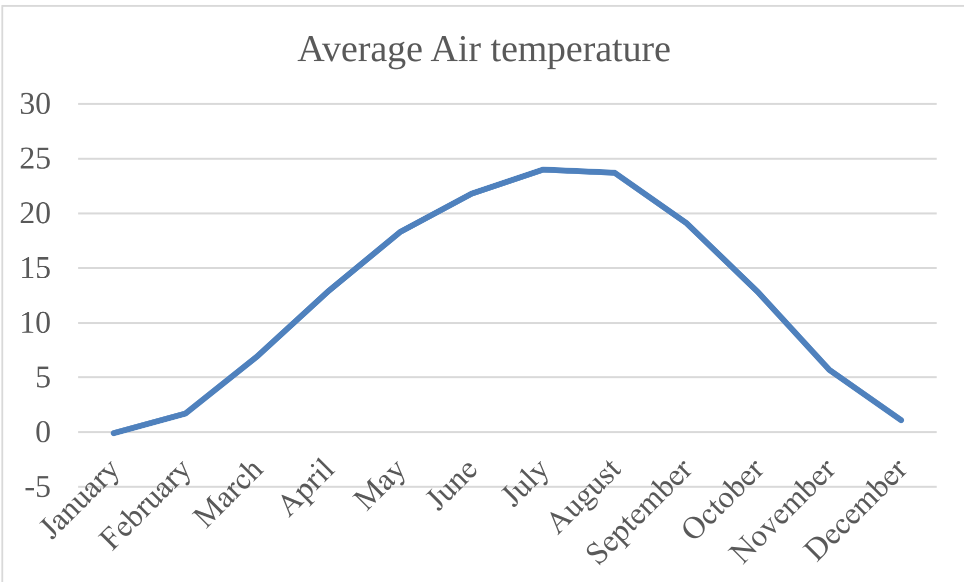

Fig. 3. Average daily ambient air temperature

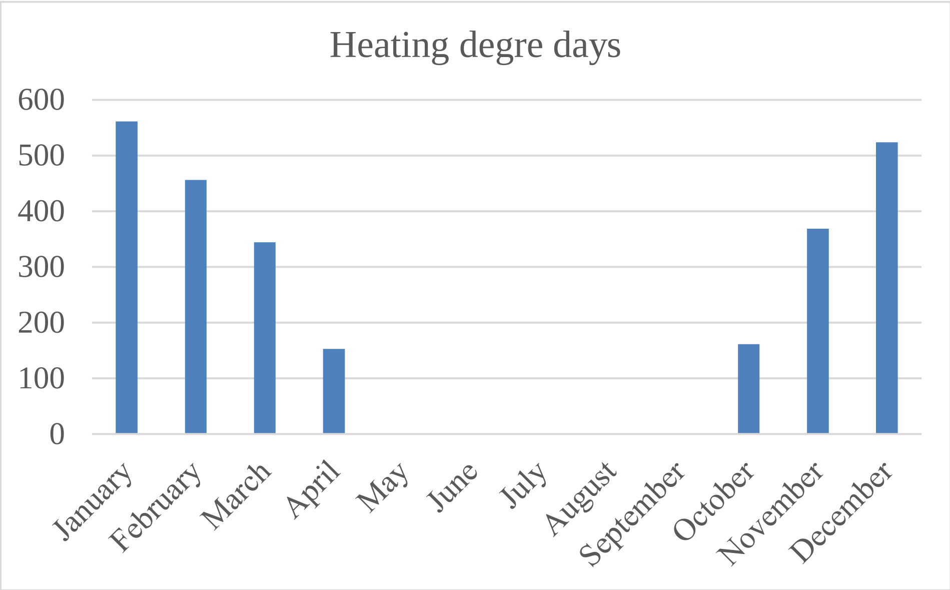

Fig. 4. Heating degree days

RESULTS

An assessment of the daily amount of heat energy received from both types of installations was carried out. For this purpose, dependence (8) has been applied for each month of the year.

$$Q_u = F_R A_C [S - U_L (T_i - T_a)] \qquad (8)$$

Ac=274.7 m^2 is the area of all thermal collectors in the installation, T_i is the inlet temperature of the heat carrier, which in this case is assumed to be a constant of +60 °C, T_a is the average daily temperature of the ambient air for the respective month.

The results of the application of equation (8) are presented in Fig. 5.

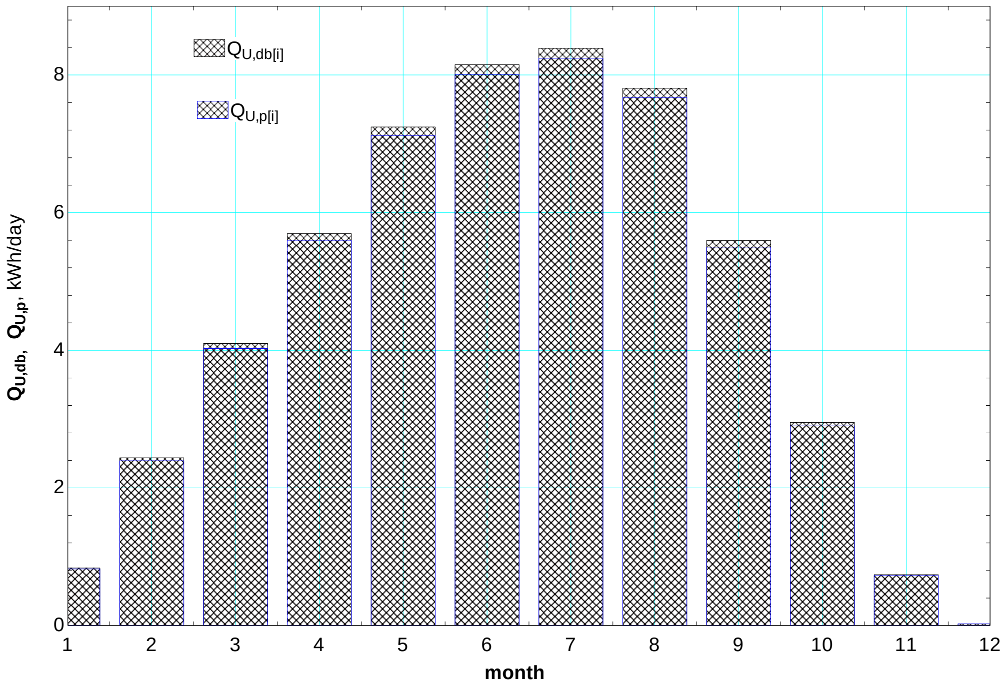

Fig. 5. Daily thermal energy from the drain back /$Q_{U, DB}$/ and normal solar thermal installations /$Q_{U, P}$/

To evaluate the effect of using a drain-back type solar thermal installation in front of the classical one the well-known f-method was applied according to literature sources (Duffie, J.A., Beckman, W.A. *Solar Engineering of Thermal Processes*. 2006 Third ed. Wiley). The load of the building was determined with the degree-day method.

The results are presented in graphical form in Fig. 6. When applying the f-method, it is assumed that the efficiency of the heat exchanger in the solar field contour is 45%, which will further reduce the solar fraction of both types of installations. This low efficiency of the heat exchanger in the solar loop was adopted because of the lack of sufficient technical data on it. A heat exchanger between the buffers and the heating installation of the building is missing, an efficiency of 100% has been assumed.

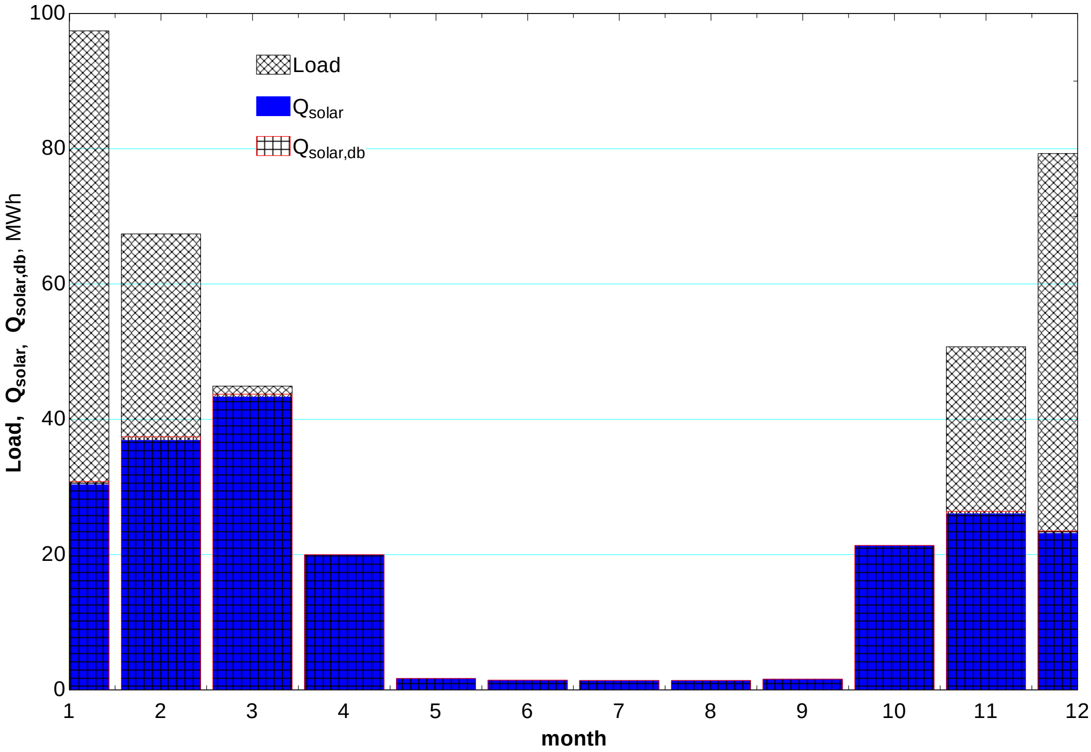

Fig. 6. Monthly thermal energy from the drain back /$Q_{solar,db}$/, normal solar thermal installation /Q_{solar}/ and heating load of the building /Load/

The solar fraction on an annual basis of the drain back type solar thermal installation amounts to 54.2% and the solar fraction of the classical one amounts to 53.65%.

The annual heating load of the building is estimated at 388.5MWh, the amount of heat energy from the drain back type solar thermal installation is 210.6 MWh and that from the classical one is 208.4 MWh.

CONCLUSION

From the performed model study of the efficiency of the two types of solar thermal installations, it is evident that the drain back type has a negligible advantage of 1.05% concerning the heat energy obtained over the classical installation.

Regarding the problems associated with stagnation temperatures in the thermal installation during the summer months and in the periods when the building does not need heat, the drainage type of installation is better.

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