Energy Performances Of A Radiant Floor Heating System Supplied By Solar Collectors With Ventilation Stream Heating By An Air To Air And An Air To Water Heat Exchanger

Oliveti G., Arcuri N., Bruno R., De Simone M.

Mechanical Engineering Department - University of Calabria – ITALY

Corresponding email: natale.arcuri@unical.it

SUMMARY

The energy analysis of a heated space with a radiant floor plant supplied by hot water provided by a storage system charged by a field of solar collectors, has been evaluated. A suitable control strategy has been evaluated to reduce thermal inertia effects. Given an external air-change ventilation flow rate must be provided, a system using two heat exchangers has been used: the first an air to air heat exchanger that uses the expulsion air from the heated space as hot fluid, and a second air to water heat exchanger that uses the outlet water from the radiant floor. The whole system has been studied by dynamic simulation code TRNSYS, to evaluate interaction between the devices and, overall, the contribution of the solar source on plant performance by the solar fraction, varying the collectors' surface and the storage system volume.

INTRODUCTION

Energy consumption in the building field is anything but ignorable considering that today, in the European Community, such a voice concurs a percentage higher than 40%. Some recent community directives [1], promote the energy consumption reduction in the winter heating of buildings. This result is obtainable by means of multiple solutions such as efficient building insulation of opaque surfaces, a wise choice of glass surfaces, the adoption of high efficiency thermal energy production, the use of low temperature energy distribution systems, powered by a supply coming from a field of solar collectors. The latter solution assumes greater value for houses situated in localities with cold climates, whose requirements for thermal energy for heating are considerable compared to other energy requirements of the building.
Among the various types of thermal energy distribution systems used for heating areas, radiant floor heating systems are those which permit the best direct use of solar radiation as a primary energy source. In fact, the floor surface temperature must be moderated so that a relatively low temperature fluid can be used which is obtainable from the solar source with elevated efficiency [2]. Another advantage deriving from the use of radiant floor heating systems lies in the reduction of winter energy requirements. The increase of average temperature radiating within the area permits the attainment of the same operative temperature with lower internal air temperatures, with a consequent reduction in transmission and ventilation losses through the building shell. On the other hand, radiant floor systems, compared to traditional ones, are characterised by notable thermal inertia, therefore the adoption of control systems which take into account this peculiarity is necessary [3]. In this article, thermal performance of radiant floor heating systems powered by an storage system

which receives energy from a field of solar collectors, with preheating of the ventilation airflow obtained by exploiting the building exiting airflow from an air to air exchanger and, successively, the exiting flow rate from the radiant floor in a second air to water exchanger are evaluated.

DESCRIPTION OF THE BUILDING USED IN THE SIMULATION

The building is considered as being for office use, it has a regular parallelepiped shape, with the longest side being 16m, and the shortest being 6m and a height of 2.75m. The walls of the largest surface area are North and South facing. On each external wall and triple glazed window is present having a global loss coefficient equal to 1.8 W/m²K and a solar gain coefficient of 70%. The glazed surfaces are respectively 8.8 m² for the south facing wall, and 3.3 m² each for the remaining three exposures. The total glazed surface area is therefore 22 m² while the total opaque dispersant is equal to 291 m². In Tab. 1 the thermal properties of the various opaque components of the building shell are reported. The thermal transmittance values are 0.198 W/m²K for the vertical walls, and 0.146 W/m²K for the covering floor and 0.207 W/m²K for the floor placed on the ground.

Table 1. Layers thermal characteristic of opaque walls.

	Thickness	Conductivity	Density	Thermal Resistance
	m	W/mK	Kg/m^3	m²K/W
Plasterboard	0.018	0.9	1800	0.02
Mineral wool with moisture stop	0.125	0.048	11	2.60
Mineral wool rigid support	0.063	0.035	100	1.80
Air gap	0.03	-	-	0.18
External surface bricks	0.075	-	650	0.28
Wooden beams and battens	0.045	-	850	0.25
Covering material in tile-concrete	0.055	-	1800	0.03
Floor covering material	0.01	1	1800	0.01
Light Concrete	0.09	1.2	1800	0.08
Heavy concrete	0.10	1.2	2000	0.08
Rigid panel insulating	0.10	0.035	37	2.85
Ground floor loose stone foundation	0.50	-	2000	0.28

In order to determine the internal energy gains it is supposed that the building is used as an office with a number of people equal to 12, present from 08.00 to 13.00 and from 14.00 to 18.00, from Monday to Friday, who work with the same number of computers. For each person a sensible load of 65W is estimated and a latent load of 55 W [4]; each computer transfers a sensible load of 80W. The lighting plant is active from 8.00 to 18.00 and causes a thermal flux equal to 5 W/m², for a sensible load of 480 W. Finally, with regards to ventilation, for each occupier an air-flow rate of 11 l/s was estimated [5].
The radiant floor properties are as follows [6]:

- pipes in polymeric material with k=0.33 W/mK, internal diameter of 0.01 m and a tube pitch of 0.15 m;
- light concrete layer with a thermal conductivity equal to 1.2 W/mK and a thickness of 0.1 m, covering material floor with k=1 W/mK and a thickness of 0.01 m.
- an inlet flow rate equal to 1191 kg/h;

DESCRIPTION OF THE HEATING SYSTEM

The plant of the system considered is reported in Fig. 1. The inlet flow rate of the radiant floor is taken from the storage tank which receives energy from the collectors field. In the case in which the temperature of the storage system is higher than that requested, a mixing by means of valves 1 and 2 is carried out to reach the desired temperature. In the case that the temperature is lower, an auxiliary system intervenes by means of the commutation of valves 3 and 4. The auxiliary system is situated parallel to the storage system, in such a way that the water flow feed is supplied either by the storage tank or by the auxiliary system. This planning configuration avoids the temperature of the water flow produced by the auxiliary system being conveyed to the storage system.
To reduce losses due to ventilation, a system which employs two heater exchangers is used. In the first exchanger, the external airflow is heated in counterflow of the expulsion airflow from the indoor environment; the second also in counterflow, uses the flow rate exiting from the radiant floor before this returns to the storage tank or auxiliary system. The sizing of the heat exchanger was carried out imposing a global thermal exchange coefficient so that for both 50% efficiency was obtained.

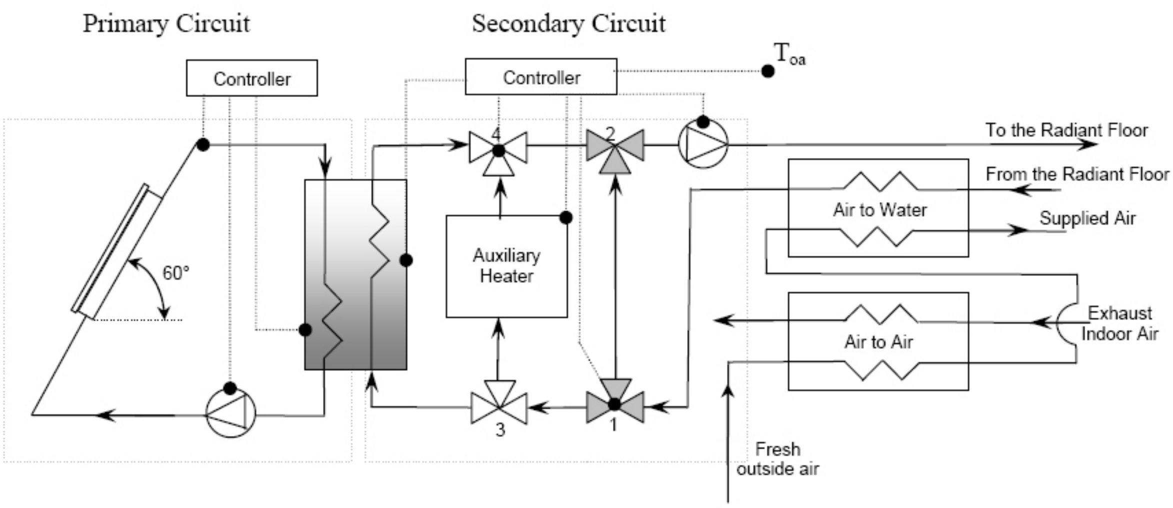

Figure 1. Plan of the solar plant for building heating.

SIMULATION CODES AND CONTROL STRATEGIES

The whole system was studied using the TRNSYS simulation code [7], which consents detailed simulation in a dynamic conditions to evaluate exchange mechanisms relating to the collector surface, storage tank, radiant floor and the indoor volume to be heated.
Moreover, the considered dynamic simulation environment has permitted the study of a adequate control strategy, with the aim of a rational use of the thermal level of the storage tank and the optimisation of the thermal performance of the radiant floor, otherwise penalised by excessive thermal inertia. In order to evaluate the energy performance of the entire system, the building was situated in three different localities in Europe having different climatic characteristic: Helsinki, Copenhagen and Munich.
The hourly values of solar radiation and external air temperature used in the simulations were generated by means of a special TRNSYS subroutine, starting from corresponding average monthly data of the locations considered, in such a way as to generate a metrological year type (TMY) [8]. For the average monthly temperature and horizontal solar radiation values

for the considered locations, those contained in the European Solar Radiation Atlas (ESRA) [9] were used. The values of diffuse and beam radiation on the horizontal plane were obtained by Reindl relations [10], which use the solar height angle and index cloudiness. The projection of such components on the field of collectors and on the building walls was carried out using the isotropic sky model [11]. The average monthly values of external air temperature and solar radiation on the horizontal plane during the heating period relating to the three considered locations are listed in Tab. 2.

Table 2. Average monthly daily values of external air temperature (T_{OA}) and of solar radiation on the horizontal plane (HSR) .

		Sep	Oct	Nov	Dec	Jan	Feb	Mar	Apr	May
Helsinki	T_{OA} [°C]	10.0	5.9	-0.1	-4.2	-6.9	-6.1	-2.4	3.6	10.4
	HSR [kWh/m²]	2.16	1.00	0.03	0.12	0.24	0.89	1.96	3.68	5.27
Copenhagen	T_{OA} [°C]	14.0	9.4	5.3	2.5	0.0	0.2	2.1	7.0	11.8
	HSR [kWh/m²]	2.67	1.43	0.66	0.32	0.47	1.08	2.02	3.81	5.07
Munich	T_{OA} [°C]	14.4	9.7	3.3	0.8	-1.0	-0.9	4.2	7.7	12.6
	HSR [kWh/m²]	3.37	2.11	1.14	0.74	0.96	1.79	2.73	3.94	5.03

Given that the function of the solar collectors is prevalently in winter, the inclination angle of the collectors field is 60° for all three locations, to maximise the solar radiation incising on the collector field. The latter are characterised by the following efficiency expression:

$$\eta = 0.7847 - 4.4894 \cdot \frac{T_m - T_{oa}}{G} - 0.027 \cdot G \cdot \left(\frac{T_m - T_{oa}}{G}\right)^2, \qquad (1)$$

with G (W/m^2) solar radiation, T_{oa} external air temperature, and T_m average water temperature in the collector field.

CONTROL STRATEGIES

In radiant floor heating systems, due to elevated thermal inertia, the type of control which must assure reduced plant response times and avoid thermal discomfort conditions assumes great importance. The methodology used is the control of the inlet temperature combined with a traditional ON/OFF type control, on the inlet flow rate in the case of overheating of the indoor environment (T_{ia}>22°C), due to eventual free energy gains, and its restoration when the temperature goes below a determined limit (T_{ia}<20°C).

The inlet temperature depends principally on the external air temperature, which in the heating period represents the prevalent external constraint on the heated area. Such a control takes into account the effects linked to floor thermal inertia, since the temporal delay with which the floor inlet temperature variation is carried out on the area is comparable with that with which a variation in external air temperature manifests itself within the building by means of dispersing walls. Therefore, no predictive estimate of the feed temperature is necessary, such a criteria is much truer as to the thermal inertia of the floor, and of the same entity of the external dispersing walls. The inlet temperature is that which guarantees an internal air temperature of 20°C in the least favourable conditions, in the absence of solar radiation, and is made to depend linearly on the external air temperature. Imposing the condition that the floor must not transfer thermal power when the external air temperature is 20°C , the law which determines the feed temperature assumes the following expression:

$$t_{INLET} = k(20 - t_{OA}) + 20, \qquad (2)$$

where t_{OA} is the external air temperature while k constant factor depends uniquely on the average global transmittance of the building's dispersing walls. For the structure taken into consideration, the value of variable "k" assumes the value -0.2183.
In order to improve the control function and minimise floor response times, the control relation (2) was further corrected multiplying it by a secondary function, linked this time to the internal air temperature. The logic is based upon the possibility of correcting the inlet temperature obtained from (2) with a minor unit coefficient when the internal air temperature passes a temperature of 20°C, thus making it unnecessary to increase the floor's thermal power, even when the external environment has a relatively harsh temperature, and of using a greater unit coefficient if the internal air temperature is lower than 20°C, in such a way as to render the plant response more rapid when the internal air temperature conditions are unfavourable. The relation used for evaluating C.C. corrective factor in function of the internal air temperature t_{IA} is the following:

$$C.C. = -0.11 \cdot t_{IA} + 3.2 \qquad (3)$$

Using relation (3), the feed temperature results as being 40% higher compared with that necessary determined by (2) if the internal air temperature is around 16°C, it is almost equal to that calculated with (2) if the internal air temperature is around 20°C, while the C.C. corrective factor is equal to about 0.8 if the internal air temperature is near 22°C. Combining relations (2) and (3), the real radiant floor feed temperatures assume the form:

$$t_{INLET} = [k(20 - t_{OA}) + 20] \cdot (-0.11 \cdot t_{IA} + 3.2) \qquad (4)$$

The use of relation (4), determines the raising of the inlet temperature of the radiant floor during the first hours of the day, in which the internal and external air temperatures assume relatively low values, limiting the plant's slow response at the moment of turning on. For such a reason, the implemented control strategy also foresees an attenuation operation at night and at the weekends, to avoid internal air temperatures going down too low. The attenuated operation was rendered necessary also to avoid too high feed temperatures which derive from the use of (4), whose field of application is effectively reduced to the internal air temperature range 16÷22°C. The attenuated operation starts when the internal air temperature goes below 16 °C. In such a case, the radiant floor feed temperature is 25°C, in order to exploit the thermal level of the tank and end at the reaching of 18°C.

RESULTS OBTAINED FROM THE SIMULATION

In Fig. 2, the hourly trends of the internal air temperature, of water in the storage tank, of the requested inlet temperature and of the inlet flow rate to the supply terminals for a period of 48 hours (7^{th} and 8^{th} of February) for Helsinki, are illustrated.
Feed temperature modulation, with values lower than 25°C avoids interruption of the plant feed thus permitting the exploitation of the storage tank even with low temperatures, is to be highlighted. Moreover, in some hours of the day, the inlet requested temperature is lower than the tank water temperature, and in the remaining hours the flow rate is provided entirely by the auxiliary system and the temperature foreseen by (4). Naturally, when the temperature in the tank is greater by less than 1°C compared to that requested, the recycling and extraction

flow rate are nil because the airflow feed is prepared entirely by the auxiliary system with the aim of equipping the control system with minimum hysteresis.

In Fig. 3, for the same period, the temperature rise undergone by the ventilation airflow as an effect of heat recovery, is illustrated. The air to air exchanger is always in use, from 8.00a.m. to 6.00p.m., given that the exchange air flow must always be guaranteed during office opening hours. Heat recovery assumes a revealing role: for example at 8.00a.m. on the first day the external air temperature of -7.3°C was raised to +6.3°C. The second heat exchanger is only operative when the radiant floor plant is in use: in this specific case, since the return temperature from the radiant floor is slightly lower than 35°C, the air temperature undergoes an increase from +6.3°C to +17.2°C,. Such a recovery amply reduces load entity induced by ventilation airflow.

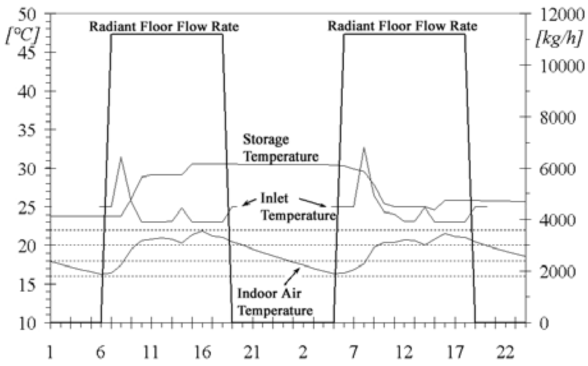

Figure 2. Trends of indoor air temperature, storage tank water, inlet temperature, flow rate to supply terminals (7th and 8th of February, Helsinki)

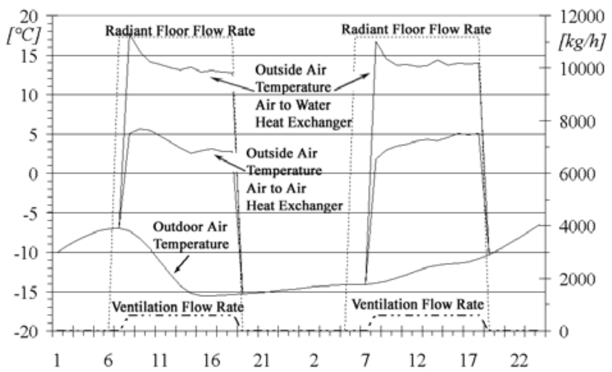

Figure 3. Ventilation flow rate temperature profiles (7th e 8th February, Helsinki).

In Tab. 3, for the three locations examined and for the entire heated period, which goes from September to May, the percentages of thermal energy requirements provided by the radiant floor (F_1), of the air to air exchanger (F_2) and of the air to water exchanger (F_3) are reported. It is possible observe how the percentage of thermal requirement recovered takes on important values, with a greater weight for the air to air exchanger which works continuously for 10 hours per day.

Table 3. Required seasonal energy for heating and seasonal fractions provided by the radiant floor (F_1), from the air to air exchanger (F_2), and the air to water exchanger (F_3).

	Heating Demand	F_1	F_2	F_3
	[GJ]	[%]	[%]	[%]
Helsinki	42.220	44.1	38.2	17.7
Copenhagen	29.053	39.1	44.9	16.0
Munich	27.255	36.4	48.7	14.9

In Fig. 4 seasonal energy provided by the storage tank, by the auxiliary, by the radiant floor, and by the two heat exchangers varying according to the collectors surface for an accumulated volume of 4 m^3 is reported for the city of Munich. It is possible to observe that both the energy provided by the radiant floor and that provided by the two heat exchangers are independent of the collectors surface, while energy provided by the fluid to the floor and to

the air to water exchanger is the sum of the energy provided by the storage tank and the auxiliary. The contribution in terms of energy provided by the tank is a high percentage.
In Fig. 5 trends of the average seasonal temperature of fluid present in the storage tank, and the efficiency of collectors, varying the collectors surface for an accumulated volume of 4 m^3 are reported. It is possible to observe that the average seasonal water tank temperatures increase with an increase in the collectors surface, with higher values for Munich due to a higher insolation level and lower for Helsinki which, in addition to having lower insolation, presents a greater thermal requirement. The average thermal efficiency of the collectors has, as a consequence, a decreasing pattern with the collectors surface, even if it does not alter very much for the three considered locations. Energy provided by the storage tank is reported in Fig. 6. Rising values are encountered with the collectors surface and strongly linked to incising solar energy. The best values are obtained for Munich which has greater insolation.
Finally, in Fig. 7, trends of the requirement fraction provided by the solar source which result as growing with the collectors surface and storage volume, are highlighted. In such a calculation, energy recovered in the air to air exchanger is not considered, in that it is not influenced by the feed system. The greater solar fraction values were obtained in Munich, reaching a value of 0.73 for a 40 m^2 collector surface and a storage volume of 4 m^3. For Copenhagen, the solar fraction varied from a minimum of 0.22 (S=10 m^2, V =1 m^3) to a maximum value of 0.52 (S=40 m^2 e V=3 m^3). Lower values were encountered in Helsinki, due not only to lower insolation, but also to a higher thermal requirement. For such a location the maximum solar fraction value resulted as being equal to 0.36.

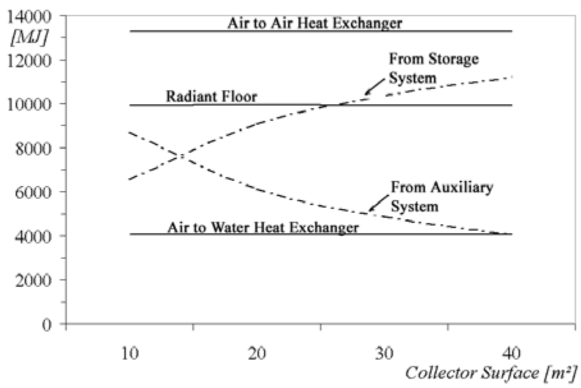

Figure 4. Seasonal energy provided by the storage tank, by the auxiliary, by the radiant floor, and by the two heat exchangers varying according to the collectors surface (Munich V=4 m^3).

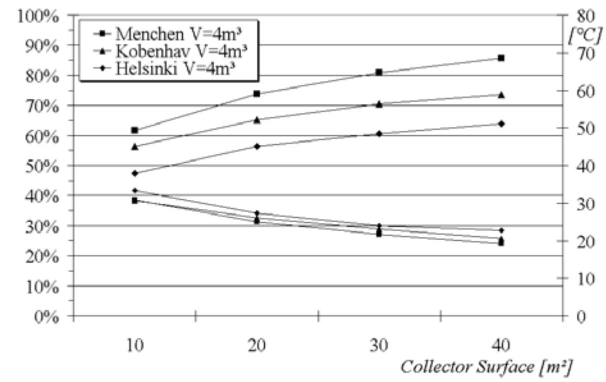

Figure 5. Storage tank average seasonal temperature trend and collectors' average efficiency.

CONCLUSIONS

The thermal performance of radiant floor heating systems supplied by a solar system were identified. The ventilation flow rate to the indoor environment is heated in two successive phases, the first using the air feed expelled from the indoor and the second with the feed exiting the radiant floor. The plants provide heating for areas used as offices in Munich, Copenhagen and Helsinki. This type of plant requires an efficient control system which assures the use of energy available from storage even when the temperature level is low, while thermal comfort conditions within the building are acceptable.
These plant configurations permit, by means of thermal recovery, the obtainment of significant and higher requirement fractions than those provided by the radiant floor.
The results obtained can be summarised as follows:

Winter requirements are delivered to the indoor environment by the radiant floor in the following percentages: 36.4% for Munich, 39.1% for Copenhagen and 44.1% for Helsinki, from the air to air heat exchanger in a percentage including between 38,2 for Helsinki and 48,7 for Munich, and finally from the second air to water exchanger a percentage slightly variable between 14,9% and 17,7%. The requirement fraction provided by the solar source principally depends upon available solar energy, the collectors surface and storage volume.
For collectors surfaces variable from 10 to 40 m^2 and storage volumes between 1 to 4 m^3 the solar fraction resulted as being variable between 0.32 to 0.72 for Munich; from 0.22 to 0.54 for Copenhagen and from 0.13 to 0.34 in Helsinki. Such results demonstrate that obtainable solar contribution with such plants, even in non Mediterranean climates, represents a non negligible fraction of the winter heat requirement.

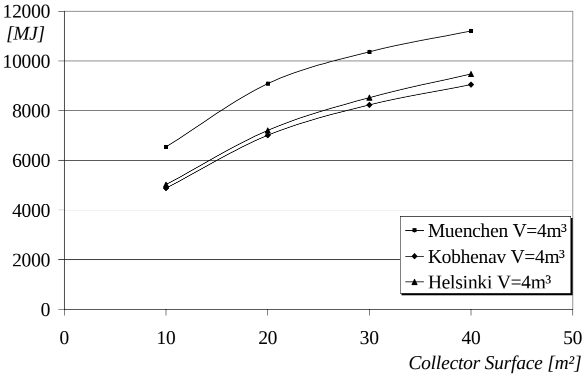

Figure 6. Energy supplied from the storage tank for a volume of 4 m^3 varying the collectors surface.

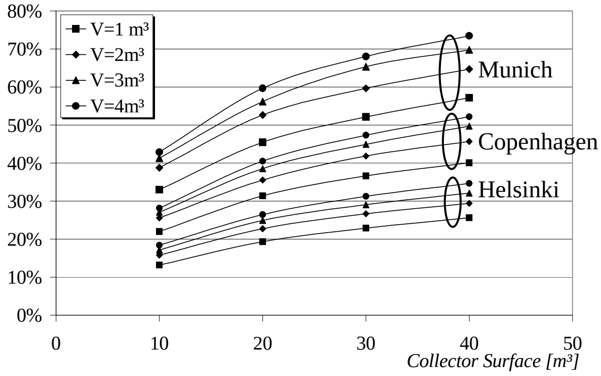

Figure 7. Fraction of the load supplied by the solar source compared to the total.

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