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Passive Cooling Assembly for Flat Panel Displays with Integrated High Power Components

Ilker Tari

Abstract — *Passive cooling of flat panel display designs with integrated high power components is investigated with the help of recently available semi-emprical and CFD based heat transfer correlations. A heat-spreader-heat-sink assembly is proposed for effective external natural convection cooling of the display panel. A flat vertical surface and plate finned heat sinks with various fin heights are considered as heat sinks in the assembly. Heat dissipation limits for both types of heat sinks are determined for various panel dimensions. It is shown that for large panels, it is feasible to use passive cooling even when integrated computer components are used in panels for demanding applications such as video games, high definition video processing and 2-D to 3-D conversion*[1].

Index Terms — Passive cooling, flat panel displays, radiative heat transfer.

I. INTRODUCTION

Flat panel displays are improving very rapidly in terms of technology and functionality. Conventionally separate components such as game consoles, satellite receivers are being integrated into TV panels together with processing units (CPUs and GPUs) and data storage media.

Common flat panel TV displays as well as the ones with basic extra functionality can be cooled by letting ambient air pass through the grills on the back cover of the panel where the electronics are placed. Air entering the back cover from the bottom of the panel through the grill carries heat from the components via natural convection and exits from the top grill. This solution provides sufficient cooling due to the very large available heat transfer area and due to the fact that there are no localized high flux heat sources. However, if one wants to integrate high power components such as Central Processing Units (CPUs) and Graphics Processing Units (GPUs), this internal cooling system may not be adequate due to the localized high flux heat sources. In that case, there is a need for fans or blowers to circulate air inside the panel. Fans and blowers generate noise, use electricity, require active temperature control, and reduce reliability due to their mechanical nature.

Integration of such high heat flux processors with display panels started to appear or is proposed for the future. As new integration applications: 3-D TV, PC TV, TV integrated game consoles, HD PVR, HD content delivery and storage can be cited. For these applications, powerful processors such as the H.264 decoder proposed in [1] are needed. Also some separate units such as the DVDR and PVR systems proposed in [2] and [3] can be integrated into flat panel TV displays. In this study, we propose and analyze a passive cooling assembly for flat panel displays with integrated high power components.

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II. PROPOSED PASSIVE COOLING ASSEMBLY

Flat panel displays are usually cooled by internal natural convection with the help of the grills at the top and bottom of the back cover of the panel. In this study, we propose an alternative passive cooling assembly to achieve effective cooling by external natural convection and radiation. The proposed assembly is shown in Fig. 1. Heat generated by the integrated components is rejected with the help of the heat spreader flat heat pipe and the heat sink. The heat sink also functions as the back surface of the panel. The heat sink shown in Fig. 1 is a vertical plate finned heat sink.

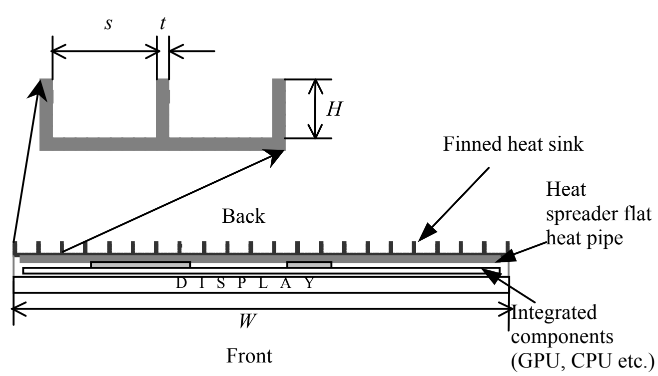

Fig. 1. Top view of the passive cooling heat sink assembly. Heat spreader flat heat pipe is in contact with the integrated components. Plate finned heat sink is attached to the heat spreader. Heat is dissipated from the heat sink with natural convection and radiation.

Two different heat sink geometries are considered for the back side of the flat panel. First one is a flat vertical surface that can only provide enough heat transfer area for low heat dissipation rates. Second one is a finned heat sink as shown in Fig. 1 and Fig. 2 with plate fins of height H (0.005 or 0.01 m) and thickness t=0.001 m. The fin dimensions are selected as dimensions that are ergonomically suitable from the design point of view (not sharp to touch), and that give an overall surface efficiency that is very close to 1 so that entire extended surface is effective in heat transfer.

Heat transfer from the panel components to the heat sink requires very good thermal contact and a heat spreader for uniform distribution of heat. Heat pipes and thermosyphons that are self-powered have been improving rapidly for the last decade and they can distribute heat from localized sources to large surface areas. Especially flat heat pipes are very suitable for this task. In this study, heat pipe heat spreaders combined with a large heat sink surface is proposed as the passive cooling assembly that does not require any power. The assembly is thermally analyzed and cooling limits (achievable heat transfer rates) for various panel sizes are determined.

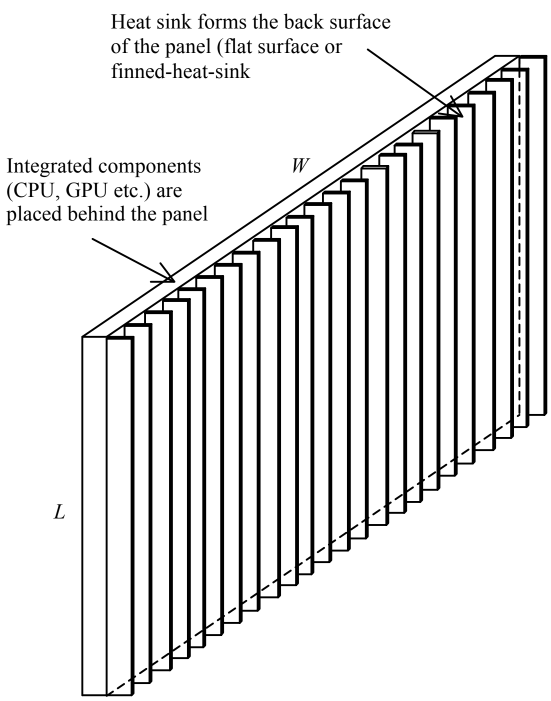

Fig. 2. Plate finned vertical heat sink geometry.

III. ACHIEVABLE HEAT TRANSFER RATES

The heat transfer calculations for the proposed assembly are done by considering both natural convection and radiation as heat transfer modes. To be able to approach the heat transfer problem analytically, some simplifying assumptions are made that are discussed in Section III-A.

A. Assumptions

The ambient air temperature is taken as 25°C. All thermo-physical properties are obtained at that temperature.

Aluminum is selected as the heat sink material for its high thermal conductivity, and lower density and cost compared to similar high conductivity metals.

There is a thin anodized layer on the surface of the heat sink for durability, electrical insulation and high radiative emissivity. By electrically insulating the heat sink, it is possible to use the heat sink as the back surface of the panel.

A simple one dimensional thermal resistance network between the lid components and the ambient air is shown in Fig. 3. Since the anodized layer thickness (t_{ox}) is very small and thermal conductivity of Aluminum (k_{Al}) is very high, the temperature difference between the heat dissipating components and the surface of the heat sink lid is dominated by the thermal contact resistance between the components and the heat sink as well as between the Aluminum and the anodized layer. The combination of the thermal contact resistances together with the resistances of the heat spreaders and heat pipes ($R''_{t,c}$) can only be experimentally determined after the complete design of the assembly. If good thermal contacts can be maintained, the surface temperature (T_s) of 50°C should be enough to keep the component surfaces below 60°C. That is less than the design maximum temperatures (T_{max}) for common integrated circuits and data storage. Therefore, heat sink surface temperature, T_s is taken as uniform 50°C with the assumption that the heat spreader flat heat pipe is capable of spreading heat uniformly. This temperature is also the effective temperature everywhere on the fins. These assumed temperatures are roughly based on the experimental study using a similar integrated heat pipe with heat spreader arrangement by Take and Webb [4] in which they report 60.0-49.2 °C and 59.9-56.1 °C as hot-cold side temperatures for two different heat spreaders.

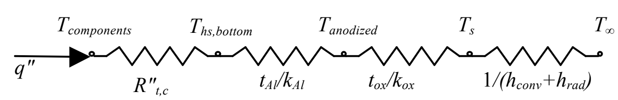

Fig. 3. One-dimensional thermal resistance network between the surfaces of the components and the ambient air. The temperature nodes from left to right are fo components, heat sink bottom surface, anodized layer, heat sink surface and ambient air. The resistance values can only be determined after the complete design of the heat spreader heat sink assembly.

B. Natural Convection Cooling

For the first heat sink option i.e. the flat heat sink, the convective heat transfer coefficient is obtained from Churchill and Chu correlation for laminar natural convection over vertical plate. Churchill and Chu correlation as presented in [5] for average Nusselt number, Nu based on the vertical length L of the vertical flat plate is:

$$\mathrm{Nu}_L = \frac{\overline{h}L}{k} = \left\{ 0.825 + \frac{0.387\mathrm{Ra}_L^{1/6}}{[1+(0.492/\mathrm{Pr})^{9/16}]^{8/27}} \right\}^2 \qquad (1)$$

where $\overline{h}$ is average convection heat transfer coefficient, k and Pr are the thermal conductivity and Prandtl number of air and Rayleigh number based on L, Ra_L is defined as

$$\mathrm{Ra}_L = \frac{g\beta L^3 \Delta T}{\nu\alpha} \qquad (2)$$

Here, g is the gravitational acceleration (9.807 m/s^2); ΔT is the temperature difference between the surface and the ambient air, T_s-T_∞; β, ν, and α are the volumetric thermal expansion coefficient, the kinematic viscosity and the thermal diffusivity of air, respectively. Using (1) for the entire range of Ra_L, the convective heat transfer rate that can be dissipated from a vertical flat plate, $q_{conv,flat}$ can be calculated using:

$$q_{conv,flat} = \frac{\mathrm{Nu}_L k}{L} A(T_s - T_\infty) \tag{3}$$

where flat surface area, A is equal to the product of the length, L and width, W.

For the second heat sink option with plate fins (see Fig. 2), the separation distance of the plates that is denoted with s is important. There is an experimental study that is performed with similar plate fin heat sinks by Yazicioğlu and Yüncü [6] in which they suggested a correlation for optimum separation between the plates, s_{opt} that is taken as the separation distance in the present study:

$$s = s_{opt} = 3.53\ L\ \mathrm{Ra}_L^{-1/4} \tag{4}$$

They also suggested a correlation for the convection heat transfer rate, q_{conv} for a heat sink with optimum plate separation distance by considering the enhancement over $q_{conv,flat}$:

$$q_{conv} = q_{conv,flat} + 0.125\ \mathrm{Ra}_L^{1/2} k H \Delta T \frac{W}{L} \tag{5}$$

In general, (5) gives conservative results, therefore the convection heat transfer rates that are calculated from (5) can be taken as minimum possible values, $q_{conv,finned,min}$. Thus

$$q_{conv,finned,\min} = q_{conv} \tag{6}$$

The separation distance, s values calculated from (4) are close to the separate plate limit, thus the plate fins do not affect each other much. There will be some disturbance where the fins meet with the base that reduces the local heat transfer coefficient due to the increase in the boundary layer thickness at those locations. The upper limit of convective heat transfer rate, although not achievable, can be obtained by taking the entire extended surface as a virtual flat plate with the total area of the finned heat sink. Thus, the same average Nu that is obtained from (1) can be used for calculating the maximum convection heat transfer rate from:

$$q_{conv,finned,\max} = \eta_o \frac{\mathrm{Nu}_L k}{L} A_{total}(T_s - T_\infty) \tag{7}$$

where η_o is the overall surface efficiency of the heat sink that can be taken as 1. Here, the total heat sink area, A_{total} is calculated using s from (4) and by slightly extending the heat sink width to form a heat sink between two plate fins (end plates). The number of fins, N_f is obtained as a function of heat sink width W, fin separation s and fin thickness t from:

$$N_f = \mathrm{ceiling}\left(\frac{W}{s+t}\right) \tag{8}$$

where ceiling() is rounding-up operation. Therefore, the total heat transfer area for the considered fin height H can be obtained from:

$$A_{total} = N_f L(2H + t) + (N_f - 1)Ls \tag{9}$$

Here, the first term on the right hand side is the finned area and the second term is the unfinned area for the heat sink.

For optimum fin spacing, s_{opt} and convective heat transfer rate, q_{conv}, there are two other recent less conservative set of correlations. One of the set of correlations suggested by Yazicioğlu and Yüncü [7] by re-evaluating available studies in literature is the following:

$$s_{opt} = 3.15\ L\ \mathrm{Ra}_L^{-1/4} \tag{10}$$

and

$$q_{conv} = q_{conv,flat} + 0.2116\ \mathrm{Ra}_L^{1/2} k H \Delta T \frac{W}{L} \tag{11}$$

The other set of correlations is suggested by Cakar [8] using Computational Fluid Dynamics (CFD) simulations:

$$s_{opt} = 3.0596\ L\ \mathrm{Ra}_L^{-0.236} \tag{12}$$

and

$$q_{conv} = q_{conv,flat} + 0.1898\ \mathrm{Ra}_L^{0.51} k H \Delta T \frac{W}{L} \tag{13}$$

These two sets of correlations are also used in the thermal analyses to obtain more reliable convective heat transfer rate estimates than the conservative ones from (5).

C. Radiative Cooling

The contribution to heat dissipation by radiative heat transfer is considerable for both flat plate heat sink and plate finned heat sink geometries. In radiative heat transfer rate calculations, the temperature of the surroundings is taken as equal to the ambient air temperature. This approximation is perfectly valid for well insulated indoor spaces without any nearby high/low temperature radiant heat sources/sinks.

For the flat plate geometry, radiative heat transfer rate is given with:

$$q_{rad,flat} = \sigma\varepsilon A(T_s^4 - T_\infty^4) \tag{14}$$

where σ is Stefan-Boltzmann constant, ε is the emissivity of the surface, A is the surface area and T_s and T_∞ are the temperatures of the surface and the surroundings.

For the finned heat sink geometry, a view factor for the composite of the fins and unfinned areas can be calculated by using view factor equations from the view factor catalogue in [5]. Specifically, "Perpendicular rectangles with a common edge" is used to find the view factor between the unfinned base and the fin side surface, $F_{b\text{-}s}$. "Alligned parallel rectangles with a separation distance" is used for finding the view factors between the fin side surfaces, $F_{s\text{-}s}$. The view factor between the tip of the fin and the surroundings is 1. By using these view factors, with the help of summation rule and reciprocity, the view factor between the composite heat sink surface and the surroundings, $F_{hs\text{-}surr}$ can be obtained from:

$$F_{hs-surr} = \frac{2A_{side}\left(1-\frac{A_{base}}{A_{side}}F_{b-s}-F_{s-s}\right)+A_{base}(1-2F_{b-s})+A_{tip}}{2A_{side}+A_{base}+A_{tip}} = \frac{2H\left(1-F_{s-s}\right)+s(1-4F_{b-s})+t}{2H+s+t} \quad (15)$$

where A_{side}, A_{base} and A_{tip} are the fin side, unfinned base and fin tip areas of the heat sink. Equation (15) is used for the inner parts of the heat sink. The view factor between the outside facing sides of the end plates of the heat sink and the surroundings is 1. Therefore, the radiative heat transfer rate from the heat sink to the surroundings, $q_{rad,finned}$ can be obtained from:

$$q_{rad,finned} = (N_f-1)(2A_{side}+A_{base}+A_{tip})\sigma\varepsilon F_{hs-surr}(T_b^4-T_\infty^4) + \sigma\varepsilon(2A_{side}+A_{tip})(T_b^4-T_\infty^4) \quad (16)$$

D. Heat Transfer Rate Calculations

The total heat transfer rates for both bare flat and finned heat sink geometries can be calculated by summing the convection and radiation heat transfer rates. As the aspect ratio, $1:\sqrt{2}$ is considered for being the typical aspect ratio for flat panel displays.

The heat transfer calculation results for the flat bare heat sink geometry are presented in TABLE I. For each considered vertical length L, Ra_L and Nu_L are obtained from (2) and (1), respectively. The convective heat transfer rates, $q_{conv,flat}$ are calculated using (3). The total heat transfer rate, q_{flat} values are obtained by adding radiative transfer rate, $q_{rad,flat}$ that is calculated from (14) to $q_{conv,flat}$. For the lowest L value, the contribution of radiative transfer to total heat transfer rate is about 50% and it increases with increasing L due to the decreasing convection heat transfer coefficient with the increasing boundary layer thickness with L.

It is observed that the total heat transfer rates that can be obtained with the flat bare heat sink are only sufficient for integrated components such as low voltage mobile CPUs and low power graphics chipsets. For more powerful integrated components, finned heat sinks are required.

TABLE I
NATURAL CONVECTION HEAT TRANSFER RATES WITH OR WITHOUT RADIATION FOR THE FLAT HEAT SINK GEOMETRY

L (m)	W (m)	Ra_L	$q_{conv,flat}$ (W)	q_{flat} (W)
0.200	0.283	2.0989E+07	7.38	15.07
0.250	0.354	4.0994E+07	11.21	23.23
0.300	0.424	7.0838E+07	15.80	33.11
0.350	0.495	1.1249E+08	21.15	44.71
0.400	0.566	1.6791E+08	27.26	58.03
0.450	0.636	2.3908E+08	34.11	73.06
0.500	0.707	3.2796E+08	41.71	89.80
0.550	0.778	4.3651E+08	50.06	108.24
0.600	0.849	5.6671E+08	59.15	128.39
0.650	0.919	7.2052E+08	68.97	150.24
0.700	0.990	8.9991E+08	79.54	173.79
0.750	1.061	1.1068E+09	90.84	199.03
0.800	1.131	1.3433E+09	102.88	225.97
0.850	1.202	1.6112E+09	115.64	254.61
0.900	1.273	1.9126E+09	129.14	284.94
0.950	1.344	2.2494E+09	143.38	316.97
1.000	1.414	2.6236E+09	158.34	350.68

If the surface emissivity is low and the radiative transfer is negligible, the contribution of the fins is essential for sufficient heat dissipation. The convection heat transfer rate results for the plate finned heat sink geometry for the same vertical length values are presented in Fig. 4 and Fig. 5 for fin heights of 0.01 m and 0.005 m, respectively. Here, s values are obtained from (4), (10) and (12). The minimum (shown as "Ref [6]") and the maximum ("Flat limit") convection heat transfer rates are calculated by using (5) and (7), respectively. The other two sets of results, "Ref [7]" and "Ref [8]" are obtained from (11) and (13).

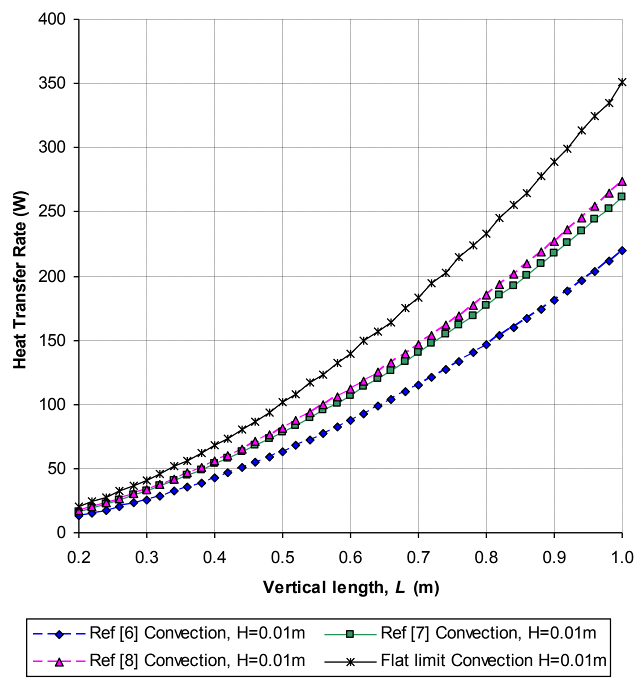

Fig. 4. Natural convection heat transfer rates without radiative transfer for the fin height, *H*=0.01 m.

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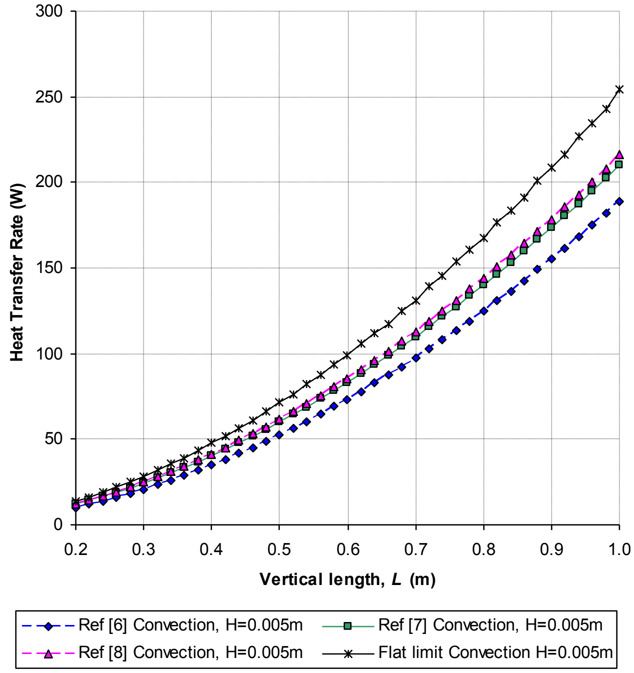

Fig. 5. Natural convection heat transfer rates without radiative transfer for the fin height, *H*=0.005 m.

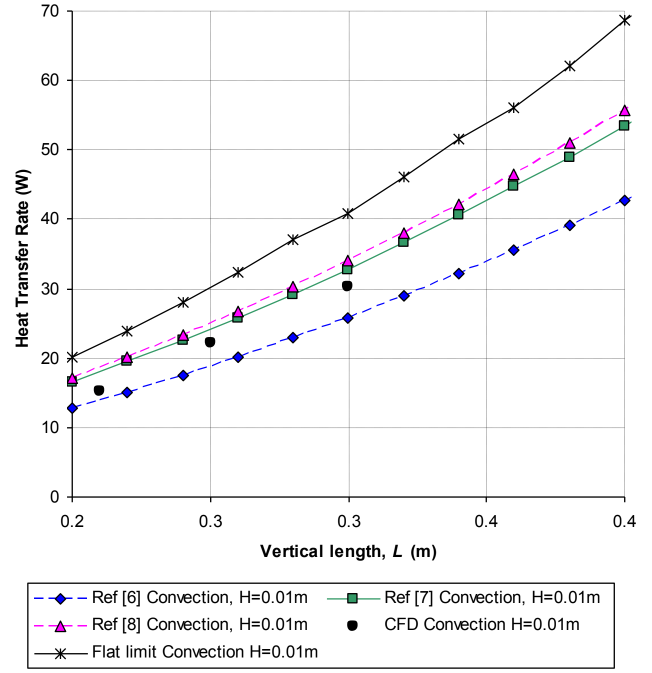

Fig. 6. Natural convection heat transfer rates without radiative transfer for the fin height, *H*=0.01 m, comparison of 3 data points obtained from CFD simulations with the results of correlations.

In order to show that the actual convection behavior should be between the maxima ("Flat limit" results) and the minima that are obtained from (5), three CFD results for H=0.01 m case are presented with black circles at L= 0.21, 0.25 and 0.3 m in Fig. 6. These simulation results are obtained using ANSYS Fluent CFD software with the exact geometries and boundary conditions of the heat sinks, after defining them on the center of a vertical wall in an air filled cube of 3 m sides, by meshing the domain with approximately 3 million cells that are fine in the vicinity of the heat sink and coarse away from it. All of the CFD results are between the minima and the maxima and they are close to "Ref [7]" and "Ref [8]" results. Thus these two sets of result can be considered as more realistic.

It is observed form Fig. 4 and Fig. 5 that heat sinks with fins of 0.005 m height perform sufficiently by providing low to moderate dissipation rates for the convection only case, reaching up to 210 W for the largest panel size of 1 m × 1.414 m. Fins of 0.01 m perform even better, reaching up to 260 W for the largest panel size.

The calculation details for the finned heat sink radiative and total heat transfer rates that are obtained by using the correlations from [6] are given in TABLE II. Here, N_f, $F_{hs\text{-}surr}$ and $q_{rad,finned}$ values are calculated from (8), (15) and (16), respectively. $F_{hs\text{-}surr}$ is increasing with increasing L due to increasing separation distance between the fins. The radiative heat transfer rate is calculated by assuming surface emissivity of 0.8 for the anodized (oxidized) aluminum. The last two columns are the minimum ("Ref [6]") and the maximum ("Flat limit") total heat transfer rates for each L. It is observed from these ranges that with the contribution from radiative transfer, it is even possible to cool powerful desktop computer components integrated into a flat panel display.

TABLE II
RADIATIVE AND TOTAL HEAT TRANSFER RATES

		$F_{hs\ surr}$		$q_{rad,finned}$ (W)		$q_{finned,min}$ (W)		$q_{finned,max}$ (W)	
L (m)	**N_f**	**0.005**	**0.01**	**0.005**	**0.01**	**0.005**	**0.01**	**0.005**	**0.01**
0.20	25	0.542	0.380	7.9	8.4	18.0	21.2	21.6	28.6
0.25	30	0.553	0.389	12.4	13.0	27.4	31.8	33.0	43.1
0.30	34	0.562	0.397	17.5	18.2	38.3	44.1	45.6	59.0
0.35	39	0.570	0.403	24.2	25.1	51.7	58.9	62.0	79.6
0.40	43	0.577	0.410	31.4	32.4	66.4	75.2	79.3	101.0

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0.45	47	0.583	0.415	39.7	40.8	83.0	93.4	98.9	125.2
0.50	51	0.589	0.421	49.0	50.2	101.5	113.6	120.8	152.1
0.55	54	0.594	0.425	58.3	59.6	120.9	134.7	142.4	178.4
0.60	58	0.598	0.430	69.7	71.1	143.1	158.7	168.7	210.6
0.65	62	0.603	0.434	82.2	83.8	167.2	184.8	197.5	245.6
0.70	65	0.607	0.438	94.4	96.1	191.9	211.5	225.2	279.1
0.75	69	0.610	0.441	109.1	110.9	219.8	241.5	258.7	319.5
0.80	72	0.614	0.445	123.3	125.1	248.0	271.8	290.5	357.9
0.85	76	0.617	0.448	140.2	142.2	279.8	305.8	328.7	403.8
0.90	79	0.620	0.451	156.4	158.5	311.7	339.9	364.8	447.0
0.95	82	0.623	0.454	173.5	175.7	345.2	375.8	402.9	492.6
1.00	86	0.626	0.457	193.9	196.2	382.8	415.7	448.2	546.8

The comparison of total heat transfer rates for both of the considered fin heights are presented in Fig. 7 and Fig. 8. Each separate data set is obtained by using its own optimum fin separation distance whereas the "Flat limit" data set is calculated for s_{opt} from (4). Here, the data points are getting farther apart with the increasing L due to the increasing contribution from the radiative transfer.

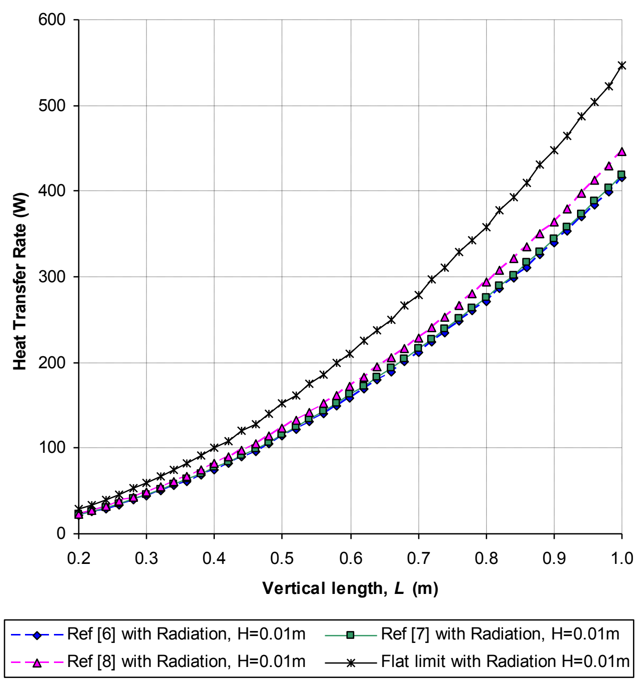

Fig. 7. Passive heat transfer rates (natural convection and radiative transfer) for the fin height, *H*=0.01 m.

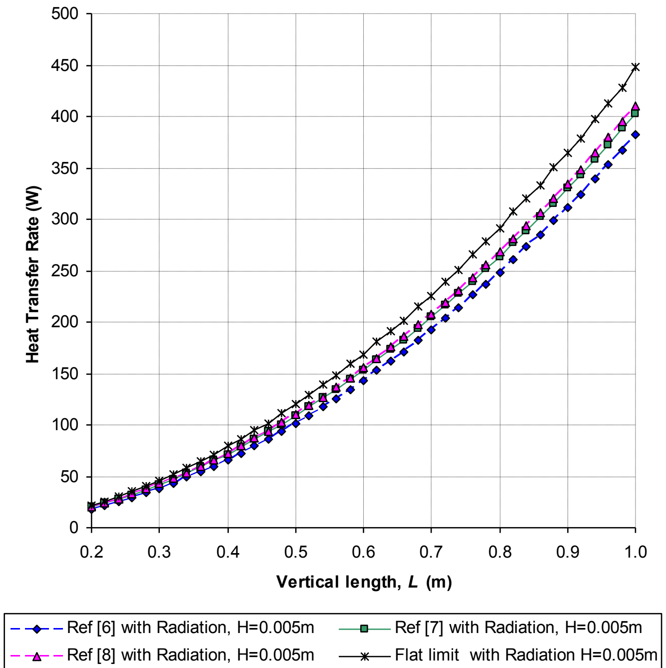

Fig. 8. Passive heat transfer rates (natural convection and radiative transfer) for the fin height, *H*=0.005 m.

When "Ref [7]" and "Ref [8]" results are taken as representatives of actual behavior in H=0.01 m case, it is observed that 0.01 m fin height does not have much advantage compared to 0.005 m fin height, due to the higher view factors of 0.005 m fins (see TABLE II $F_{hs\text{-}surr}$ results). For the largest panel size, the heat sink with 0.01 m fins can dissipate 445 W whereas the one with 0.005 m fins can dissipate 410 W.

IV. Discussions and Other Considerations

The thermal analysis in Section III is valid with the following limitations:

- The analysis is for steady state, therefore there should not be a transient surpassing the calculated power budget (Table I and III heat transfer rates).
- Uniform 50°C heat sink surface temperature is assumed. If heat spreaders and heat pipes beneath the heat sink are designed to keep the hot and cold spot temperature difference low, maintaining a 50°C average should be enough to make the analysis valid.

• Both ambient air and surroundings temperatures are taken as 25°C giving a base-to-ambient temperature of 25°C. On a very hot day, it is still possible to keep a component such as a mobile computer CPU below its T_{max} which is 99°C.

• The analysis is only for aluminum heat sinks with anodized surfaces. The assumed anodized aluminum emissivity of 0.8 is an achievable value, however degrading surface conditions and dust accumulation may reduce radiative heat transfer from the surface.

Overall, it is observed that the proposed passive cooling solutions are feasible for the thermal management of flat panel displays with integrated processors.

Heat dissipation from the panel components is not uniform. Since the thermal conductivity of aluminum does not show considerable directional difference, it is not possible to make the heat sink thick enough for even spread of heat. However, the proposed heat spreader flat heat pipes can make the spread uniform [4]. The sizes of flat heat pipes are limited due to manufacturing constraints and due to the fact that for large flat heat pipes, the thickness can be prohibitively large, therefore small heat pipe sections should be organized to form a complete heat transfer pathway between the components and the heat sink.

According to Yazicioğlu and Yüncü [6] "For a given fin length and base-to-ambient temperature difference, the values for optimum fin spacing do not vary more than an amount of 0.1 mm i.e. the optimum fin spacing is almost insensitive to the variations in fin height. The dependence of optimum fin spacing on base-to-ambient temperature difference is not very strong. For a given fin height and fin length, the values for optimum fin spacing do not vary more than an amount of 0.4 mm." Therefore, the proposed finned heat sink design can be used at various base-to-ambient temperature differences. Also, the fin height can be varied with a little effect on the fin separation.

V. Conclusion

Thermal analyses of the proposed passive cooling assembly for flat panel displays are performed considering a wide range of panel sizes with the aspect ratio of $1:\sqrt{2}$. The passive heat transfer rate limits are calculated for two different heat sink geometries, i.e. a flat vertical surface and plate finned heat sinks. It is observed that even these simple heat sink geometries provide enough passive heat dissipation for a wide range of applications requiring integrated high power components.

The proposed passive cooling assembly may especially be useful in 3D TV applications that require powerful processors to process 3D data or to convert 2D video to 3D. Early examples of such applications, among others, are presented in [9] and [10].

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