Impact of TXV and Compressor in the Stability of a High-End Computer Refrigeration System

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The combination of increased power dissipation and increased packaging density has led to substantial increases in chip and module heat flux in high-end computers. The challenge is to limit the rise in chip temperature above the ambient. In the past, virtually all commercial computers were designed to operate at temperatures above the ambient. However, researchers have identified the advantages of operating electronics at low temperatures. Until recently, large-scale scientific computers used direct immersion cooling of single-chip modules. The current research focuses on mainframes (computer system), which uses a conventional refrigeration system to maintain chip temperatures below that of comparable air-cooled systems, but well above cryogenic temperatures. Multivariable control of compressor speed along with thermostatic expansion valve (TXV) opening can give better stability and performance. TXV is a mechanical controlling device used in the refrigeration system. The compressor is the only mechanical-working component in the refrigeration cycle that circulates refrigerant through the system continuously. Hence, controlling the compressor is an important aspect. The control objective is defined as improving the transient behavior of the vapor compression cycle for the refrigeration system operating around an evaporator set-point temperature. The system behavior is studied in two cases, TXV being the only control element in the first case, while TXV and a compressor both act as control elements in the other case. [DOI: 10.1115/1.1827272]

Introduction

With the ever-increasing power density in CMOS chip technologies; cooling has now become a strong influence on computer performance [1]. It is known that, mobility that is a ratio of electron or hole velocities to electric field is a function of temperature. Mobility increases as temperature decreases due to a reduction of carrier scattering from thermal vibrations of the semiconductor crystal lattice. Also, there is an exponential reduction in leakage currents. Depending on the doping characteristics of the chip, performance improvement ranges from 1% to 3% for every 10°C lower transistor temperature that can be realized.

Researchers have always known the advantages of operating electronics at low temperatures. This facilitates faster switching time of semiconductor devices, increased circuit speeds due to lower electrical resistance of interconnecting materials, and reduction in thermally induced failures of devices and components. Over the past several years, computer systems have been shipped with refrigeration, utilizing the vapor compression cycle to attain low temperatures. A typical evaporator temperature ranges from −40°C to 20°C. Also, the second major reason for designing a system with low-temperature cooling is the improvement achieved in reliability to counteract detrimental effects, which rises as technology is pushed to the extremes. Many wear-out failure mechanisms follow the Arrhenius equation [2], showing that for die temperatures operating in the range of −20°C to 140°C, and every 10°C decrease in temperature reduces the failure rate by approximately a factor of 2. Therefore, a significant improvement could be achieved in chip failure rates with lower temperatures achievable through electronic cooling.

With the increase in CMOS performance achieved with lower temperatures, a number of companies have embarked on programs to investigate cooling electronics, some evolving to major product announcements and shippable products. Intel, DEC, AMD, Sun Microsystems, IBM, SYS Technologies, and Kryotech, Inc., have all shown computers utilizing the vapor compression refrigeration cycle.

The IBM S/390 G4 CMOS system [1], first shipped in 1997, was the first such IBM design to employ refrigeration cooling. Bulk power for the system, at the top of the frame, distributes 350 VDC throughout the frame. Below the bulk power is the central electronic complex (CEC) where the multichip module (MCM), housing 12 processors, is located. Various electronic "book" packages (memory, control modules, DC power supplies, etc.) are mounted on each side of the processor module. Below the CEC are blowers that provide air cooling for all the components in the CEC except the processor module, which is cooled through refrigeration. Below the blowers are two modular refrigeration units (MRUs), which provide cooling via the evaporator mounted on the processor module. Only one MRU at a time runs during normal operation. Should one MRU fail, it can be replaced via quick connects located at the evaporator. Thus, a new MRU can be installed while the system continues to operate. The evaporator mounted on the processor module is redundant in that two independent loops utilizing copper tubes are interleaved through a thick copper plate, each loop attached to separate MRUs. Refrigerant passing through one loop is adequate to cool the MCM (which dissipates a maximum power of 1050 Watts for G4) under all environmental extremes allowed by the system. In the bottom of the frame, the I/O electronic books are installed along with the associated blowers to provide the air cooling. Air cooling for the condenser located in the MRUs is provided by air exiting the I/O cage at the bottom of the frame. Airflow through the condenser as well as through the I/O cage is increased for room temperatures above 27°C.

Evaporator Time Constant

A considerable amount of work has been done to investigate the hunting phenomena. Wedekind and Stoecker [3] formulated a the-

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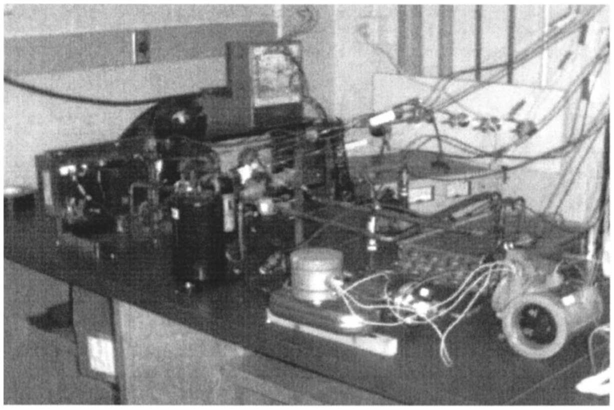


Fig. 1 Experimental set up

Valve	Bulb Size (OD x L)	Time Constant (sec)
JC	9.5mm x 75mm	6.58
JCP60	12.75mm x 76mm	13.54
JZPM	19mm x 50mm	19.2

Fig. 2 Evaporator time constants for different bulb sizes

oretical model to predict the mean transient response of the mixture vapor transition point under sudden change in refrigerant mass flow rate. Two equations have been derived, one describing the mixture vapor behavior under step decrease in refrigerant inlet mass flow rate and the other describing the step increase [4]. The time constant associated with step decrease is the time required to evaporate the entire excess refrigerant under the assumption that the rate of heat transfer to the excess refrigerant remains constant. The time constant associated with step increase is defined as the time required to overcome the refrigerant shortage provided the rate of inlet mass remains constant. Time constant was reported to be on the order of 6 s. Danning reported an evaporator time constant of 3.9 min [5].

Kulkarni [6] presented a mathematical model in which he derived an equation for an evaporator time constant. The equation was derived on the assumption that evaporator and thermostatic expansion valve are dynamic, neglecting the condenser and compressor.

In the current research, the evaporator time constant τ was calculated (as shown in Fig. 2) for three different thermostatic expansion valves (TXVs), namely, EQ-2-JC, EQ-2-JCP60, and EQ-2-JZPM. The experimental results for this calculated evaporator time was then found to be very close. The effect of using two control elements, the TXV, and the compressor on the response of the system was studied.

Vapor Compression System

Refrigeration is the withdrawal of heat from a substance or space so temperatures lower than that of the natural surroundings are achieved. A vapor compression cycle is shown in Fig. 3(*a*) with its P-h curve in Fig. 3(*b*). Vapor compression systems are employed in most refrigeration systems. Here cooling is accomplished by evaporation of liquid refrigerant under reduced temperature and pressure. The fluid enters the compressor at state 1 where the temperature is elevated by mechanical compression (state 2). The vapor condenses at this pressure, and the resultant heat is dissipated to the surrounding. The high-pressure liquid (state 3) then passes through the expansion valve through which the fluid pressure is lowered because of the expansion process. Here the temperature along with pressure reduces significantly. The low-pressure fluid enters the evaporator at state 4 where it evaporates by absorbing heat from the refrigerated space and reenters the compressor. The whole cycle is repeated [7]. P-h variations can be studied for this system in Fig. 3(*b*), where all the states are mentioned.

Bulb Location

The function of the sensing bulb is to measure refrigerant vapor temperature at the evaporator outlet. To sense this temperature, the bulb must not be influenced by oil in the system or by sources of external heat [8]. The sensing bulb should be located on the horizontal section of the line near the evaporator outlet. If the suction line rises above the sensing bulb location, a trap should be installed immediately downstream of the bulb so that the oil does not collect at the bulb location. The bulb must not be installed on a trap or downstream of trap. It should not be installed on the bottom of the suction line due to the possible presence of oil. Installation of the sensing bulb on the vertical section of the suction line should be avoided because the bulb is more likely to be influenced by oil. If the bulb must be located on the vertical line, then experiments must be performed to determine the best location for proper system performance.

Effect of Masterflux Compressor

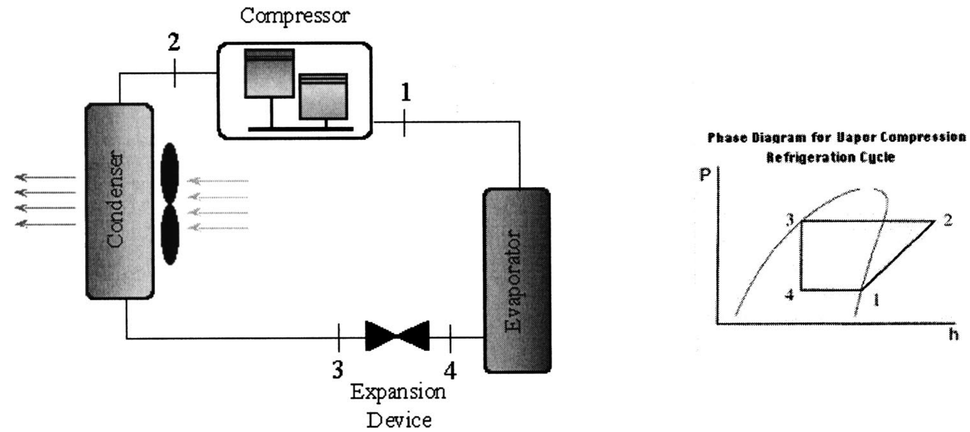
In previous work, Kulkarni [9] determined the optimum bulb location in order to reduce hunting and improve the stability of the system. His work was further continued by Mulay [10], who studied the effect of different thermophysical properties of the bulb mentioned previously.

In both the studies, however, a thermostatic expansion valve was the only control element adjusting the refrigerant quantity passing through the evaporator upon receiving the feedback about change in heat load while the typical compressor used was running at constant speed. The Masterflux compressor was introduced in the system so as to have a multivariable control that will enable one to regulate the refrigerant quantity by not only varying the TXV opening, but also by varying the compressor speed. The data are then analyzed for both setups.

Experimental Bench

Figure 1 shows the experimental bench. Compressor, condenser, thermostatic expansion valve and evaporator, accumulator, and hot gas bypass valve are the main components shown in the figure. Sensor bulb is placed on the evaporator return line. The temperature and pressure sensors are mounted at various places in the system. A flow meter is also placed to measure the flow rate. The cold plate attached to the single plate of evaporator is made up of copper. The heater block is mounted on this copper plate. This heater block, which simulates the multichip module, has resistive heaters embedded in it that can provide a heat load of more than 1 KW. This heat load can be varied from zero to full scale. At the interface of the heater block and the evaporator cold plate, thermal paste is applied to minimize the contact resistance and ensure the proper heat transfer. The schematic layout is presented in Fig. 4.

As stated earlier, the system is operated in two different configurations. The evaporator exit temperature is sensed by the sensor bulb of TXV. If this temperature is higher than the evaporating temperature, it will result in increasing the TXV opening, thereby



(*a*) (*b*)

Fig. 3 Vapor Compression Cycle (*a*) with its P-h diagram (*b*) [1]

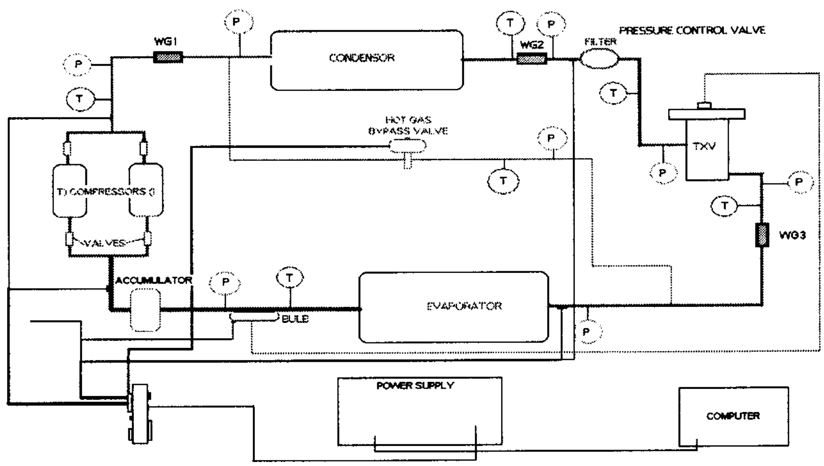


Fig. 4 Schematic of the current system

allowing more refrigerant to pass through. If the evaporator exit temperature is lower than the evaporating temperature, the TXV opening will decrease, restricting the refrigerant flow. This is the configuration where only TXV acts as the control element.

The other configuration consists of a Masterflux compressor with a controller unit. A temperature sensor senses the evaporator exit temperature at the same location as that of TXV sensor bulb. If the evaporator exit temperature is higher than the evaporating temperature, then the signal from temperature sensor will cause an increase in compressor speed while the TXV sensor bulb signal will cause a bigger opening, thus, effectively increasing the refrigerant flow. An evaporator exit temperature lower than the evaporating temperature will result in a reduction of both the compressor speed and the TXV opening, thus, reducing the refrigerant flow.

Results

The system response was measured in terms of interface, superheat, and evaporator outlet temperatures. Results using the Masterflux compressor are shown in Figs. 4–6. These data were obtained at different loads starting from no load to full load (1000 W) at an interval of 250 W.

The evaporator time constants for different TXVs were calculated and were checked with experimental values. The values are shown in Fig. 2.

Interface results show an increase in temperature as we go from no load to full load. For no load conditions, the average temperature range (ATR) is about 12°C. Similarly, for the 250 W load, the ATR is 15°C; for the 500 W load, the ATR is 18°C; for the 750 W load, the ATR is 22°C; and for the 1000 W load, the ATR is 25°C. This can be seen in Fig. 5. This gives a smooth increase in temperature as we go on increasing the heat load. This shows that for

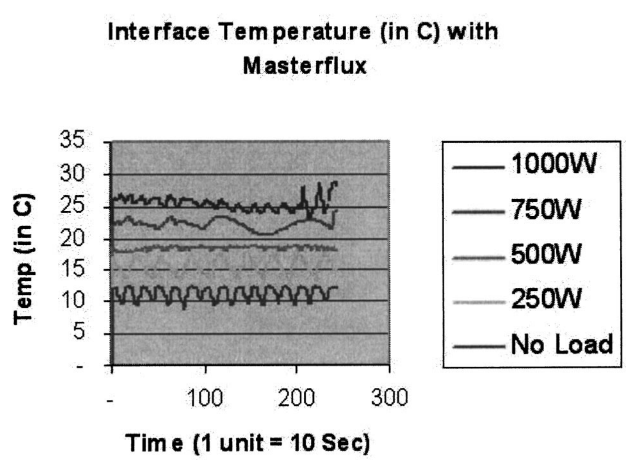


Fig. 5 Interface temperature with and without Masterflux compressor at different loads

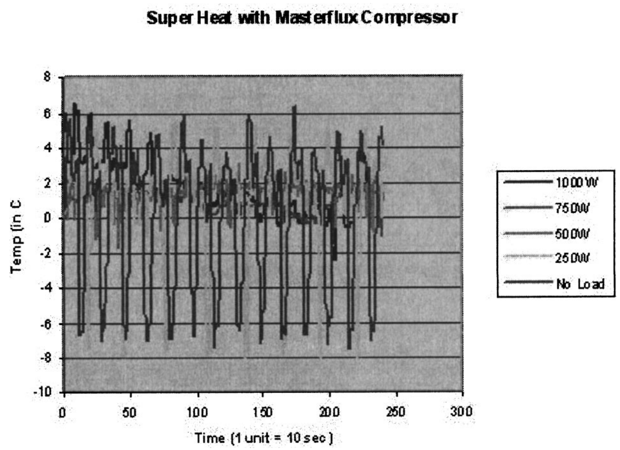


Fig. 6 Superheat outlet temperature with and without Masterflux compressor at different loads

every 250 W increase in load, the temperature roughly increases by 3°C. With this we can roughly estimate the temperature at higher loads.

As load increases from no-load to full-load (1 KW) condition, the fluctuations are reduced notably. This can be seen in Fig. 6.

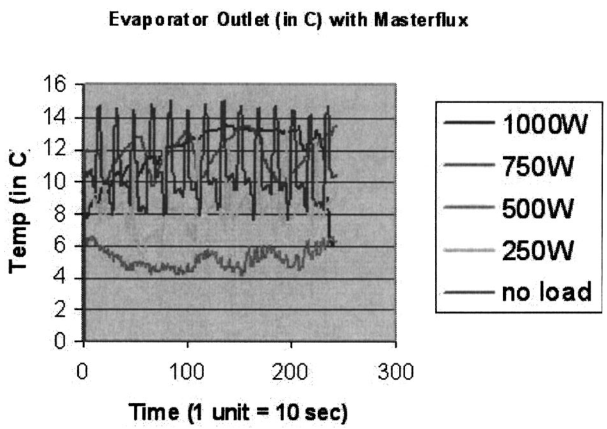


Fig. 7 Evaporator outlet temperature with and without Masterflux compressor at different loads

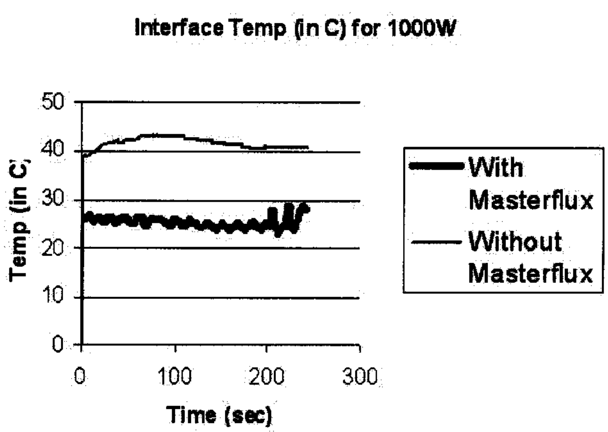


Fig. 8 Interface temperature with and without Masterflux compressor at 1000 W load

Thus, it can be seen that system performance is better at full load. The evaporator data as seen in Fig. 7 again show that fluctuations decrease with the increase in load.

Figures 8–17 show a comparison of the performance of the typical commercial compressor versus the Masterflux compressor. Figures 8 and 9 show that difference in the interface temperature in two cases is very significant (about 20°C). The results show that stability has been improved greatly using the Masterflux compressor. Also, there is a significant decrease in temperature when the Masterflux compressor is used.

As the temperature decreases significantly we can say that the flow rate should have improved for the same size unit. Flow is a very important parameter for the refrigeration unit for both deal-

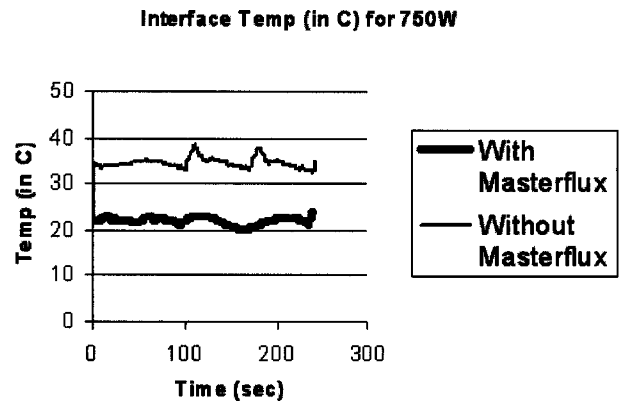


Fig. 9 Interface temperature with and without Masterflux compressor at 750 W load

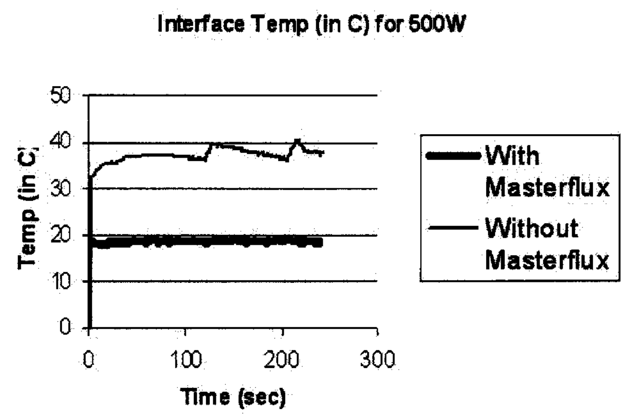


Fig. 10 Interface temperature with and without Masterflux compressor at 500 W load

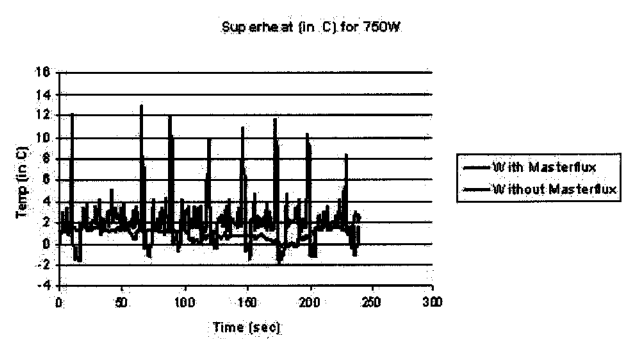


Fig. 11 Superheat temperature with and without Masterflux compressor at 750 W load

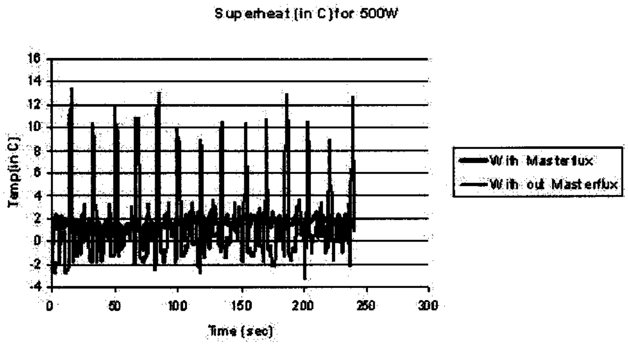


Fig. 12 Superheat temperature with and without Masterflux compressor at 500 W load

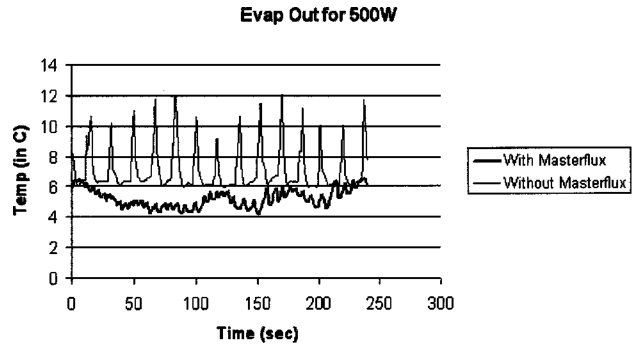


Fig. 16 Evaporator outlet temperature with and without Masterflux compressor at 500 W load

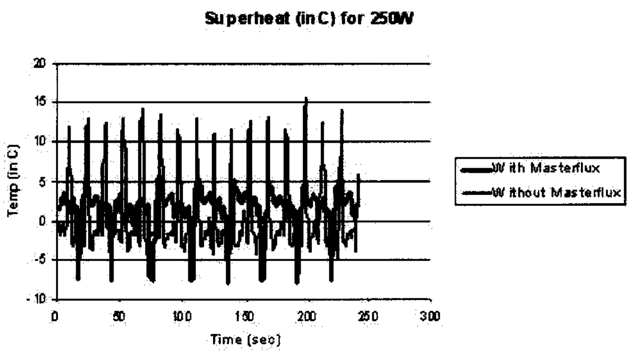


Fig. 13 Superheat temperature with and without Masterflux compressor at 250 load

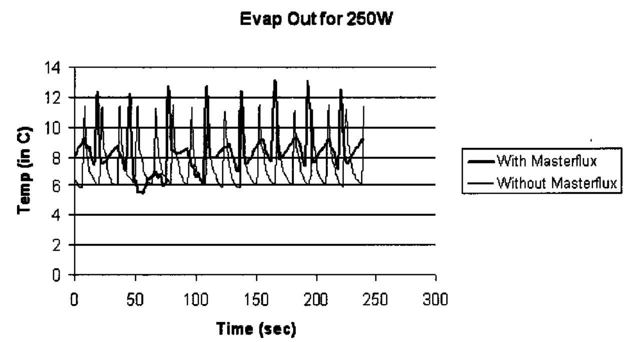


Fig. 17 Evaporator outlet temperature with and without Masterflux at 250 W load

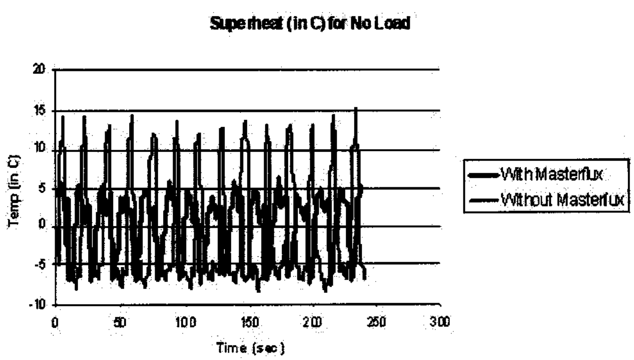


Fig. 14 Superheat temperature with and without Masterflux compressor at no load

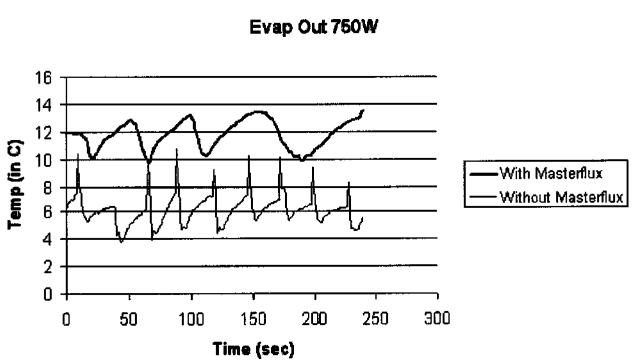


Fig. 15 Evaporator outlet temperature with and without Masterflux compressor at 750 W load

ing with the temperature fluctuation and controlling the temperature set point. Improved flow means better coordination between TXV and the compressor (the two control systems).

Another important result that can be seen is that the average value of superheat in all the different load cases remains almost at 2°C superheat. This is not true for the case without Masterflux. Evaporator results were taken just to double-check the conditions reflected from interface temperature, and the superheat temperature are true. The evaporator results are shown in Figs. 16 and 17. The size of the Masterflux compressor is almost half the size of the existing compressor.

Conclusion

The experimental data indicate better system response and transient behavior when a two-feedback system is used as compared to a single-feedback system. In a single-feedback system, the thermal resistance between the bulb and evaporator return line can considerably affect system stability, and by increasing this thermal resistance, the stability can be further improved. Similarly the size of the bulb and the two-phase heat transfer coefficient have an effect on system stability. By using a feedback signal to adjust the compressor speed, the effect of thermal resistance between the sensor bulb and evaporator return line can be minimized.

Another big advantage for system architecture is that by using Masterflux compressor, considerable reduction in volume of the cooling system is achieved. The Masterflux compressor has essentially the same footprint, but is almost 25% shorter than a typical commercial compressor.

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