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Strategies for Improving Temperature Control and Energy Use in Household Refrigerators

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ABSTRACT

Conventional control of household refrigerators is typically accomplished with simple, single-sensor control systems. Though cost-effective this approach allows temperature dependencies to exist between compartments, producing non-optimal energy consumption. In order to study the performance of the conventional refrigerator a computer model was developed. Analytical results were generated to determine transient performance of the plant while operating under various control approaches. The model consists of refrigerant and airflow subsystems to represent the plant's refrigerant system thermodynamics and air flow control algorithm. Improved performance can be obtained by adding "dual" control components and using nonconventional control strategies. Specifically, variable temperature bandwidths and uncoupled compressor and evaporator fan algorithms were examined. Also, plant configurations using automatic air flow dampers to regulate the refrigerator fresh food temperature, with separate thermostatic control of the freezer were examined. Two types of air flow dampers were considered: the open/closed (snap-action) type and the proportional (creep) damper. This study indicates that use of automatic dampers in combination with these nonconventional strategies can provide near optimal control at *all* plant operating points.

INTRODUCTION

An air flow diagram for a conventional refrigerator is shown in Fig. 1. Typical temperature control uses two basic features: (i) a compressor on/off thermostatic sensor located in the fresh food compartment, and (ii) a manually-adjusted damper (D in Fig. 1) regulating air flow between the freezer and fresh food compartments. The damper controls freezer temperature by regulating the flow of cold air to the fresh food compartment. The air flow is generated by the evaporator fan which cycles simultaneously with the compressor. Typically, the compressor and evaporator fan are turned on and off by a thermostatic sensor with a constant (i.e., fixed) temperature bandwidth. The set point is manually adjustable but the bandwidth remains constant. This single control approach permits coarse control of the freezer but relatively good fresh food temperature control.

The advantages of conventional control lie in its simplicity and domestic market-acceptance. Drawbacks include temperature dependency between compartments, temperature shifts between ambients, limited control range, and energy inefficiency. To improve on conventional refrigerator performance, approaches were considered which added degrees of freedom to the control problem using non-traditional control algorithms. Four strategies were studied: constant and variable temperature bandwidths with compressor and evaporator fan coupled and uncoupled. These four strategies were also investigated using two types of automatic air flow dampers: "snap action" and, "creep" type dampers. When "uncoupled" the evaporator fan operates whenever the compressor runs but it also can run without the compressor. "Variable temperature bandwidths" refers to the use of sensor limits which can be set uniquely for each plant control point.

THE PLANT MODEL

A steady state thermodynamic cycle model developed by Jaster [1] was extended by Bessler [2] to include transients and the air flow analysis. It consists of refrigerant and air flow system submodels which compute the states of both working fluids as they circulate around their respective plant subsystems. The control volumes useful for formulating the one dimensional energy relationships required for computing the air temperatures inside the plant are shown in Fig. 1. The applicable compartment "energy equations" are given below. In these equations the subscripts E, Z, F refer to evaporator, freezer, and fresh food control volumes. The discharge air properties are the bulk (or average) values for the compartments.

$$-\dot{Q}_E + \dot{Q}_{fan} = c_p(\dot{m}_Z + \dot{m}_F)T_E - c_p\dot{m}_Z T_Z - c_p\dot{m}_F T_F + dU_E/dt \tag{1}$$

$$\dot{Q}_Z + \dot{Q}_{ZF} + \dot{Q}_{ZH} = c_p\dot{m}_Z T_Z - c_p\dot{m}_Z T_E + dU_Z/dt \tag{2}$$

$$\dot{Q}_F - \dot{Q}_{ZF} + \dot{Q}_{FH} = c_p\dot{m}_F T_F - c_p\dot{m}_F T_E + dU_F/dt \tag{3}$$

Enthalpies are written in tems of temperature, using the ideal gas assumption for the air, as: $\Delta h = c_p \Delta T$, where c_p is the specific heat at constant pressure for the air. The left sides are the heat transfer rates into the control volumes where subscript ZF refers to the transfer from fresh food to freezer. The terms with subscripts ZH , FH are heat generated by heating elements in the mullion and the defrost system. The $\dot{m}$ are air mass flow rates in the indicated compartments. The right sides are the energy convected with the air flow across the boundaries and the time derivatives of the internal energy of all the air and thermal mass within the control volume boundaries. Since the evaporator has no such thermal mass, and the heat capacity of the air alone is negligible compared to that of the freezer and fresh food compartments so that the term $dU_E/dt \cong 0$. The ambient temperature T_0 and compartment temperatures (T_E ,T_Z , T_F) are used to compute heat leakage through freezer and refrigerator walls, doors, and divider material (mullion) by solution of the heat flows from heat conduction: $\dot{Q}_Z = U_Z(T_0 - T_Z)$, etc. Values of the conductance U_Z, etc. and the heater elements producing $\dot{Q}_{ZH}$ and $\dot{Q}_{FH}$ are furnished from user input data. Also, the air flow rates ($\dot{m}_Z$, $\dot{m}_F$) are specified from user input, or calculated depending on the type of air flow damper and setting. $\dot{Q}_E$ is obtained from the systems submodel and T_E is determined from eqn (1), and dU_Z/dt and dU_F/dt from eqn's (2)-(3). The bulk compartment air temperature variations are determined from the dU/dt and the corresponding derivatives:

$$dT_Z / dt = (1/C_Z)\, dU_Z / dt \qquad \text{and} \qquad dT_F / dt = (1/C_F)\, dU_F / dt \tag{4}$$

where the heat capacities (or thermal inertia's) C_Z and C_F are determined from: $C = m_{air}\, c_v + m_x\, c_x$ and c_v, c_x and m_{air}, m_x are specific heats and food mass of the air and thermal mass in the control volume. Typical values for household refrigerators produce values of $m_x\, c_x >> m_{air}\, c_v$ so that the heat capacity of the air is negligible compared to the thermal loads. Values for the C_Z and C_F are provided from user input. With user provided initial conditions on T_Z, and T_F , eqn's (4) produce the temperature derivatives. A straight forward finite difference was used for the integrations. The entire model developed for household refrigerator performance analysis relies on over 100 inputs to describe the physical and thermal features of the plant, and control system. Control logic is also included so that algorithms based on control of air flow, temperature bandwidth, and component coupling can be represented. Validation has been obtained in a limited number of comparisons with experimental test data.

TRADITIONAL TEMPERATURE CONTROL PERFORMANCE

The compressor thermostat set point temperatures are prescribed as warm, mid and cold. Desired temperatures for freezer/fresh food settings are: warm (2/41°F), mid (-2/37°F), and cold (-6/33°F). The damper setting strongly influences the actual temperatures achieved and general performance of the plant. To study the effectiveness of conventional control, the limit settings were chosen for computer simulation. Temperature control windows computed with the traditional single control system, for two ambient temperatures (70 and 90°F), are plotted in Fig. 2. Each point on the figure is computed from the compartment temperatures averaged over the compressor duty cycle for specific thermostat and damper settings. The compressor is switched on and off by the fresh food thermostat using fixed bandwidth limits of ±4°F of the set point temperature.

In Fig. 2, the two leftmost fresh food temperatures [T_F(average)] are obtained with the compressor/fan thermostat at the cold set point, while the two rightmost fresh food temperatures result from the warm thermostat set point. The two lowest freezer temperatures [T_Z(average)] result from setting the damper to "cold" (fully closed for minimum air flow to the fresh foods), while the two highest freezer temperatures are produced by the damper "warm" setting (fully open to maximize air flow to the fresh foods). An ideal plant controller would produce a rectangular temperature control window. The amount of temperature dependence between compartments is indicated by the deviation of the window from rectangular. From Fig. 2, it is evident that traditional plant temperature control is not ideal as it produces non-rectangular control windows with sizable cross-ambient temperature shifts. The data indicate that coupled operation of the compressor and evaporator fan, along with the use of a single constant temperature bandwidth at all fresh food thermostat set points, results in strong temperature dependency between compartments.

ALTERNATE CONTROL APPROACHES

In the following the compressor is activated by a thermostat in the freezer compartment, and the dampers are activated by their own sensors in the fresh food compartment. The evaporator fan is "coupled" to operate when the compressor is on or is "uncoupled" from the compressor and operates instead when the fresh food requires cooling regardless of the compressor. The condenser fan is tied to compressor operation, as before.

Snap damper control

The computer analysis using an automatic "snap" damper produced data for Fig's 3 and 4. The snap damper opens (fully) only when the fresh food compartment requires cooling, otherwise it is fully closed. Fig. 3 shows the temperature window for operation with the coupled compressor and evaporator fan using constant temperature bandwidths in both compartments. The window indicates a much improved controller performance over conventional control except under the high ambient temperature (90°F) at the warm freezer, cold fresh food ("wc") setting. In cases like these the damper is open to allow cold air in, but freezer temperature is satisfied so the compressor and fan are off and no cooling occurs. This happens at control points requiring a relatively warm freezer but cold fresh food compartment.

Significantly better control quality is achieved using both variable temperature bandwidths and uncoupled compressor and fan as indicated in Fig. 4. In this mode the evaporator fan provides cooling of the fresh food compartment while the freezer is satisfied and the compressor is off. It has been shown that variable temperature bandwidths allow "fine-tuning" of the plant's control window [3]. Improved control is achieved by adjusting the temperature bandwidth and center point to accommodate the specific requirements of the plant at each individual control point. However, negligible savings in plant energy consumption results. Uncoupling the compressor and evaporator fan, on the other hand, provide a coarse adjustment of the control window. Plant energy usage results given in Table 1 indicate that compared to the traditional control approach, uncoupled compressor/fan and variable bandwidths allow energy savings that average 9.8% and 20% across the major control points at the two ambient conditions of 70 and 90°F.

Creep damper control

Similar results were obtained for a creep style damper as shown in Fig. 5 and 6. To implement such uncoupling here requires use of a switch on the damper door to activate the evaporator fan semi-independently of the compressor. Note that the creep damper provides a wider range of refrigerator temperature control for the same range of freezer control compared to the snap damper, particularly at the cold end of the window. There is also a higher degree of temperature dependence between compartments in both ambients, compared to the snap damper. Compared to the traditional single control approach however the creep damper provides greater temperature independence between compartments.

Table 1 indicates that plant energy efficiency is not as favorable with a creep damper compared to a snap damper. Thus in contrast to the snap damper results, use of a creep damper with variable temperature bandwidths allows a slightly better savings in energy than uncoupling the compressor and fan. Again, the greatest reduction in energy occurs at the higher ambient temperature. Consistent with the snap damper results, the creep damper shows the greatest benefit to control quality and energy efficiency when used with both the uncoupled and variable bandwidth control algorithms. Fig. 7 and 8 give plant energy predictions for the various control approaches and set points where "c" = cold, "m" = mid, and "w" = warm control settings.

TABLE 1 - Plant Energy Consumption

damper type	compressor/fan control	temperature bandwidths	average energy consumption across major control points at 70°F (kWhr/day)	at 90°F	energy savings relative to traditional control at 70°F (%)	at 90°F
manual	coupled	fixed	553.1	1028.2	------	------
snap	coupled	fixed	580.3	907.6	-4.9	11.7
snap	uncoupled	fixed	528.7	886.6	4.4	13.8
snap	coupled	variable	554.9	911.2	0.0	11.4
snap	uncoupled	variable	499.1	821.5	9.8	20.0
creep	coupled	fixed	575.2	976.3	-4.0	5.0
creep	uncoupled	fixed	567.6	964.6	-2.6	6.2
creep	coupled	variable	561.5	975.0	-1.5	5.2
creep	uncoupled	variable	555.1	936.4	-0.3	8.9

CONCLUSIONS

The plant model indicates that at the major plant control points better temperature control is possible with less energy using automatic dampers and nonconventional control strategies. Using the traditional control approach energy savings with a snap damper is twice that of a creep damper (11+% compared to 5%) but the savings occurs only at the higher ambient regardless of bandwidth. The plant's control windows are more ideal for the snap damper. The snap damper and creep damper energies can be reduced using variable bandwidths. Both types of dampers provide greater cross-ambient temperature shift compensation. The creep damper does allow a greater range of refrigerator temperature control but with a higher degree of temperature dependence between compartments than the snap damper. In general, with either automatic damper, uncoupling the compressor and fan results in an improved control window in both ambients. The greatest benefit to plant control quality and energy consumption occurs when either damper is used in combination with the uncoupled *and* variable bandwidth approaches. Thus optimal control of the plant appears possible at a cost of increased algorithmic complexity. Because of this it may be necessary to use nontraditional control paradigms to implement it [4].

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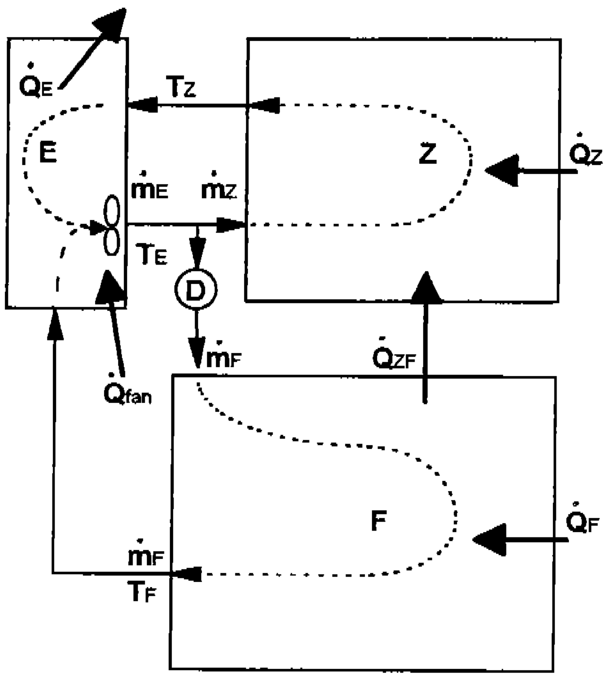

Fig. 1, Air flow diagram showing control volumes. Note, **D** represents the damper valve.

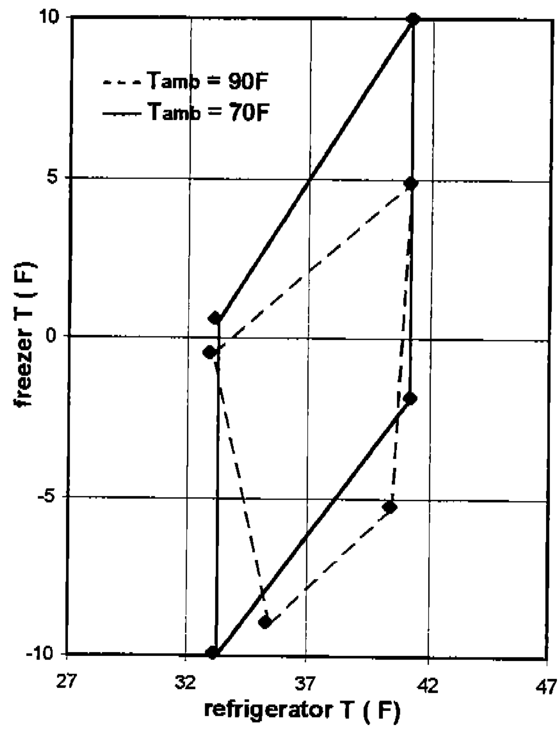

Fig. 2, Coupled compressor/fan with manual damper and fixed bandwidth control.

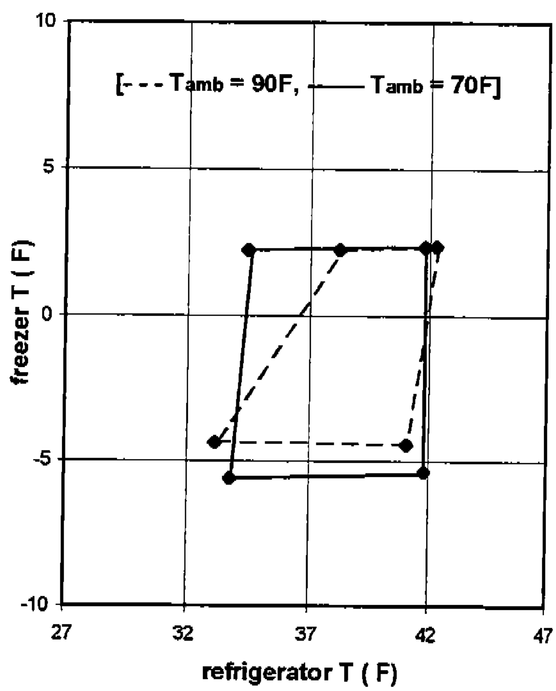

Fig. 3, Coupled compressor/fan with snap damper and fixed bandwidth control.

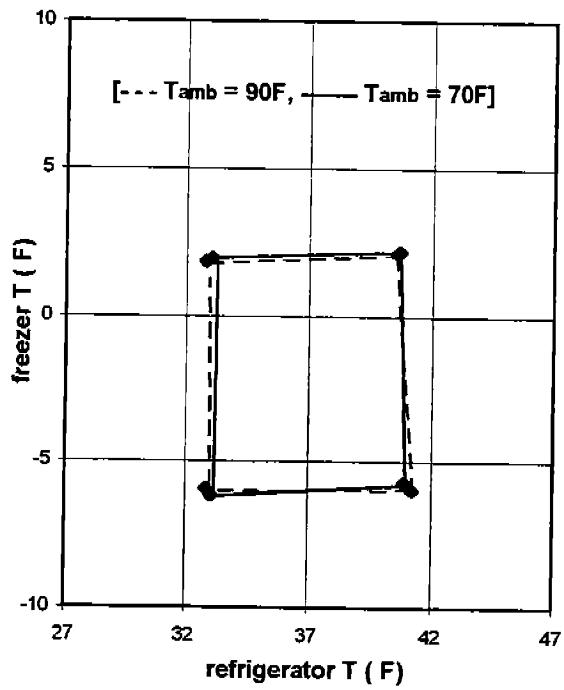

Fig. 4, Uncoupled compressor/fan with snap damper and variable bandwidth control.

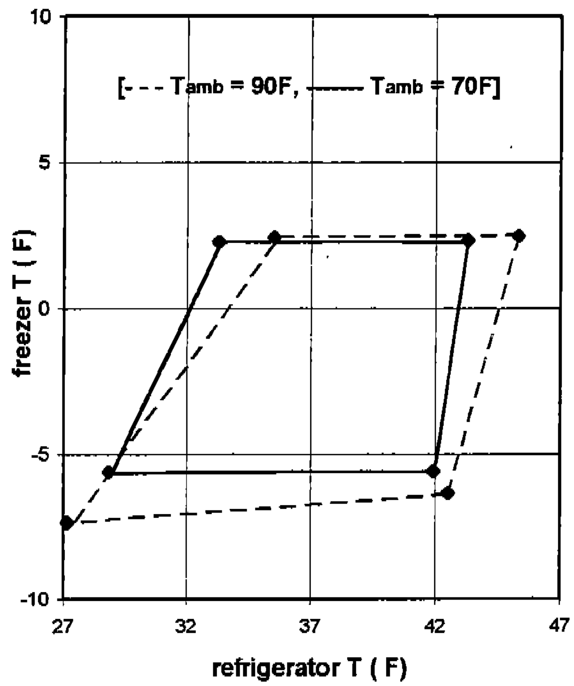

Fig. 5, Coupled compressor/fan with creep damper and fixed bandwidth control.

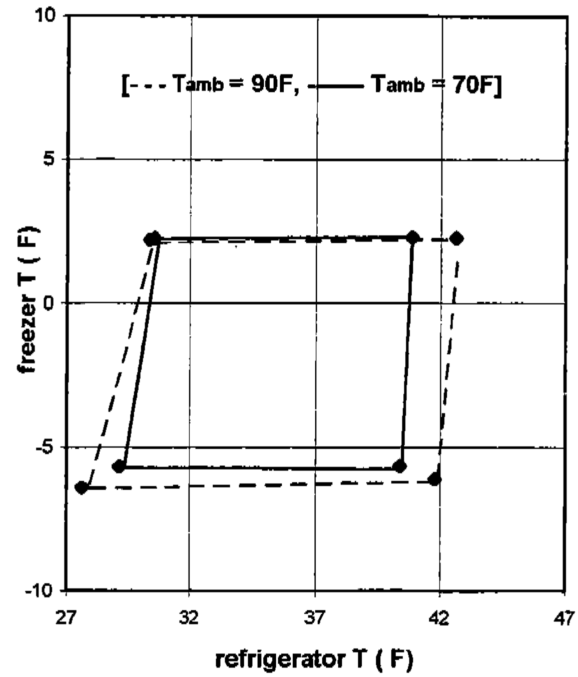

Fig. 6, Uncoupled compressor/fan with creep damper and variable bandwidth control.

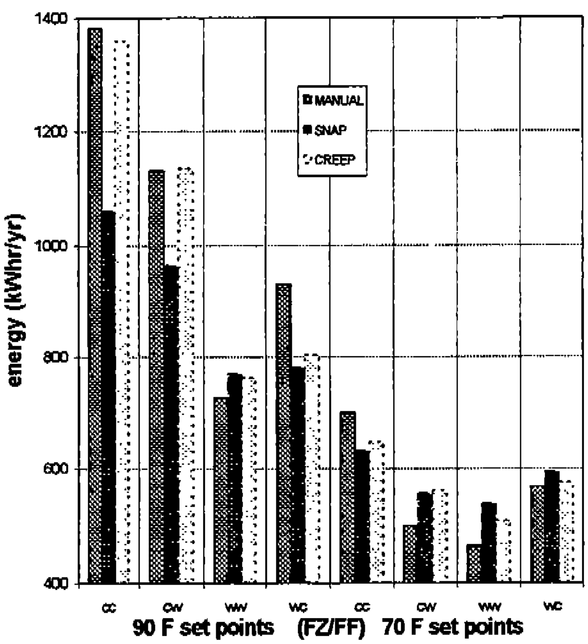

Fig. 7, Plant energy use for all dampers under conventional control, i.e., coupled compressor/fan and fixed bandwidths.

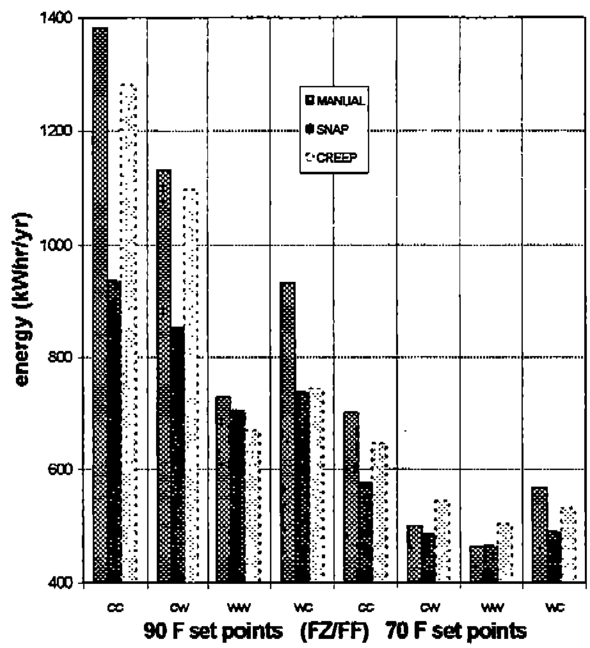

Fig. 8, Plant energy use for manual damper under conventional control and snap and creep dampers with uncoupled compressor/ fan and variable bandwidths.