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Perry Johnson, Arvind Padgaonkar, John Bush, and George Anderson
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ABSTRACT

This paper describes the development of an analytical tool for the design automation of the temperature blend door mechanism in an automotive HVAC system. The function of the blend door is to control the temperature of the air blown into the cabin interior by regulating the mix of air passing through the heater core. The objective in the design process is to achieve a prescribed function of temperature with respect to control position at the instrument panel. The control effort to effect the desired temperature change is also another important consideration for customer satisfaction.

The current design process is empirical in nature and relies on laboratory and vehicle testing with prototypes. The process is also iterative in nature and may continue until the end of the overall design cycle of the complete air handling subsystem.

A parametric feature-based computer model, described subsequently in detail, allows for virtual prototyping of the blend door control mechanism. With this model, various blend door designs can be explored early in the design cycle. Late changes required for packaging can also quickly be analyzed. Kinematic analysis of the blend door mechanism allows the control curve and control effort to be predictive. This desktop simulation tool will enable designs to be optimized or permit the number of options to be narrowed, which will reduce test time and shorten design cycle time.

Illustration of the desktop tool is based upon predicting control curves and efforts for a production HVAC system. Also, airflow and temperature data from CFD analysis for this production system is presented. Finally, predicted results are compared and correlated with actual laboratory and vehicle test data.

INTRODUCTION

The function of the temperature blend door is to control the ratio of air which passes over the heater core so that air temperature exiting the ducts of the air handling system can be regulated or adjusted for occupant thermal comfort. Figure 1 illustrates a typical air handling case, which houses the blend door and heater core, along with the control head and cable assembly. To control the temperature, the operator either turns a rotary control knob or slides a lever located on the instrument panel (I/P). The control positions are typically color coded with a blue (cool) to red (warm) scale. An actuator control mechanism adjusts the angular position of the blend door based on input from the control knob or lever at the instrument panel. This actuator mechanism is typically electric or cable driven. The cable designs can be pull/pull or push/pull (Reference 1) and often include a cam or cam slot device to obtain the desired door travel and position.

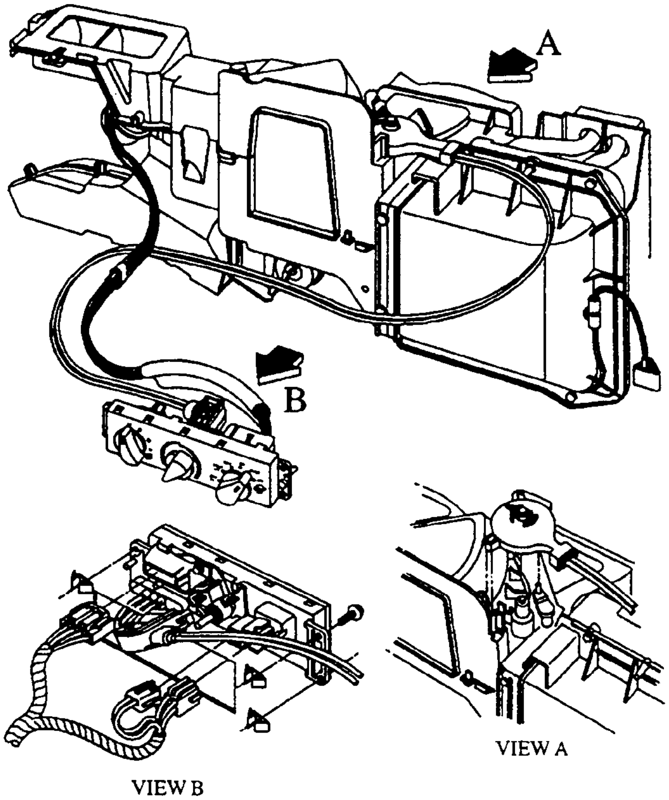


Figure 1: Case, Control Head and Cable Assembly

When the I/P temperature control is set to full cool (blue), the actuator adjusts the angular position of the blend door angle such that no air passes through heater core (see Figure 2A). When the control is adjusted to full warm (red), the actuator adjusts the blend door angle such that all of the air flow passes through the heater core (see Figure 2B). Correspondingly, all other control positions between full cool and full warm represent intermediate blend door angular positions in which some ratio of the total air flow is diverted through the heater core, while the remainder bypasses it. Given a specified temperature output for each I/P control setting, the fundamental design goals are:

A.) Determine the blend door position that will provide the desired temperature output.

B.) Determine the actuator design that will position the blend door to this correct angular position.

C.) Determine the corresponding amount of operator effort in terms of torque for a rotary knob or force for a slider lever, compare to desired limits and allow for parametric design changes if not acceptable.

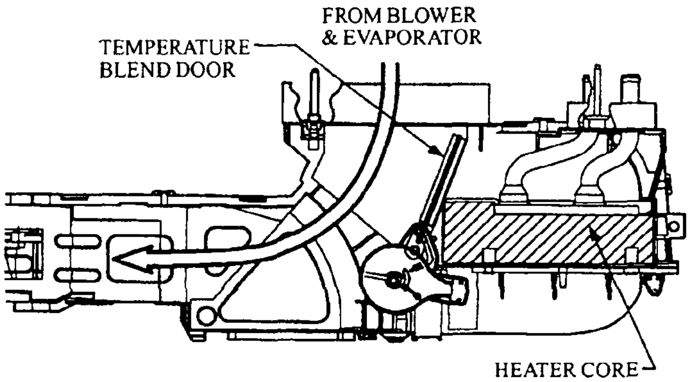


Figure 2A: Air Flow Path – Full Cool

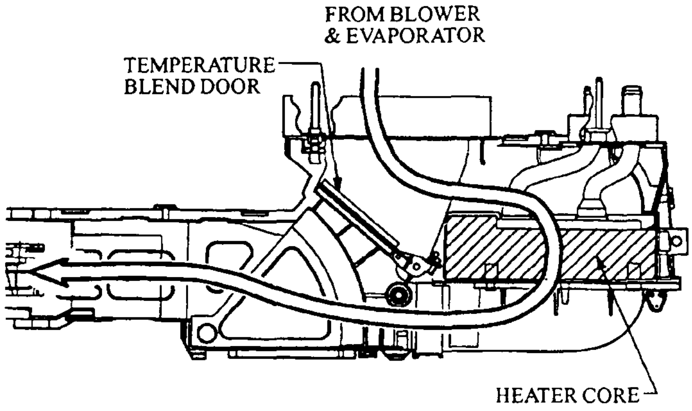


Figure 2B: Air Flow Path – Full Warm

This paper presents the development of the desktop design tool using the pull/pull cable with cam type actuator and rotary control knob. The effort can be extended, however, to any combination of actuator and control design.

TECHNICAL APPROACH

The Adaptive Modeling Language™ , (AML), (References 2-5) provides high level class definitions necessary to define parametric, feature, and simulation based design systems for specific engineering processes. The advantage to this type of modeling paradigm is that instead of requiring a designer to navigate through a large general purpose CAD tool, a customized user friendly desktop tool is developed to present to the designer only the information necessary to address the design issue at hand. References 6 and 7 describe other efforts by fellow researchers to develop customized design tools for a specific family or class of problems. Because AML is object oriented and extensible, the resulting desktop tool can be maintained and enhanced over time to reflect changes in the engineering process and/or extended to incorporate aspects of the engineering process that were initially ignored based upon some engineering idealization assumption.

In developing the desktop design tool, AML was used to define the parametric feature based model of the cam and cam slot actuator design. From a parametric solid modeling perspective, AML classes were used to parametrically define the actuator components, i.e., the pivot, the shaft, the pulley, the cam and cam slot, and the blend door. The design constraints which define how the blend door rotates about the shaft center, and how the shaft pin slides along the cam slot while the pulley rotates, are incorporated in the simulation-based design model by providing the appropriate design constraint formulas. AML's Class-Subclass and Whole-Part relations and constraint mechanism enable the overall model to possess the ability to simulate the actuator mechanism, as well as reverse engineer the proper cam slot for a given set of geometric inputs and the desired temperature versus control knob outputs.

The components of a cam slot actuator mechanism are the pivot, the shaft, the pulley, the cam and cam slot, and the blend door. Parametrically, these design components can be defined by the shaft center, the shaft arm length, the shaft start angle, the pulley center, the pulley start angle, the door start angle, the door travel angle, and the door size, i.e., length and width as illustrated in Figure 3. Other input design parameters for the design process are temperature versus blend door angle positions and force components on the blend door versus blend door angle positions. Specifying these parameters and the desired type of control curve (Reference 8), i.e., mean, mean with dwell, linear, or linear with dwell, the desktop tool then calculates the shape of the cam and its slot.

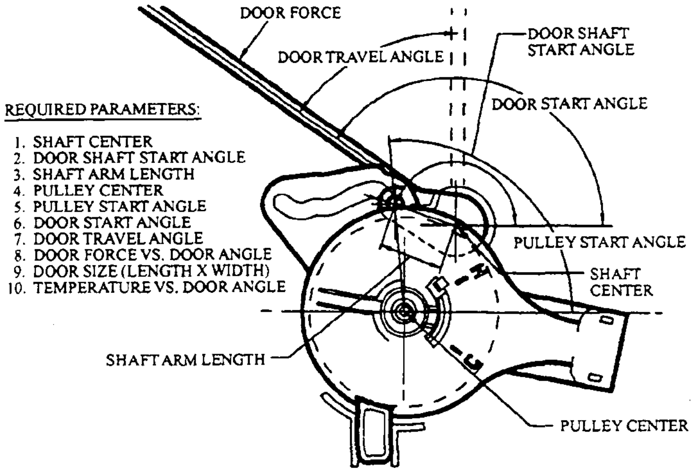
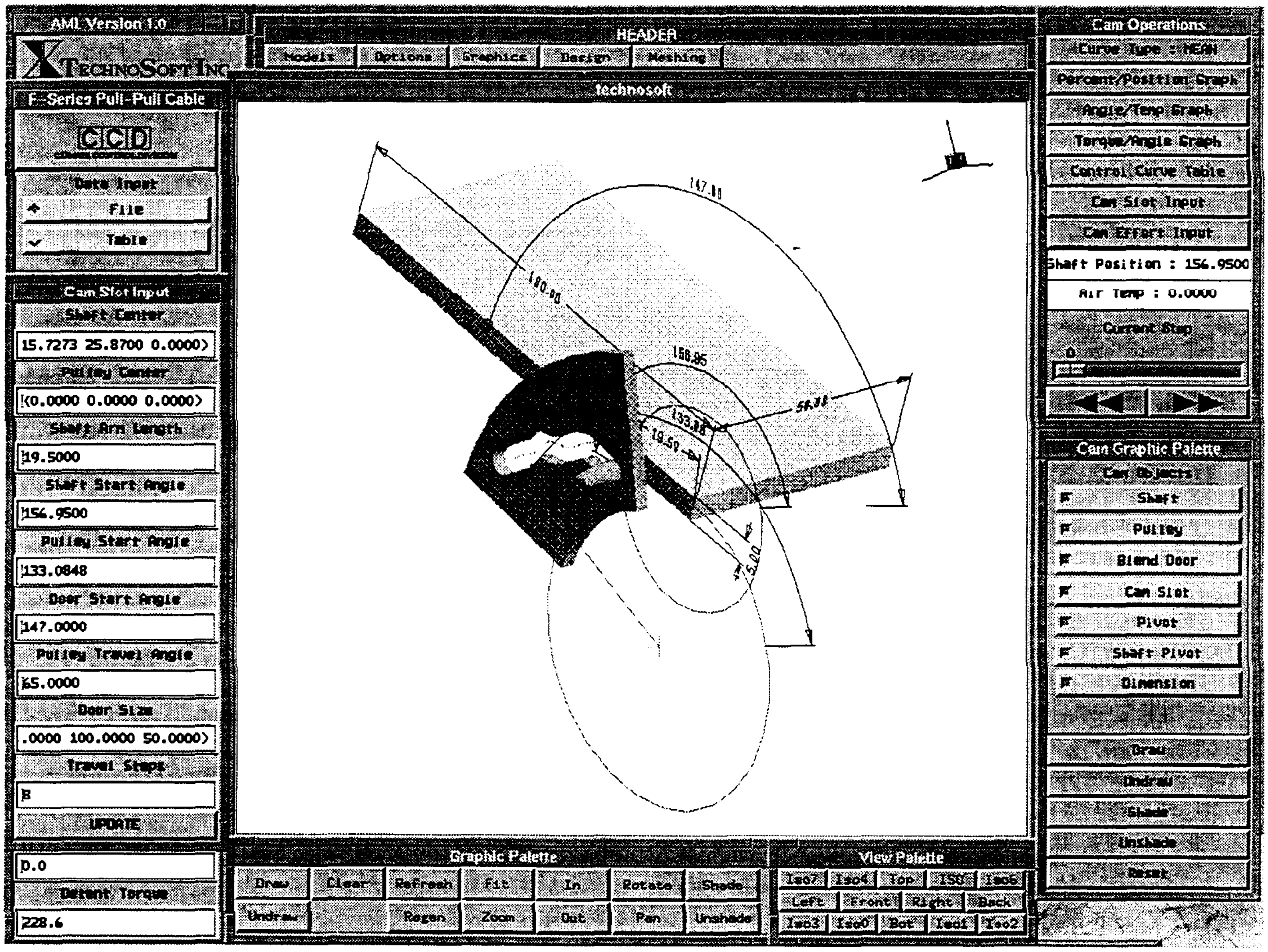


Figure 3: Parameters for Pulley, Cam and Door



Shaft Center: 15.7273 25.8700 0.0000>

Pulley Center: (0.0000 0.0000 0.0000>

Shaft Arm Length: 19.5000

Shaft Start Angle: 156.9500

Pulley Start Angle: 133.0848

Door Start Angle: 147.0000

Pulley Travel Angle: 65.0000

Door Size: .0000 100.0000 50.0000>

Travel Steps: 8

0.0

Detent Torque: 228.6

Curve Type : MEAN

Shaft Position : 156.9500

Air Temp : 0.0000

Current Step: 0

Figure 4: Graphical User Interface

The actual desktop tool comprises a graphical user interface made up of various forms and tables with mouse sensitive buttons that are used to input the design parameters and drive the design process. A graphics window illustrates the geometric aspects of the design along with pertinent graphs (see Figure 4).

The designer specifies whether the temperature and force versus blend door position information will be input as a file or as a table. If the file radial button is selected, another form appears in which the directory and file name can be input. If the table radial button is selected, a form appears in which the designer inputs the minimum and maximum blend door angles, the minimum and maximum temperature values, and the number of table entries. Intermediate temperature and angle positions within the table can be edited to represent actual test stand or CFD simulation values.

Once the method and values for the temperature versus the blend door position are defined, the designer specifies the geometric design parameters using another form. A graphical display of the actuator for the specified geometric design parameters within the graphic window can be controlled by selections to draw, shade, and dimension the various objects from another form. Each time the curve type is changed by selecting the curve type button on the geometric input form and subsequently choosing between mean, mean with dwell, linear, or linear with dwell, the shape of the cam and the cam slot is updated. The motion of the blend door, as controlled by this cam and cam slot design, is animated within the graphical window by selecting the increment forward and backward buttons on the geometric input form. The display of graphs of the desired control curve and blend door angle versus temperature in the graphic window is controlled by additional buttons on the geometric input form.

Similarly, input parameters for determining the torque effort to adjust the control knob, given a specific cam and cam slot mechanism is provided. Here the input quantities are the friction coefficients at the door pivot, the door pin, the pulley and the pulley to control knob cable, the detent torque, the number of pinion gear teeth, the number of ring gear teeth, and the radii of the cable control knob to pulley mechanism at the pulley and the control knob respectively. Once these inputs are defined, a graph of the control effort versus the control knob position can be drawn in the graphic window by selecting this option.

The analytical approach to calculate the control knob torque effort to position the blend door at its various settings is based upon solving eleven simultaneous equations representing the force and moment balance at the blend door pivot point or shaft center, at the cam slot pin, and at the pulley center. Figures 5 and 6 represent pictorially the Free Body Diagram (FBD) that illustrates these force moment balances. In particular, if one first focuses on the shaft center (i.e., axis system number 1), the first four equations describing two force and one moment balance equation about the 1 axis system and a moment equation to define the frictional moment (Reference 9) about the pivot are defined:

$$\Sigma M_a = 0 = (F_{dx_1} \times r_d) + (-F_{px_1} \times r_p) - M_f \qquad (1)$$

$$\Sigma F_{x_1} = 0 = -F_{dx_1} + F_{px_1} - F_{ax_1} \qquad (2)$$

$$\Sigma F_{y_1} = 0 = -F_{dy_1} + F_{ay_1} + F_{py_1} \qquad (3)$$

$$M_f = [(F^2_{ax_1} + F^2_{ay_1})^{1/2} \times r_{ds}]\mu_1 \qquad (4)$$

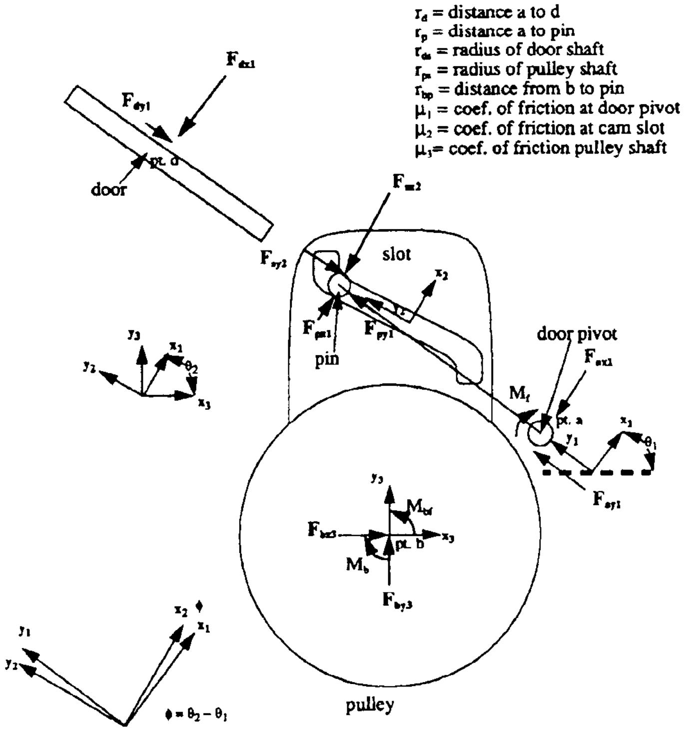

r_d = distance a to d
r_p = distance a to pin
r_{ds} = radius of door shaft
r_{ps} = radius of pulley shaft
r_{bp} = distance from b to pin
μ_1 = coef. of friction at door pivot
μ_2 = coef. of friction at cam slot
μ_3 = coef. of friction pulley shaft

Figure 5: Free Body Diagram – Pulley, Cam and Door

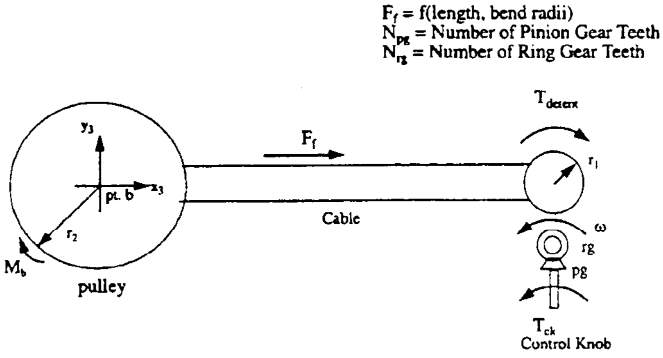

F_f = f(length, bend radii)
N_{pg} = Number of Pinion Gear Teeth
N_{rg} = Number of Ring Gear Teeth

Figure 6: Free Body Diagram – Cable and Control

Next the force balance about the cam slot and pin along with the definition of the frictional force balance are represented by the following three equations:

$$\Sigma F_{x_2} = 0 = -F_{s_{x_2}} + F_{p_{x_1}} \cos\phi + F_{p_{y_1}} \sin\phi \qquad (5)$$

$$\Sigma F_{y_2} = 0 = -F_{s_{y_2}} - F_{p_{x_1}} \sin\phi + F_{p_{y_1}} \cos\phi \qquad (6)$$

$$F_{s_{y_2}} = \mu_2 F_{s_{x_2}} \qquad (7)$$

Finally, the force and moment balance about the pulley due to the forces exerted from the cam slot along with the definition of the frictional moment about the pulley axis system, i.e., axis system 3 are represented by the following four equations:

$$\Sigma M_b = 0 = -M_b + M_{b_f} + (F_{s_{x_2}} \cos\theta_2 - F_{s_{y_2}} \sin\theta_2) r_{b_p} \qquad (8)$$

$$\Sigma F_{x_3} = 0 = F_{b_{x_3}} - F_{s_{x_2}} \cos\theta_2 + F_{s_{y_2}} \sin\theta_2 \qquad (9)$$

$$\Sigma F_{y_3} = 0 = F_{b_{y_3}} - F_{s_{x_2}} \sin\theta_2 - F_{s_{y_2}} \cos\theta_2 \qquad (10)$$

$$M_{b_f} = [(F^2_{b_{x_3}} + F^2_{b_{y_3}})^{1/2} \times r_{p_s}]\mu_3 \qquad (11)$$

Of the 21 variables, 10 variables are either known geometric quantities (e.g., r_d, r_p, r_{d_s}, r_{p_s}, r_{b_p}), known frictional quantities (e.g., μ_1, μ_2, μ_3), or force components on the blend door which are known either from test or CFD results (e.g., $F_{d_{x_1}}$ and $F_{d_{y_1}}$). The remaining 11 variables (i.e., $F_{a_{x_1}}$, $F_{a_{y_1}}$, M_f, $F_{p_{x_1}}$, $F_{p_{y_1}}$, $F_{s_{x_2}}$, $F_{s_{y_2}}$, M_{bf}, M_b, $F_{b_{x_3}}$, $F_{b_{y_3}}$) are the unknowns to be solved for using the 11 equations above. Having solved for these 11 unknown variables, the control knob torque can be calculated by the following equation (see Figure 4):

$$T_{ck} = M_b \left(\frac{r_1}{r_2}\right)\left(\frac{N_{pg}}{N_{rg}}\right) + \left[F_{f_{cable}} \times (r_1)\left(\frac{N_{pg}}{N_{rg}}\right) + T_{detent}\left(\frac{N_{pg}}{N_{rg}}\right)\right] \qquad (12)$$

ANALYTICAL PREDICTION AND EXPERIMENTAL VERIFICATION

Thermal flow calculations were computed for a sample vehicle air handling subsystem with the blend door configured at various positions. The three dimensional computational fluid dynamics (CFD) calculations were performed using SPECTRUM™ (Reference 10) to solve the incompressible-thermally coupled Navier Stokes equations. In addition, turbulence modeling in the flow was performed. SPECTRUM™ is a finite element solver which employs a Galerkin Least Squares scheme and is approximately second order accurate.

Steady state thermal flow calculations were performed for various blend door configurations ranging from fully open to fully closed with three intermediate positions (i.e., 0, 1/4, 1/2, 3/4, and 1). Figure 7 illustrates the correct control volume for the blend door position at 1/2. In Figure 7, the inflow, blend door, heater core and the outflow surfaces are highlighted with surface elements, nodes, and/or normal vectors in order to make these components clear to the reader. In order to perform the CFD calculations, the control volume defined for each blend door position is discretized by means of an automatic mesh generator (Reference 11) that produces unstructured tetrahedral or hexadedral elements. Mesh attributes were set to two settings to provide unstructured tetrahedral meshes that are coarse in nature (~40,000 elements) and that are fine in nature (~300,000 elements), see Figure 8 for an example of a fine mesh). The coarse meshes where used for debugging the solution strategy whereas the fine meshes were for correlation purposes. In addition, comparing results for a particular blend door setting for both coarse and fine meshes provides a qualitative feeling about how mesh sensitivity affects the CFD results of interest.

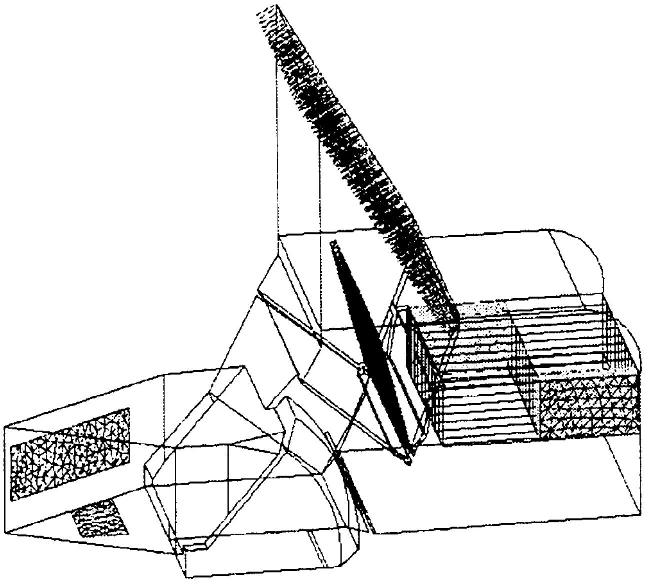


Figure 7: Control Volume – Blend Door at 1/2 Position

Inflow conditions for the thermal flow analysis were based on the blower being set to a mass inflow rate of 150 scfm. However, as the door position varies the inflow mass rate also changes form the nominal 150 scfm value. This flow rate dependence on the door position was taken into account when performing the CFD calculations. The inflow temperature of 34.0° F was also prescribed as a boundary condition. Finally, the heater core capacity was adjusted for each blend door position Flow analysis results were correlated with test stand data by comparing temperature at the floor mode position outflow for each blend door position. Figure 9 provides a temperature contour for a "mid" slice of the control volume with velocity vectors for the blend door position of 0 using the fine mesh. Figure 10 provides a graph showing how the CFD results correlate with the experimental results.

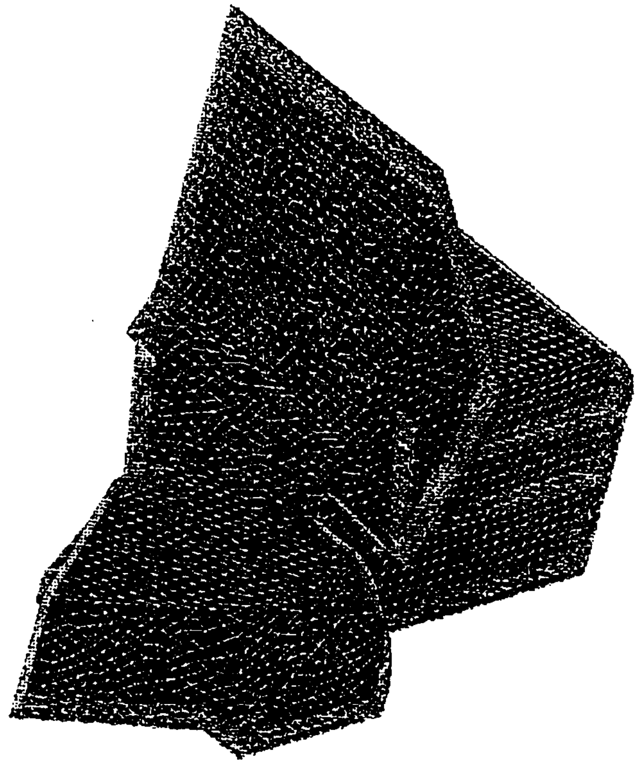


Figure 8: CFD Mesh

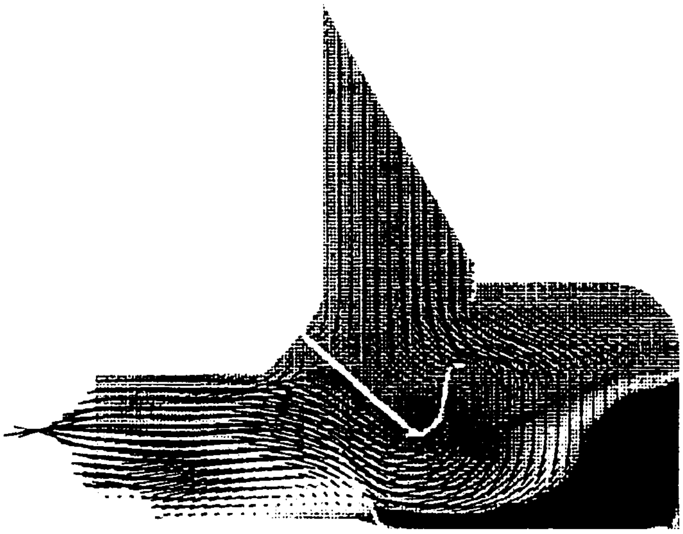


Figure 9: Temperature Contour Plot with Velocity Vectors

Control knob efforts for a particular cam slot design (Figure 14) were measured. Figure 11 presents the experimental effort result for the floor mode case with a high blower setting. A difference in values between turning the knob from cold to warm and warm to cold was observed. This effect is due to the fact that when turning the knob from warm to cold the moment due to the air blowing on the door must be overcome as well as the detent and frictional cable torques. Figure 11 also presents the predicted effort for the same cam using the 11 equation method described earlier. As a result, the correlation between the experimental and predicted values can be assessed and are judged to be satisfactory for design purposes.

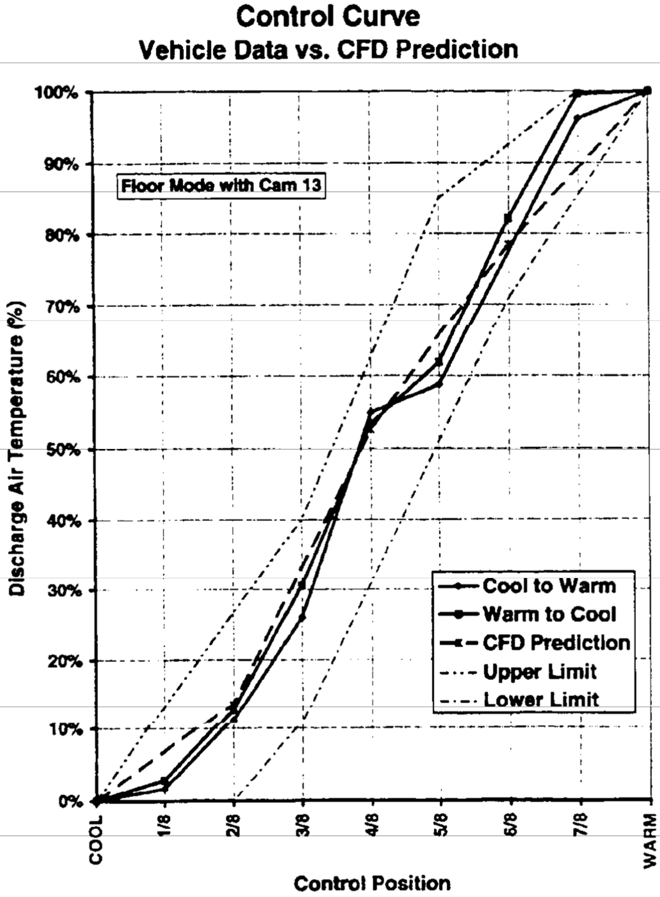


Figure 10: Control Curve Correlation

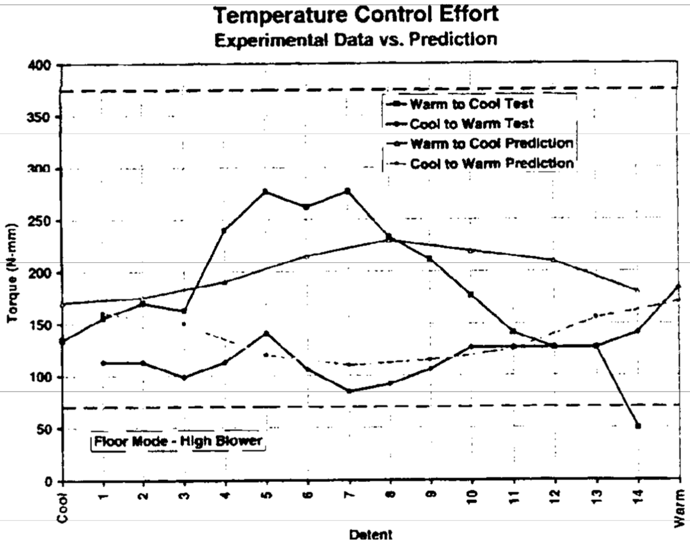


Figure 11: Control Effort Correlation

Finally, four new cam slot profiles were designed using the desktop tool based upon specified control curves with mean, mean with dwell, linear, and linear with dwell distribution. Figure 12 is the mean control curve and Figure 13 is the resulting cam profile and can be compared to the original cam design in Figure 14. Figure 15 is the resulting effort curve for this new cam profile and should be compared with Figure 10.

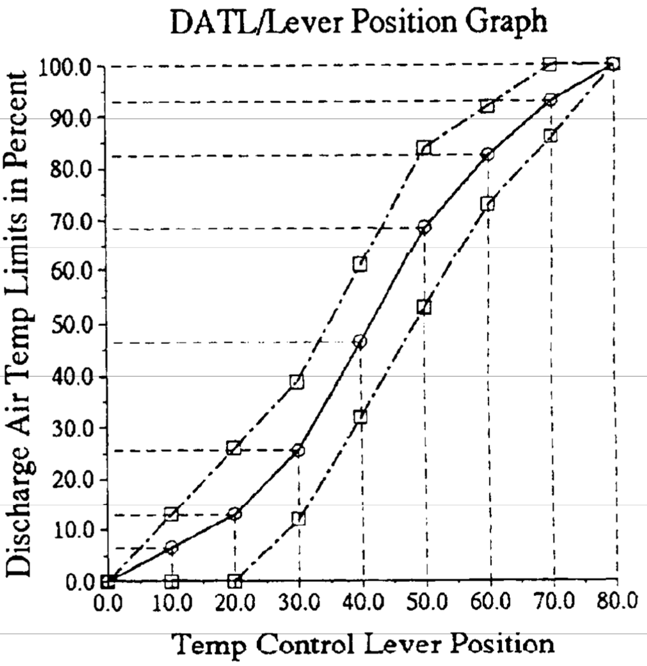


Figure 12: Mean Control Curve

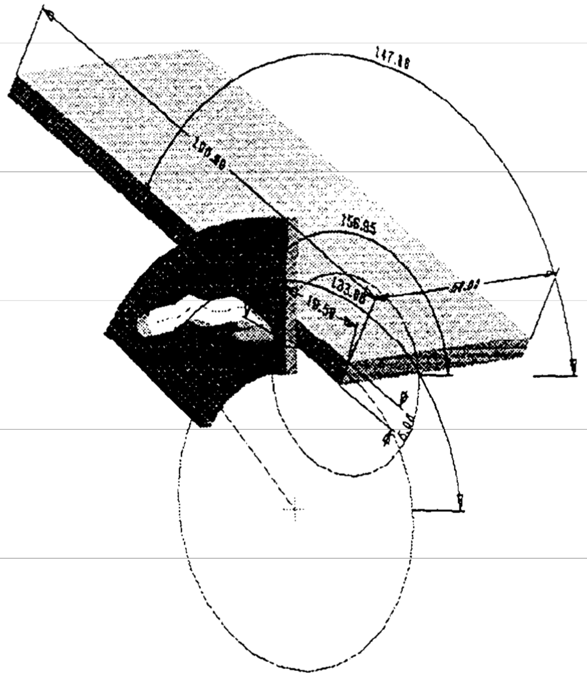


Figure 13: Derived Cam Profile

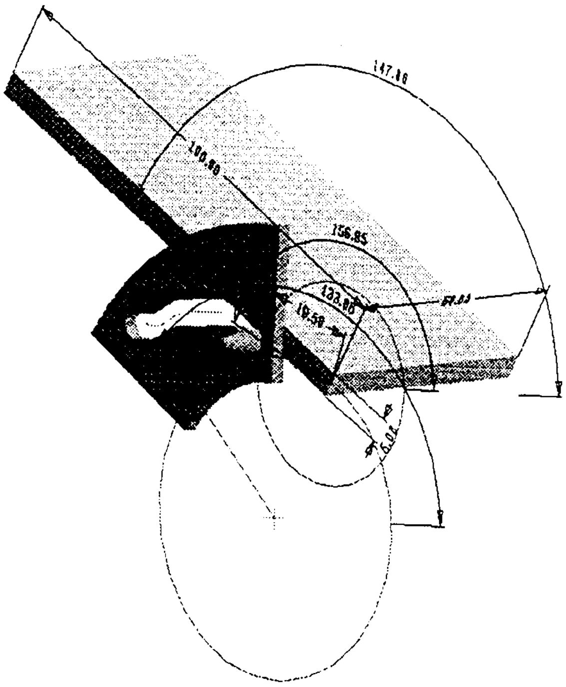


Figure 14: Original Cam Profile

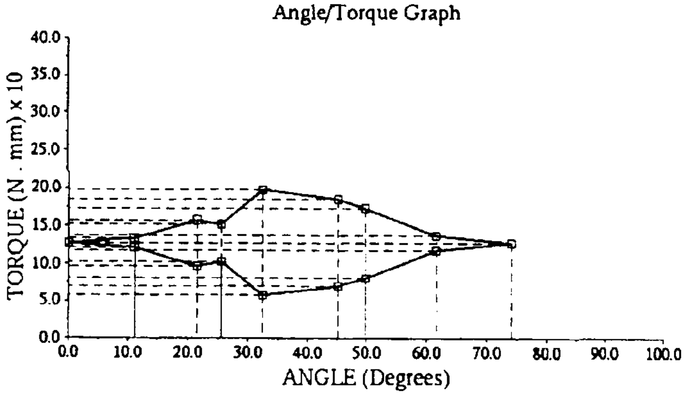


Figure 15: Control Effort for New Cam Profile

CONCLUSIONS

This paper reviews the development of a new desktop design tool which allows virtual prototyping of the temperature blend door and related control mechanism in an automotive HVAC system. Using this tool, the blend door control mechanism, including the actuator, can be designed up front to achieve the desired temperature relationship with the control knob or lever position. Also, through kinematic analysis, the operator control effort can be predicted and compared to specifications or desired limits. By using parametric analysis early in the design cycle, a reduction in the number of design iterations and tests with physical prototypes is achieved. The expected result, therefore, is a design cost savings and shorter design cycle times.

A correlation is demonstrated for a sample air handling system between the analytical predictions and experimental data for several temperature points on a control curve. A similar correlation is also demonstrated for the operator effort (torque) at the temperature control knob.

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