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PRECISION TEMPERATURE CONTROL FOR OPTICS MANUFACTURING

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Precision Temperature Control for Optics Manufacturing

by

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Abstract

The principles of precision temperature control are presented in this paper. Emphasis is placed on the use of heat exchangers with large flowrates of coolant. Design considerations for such systems are highlighted. In particular, the selection of system components appropriate to the desired level of performance is discussed. Several feedback control techniques are also presented.

Introduction

One of the largest sources of inaccuracy in optical components arises from thermally induced distortions. These can occur in the manufacture, inspection, and assembly of optical components. A number of techniques are presented for achieving temperature control at appropriate levels of precision. The emphasis is on the design principles required for constructing a system for supplying temperature controlled coolant to a machine or instrument. The specific details of how that coolant is used are not discussed. A number of practical, precision feedback control techniques are presented, and alternative system designs, consisting of commercially available components, are discussed.

At the Lawrence Livermore National Laboratory (LLNL), precise diamond turning machines are brought to a thermal equilibrium (and maintained there) by using massive flows of temperature controlled air and liquids. Bryan[1,2] has used external oil showers on machine tools to greatly improve their thermal stability and others[3,4] have used air showers. Other machines at LLNL[5,6] combine an external air shower with internal water cooling in areas of concentrated heating. Air showers are not effective for removing large quantities of heat, but they are convenient for minimizing the influence of external heat sources on structures with negligible internal heating. For most applications, a thermal gradient in a structure is permissible, as long as it is stable with respect to time. Therefore, all heat loads on precision machines should be made as constant as possible. It is the suppression of time-varying heat loads that determines the amount of cooling required for a machine. However, if a short thermal "soak out" time is not necessary, critical structural elements should be insulated. This reduces their sensitivity to time-varying heat loads. The thermal response time of a machine also has a great impact on the temperature control system. It determines the frequency range over which the control system must perform. This implies that the required temperature stability of the machine coolant is frequency dependent, with large, high-frequency fluctuations being tolerable.

Heat Exchanger Design

At LLNL, the primary method for controlling machine-coolant temperature uses a chilled-water-cooled heat exchanger. This is true whether the machine coolant is air, water, oil, or some other fluid. Water, however, is an excellent coolant because of its high heat capacity and low viscosity. Both the flowrate and the temperature of the chilled water influence the machine-coolant temperature at the heat exchanger outlet. However, heat exchangers respond faster to flowrate changes, and flowrate is easier to manipulate than chilled-water temperature. Therefore, most systems at LLNL manipulate chilled-water flowrate to achieve temperature control, while holding the chilled-water temperature constant.[6] By controlling a heat exchanger with this technique, the temperature control problem becomes one of flowrate control. Different flowrate-control methods are presented in the following section.

The design or selection of a heat exchanger is critical if precise temperature control is to be achieved. The following discussion focuses on liquid heat exchangers cooled by chilled water, but most of the comments are applicable to air heat exchangers as well. The static and dynamic characteristics of candidate heat exchangers must be examined to determine their ability to provide precise temperature control. This is complicated by the fact that the transient response and the steady-state heat transfer are a nonlinear function of chilled-water flowrate. The steady-state heat balance for a heat exchanger is given by the equation:

$$Q = UA\,\Delta T_m = C_p \dot{m}\,(T_o - T_i) \tag{1}$$

where

- Q - heat load (energy/time)
- UA - overall heat transfer factor of the heat exchanger
- ΔT_m - logarithmic mean temperature difference across the heat exchanger
- C_p - specific heat of either fluid
- $\dot{m}$ - mass flowrate of either fluid
- T_o - bulk outlet temperature of either fluid
- T_i - bulk inlet temperature of either fluid

Here the bulk temperature is defined as the average temperature of a fluid, integrated across its cross section at a particular point in a pipeline. A typical solution to this equation is shown in Fig. 1. It indicates the chilled-water flowrate and temperature required to obtain a specified rate of heat transfer. For this example, it is assumed that the flowrate of the machine coolant is fixed and its outlet temperature is nominally 0°C. This type of a plot is very useful in the evaluation of candidate heat exchangers. For a constant inlet temperature, Equation 1 shows that a change in the heat transfer rate Q produces a proportional change in the outlet temperature of the machine coolant. Therefore, the slopes of the curves in Fig. 1 indicate the temperature sensitivity to flowrate errors at different chilled-water temperatures. For control system design, the slope also represents the input-output gain of the system, which ideally should be constant. However, the gain varies greatly with flowrate, with the steepest slope occurring at small flowrates. Also, the slope increases as the chilled-water temperature drops. Therefore, if the heat load is reduced, then the chilled water temperature can be raised to reduce the sensitivity to flowrate errors.

The chilled-water set-point temperature is chosen from Fig. 1 so that the necessary chilled-water flowrate is in the mid-range of the flow capacity. However, the flow capacity should not be greatly oversized, because it degrades the precision of the flow control. The required range of the flow controller depends on the range of heat loads and the fluctuations in the chilled-water supply temperature. The operating region shown in Fig. 1 indicates the range of flowrates required to remove different heat loads when the chilled-water temperature varies ±1.0°C about its set point. If a wide range of flowrates is required, the nonlinear effects of manipulating flowrate are more significant. An alternative solution is to vary the set-point temperature of the chilled-water supply so that the variation in flowrate is reduced when the heat load changes. Fortunately, with careful design and operation, many precision machines do have a fairly constant heat load. A large fraction of the heat often comes from pumping the coolant. It is assumed here that the temperature control system has only to provide net cooling, and never net heating. This greatly simplifies the system design and the control task.

To provide precise temperature control, the process of heat removal from the machine coolant must be closely controlled. In most applications, heat exchangers are designed for efficiently transferring heat, but for precision temperature control, accurate manipulation of small amounts of heat is most important. Therefore, the heat exchanger must be selected in a different manner from the industrial norm. For instance, if a 400 l/min flow of water requires temperature stability to ±0.0001°C under a heat load of 2.3 kW, the equivalent heat removal accuracy is only ±0.1% of full load. In terms of heat removal control, the temperature control problem is actually made easier with large flowrates of machine coolant. This is true because the temperature error corresponding to a given error in heat removal is smaller for larger flowrates (see Equation 1).

Many types of heat exchangers can satisfy the basic design goals of adequate heat capacity and structural integrity. However, the following factors, in approximate order of priority, must be considered in the design or selection of heat exchangers for precision temperature control:

(1) Low sensitivity to flowrate errors
(2) Good mixing and uniform heat transfer
(3) Fast transient response
(4) Ease of fabrication
(5) Low pressure drop

Decreasing the sensitivity to errors in the chilled water flowrate is most important, because it dictates how closely the flow must be controlled. Good mixing and uniform heat transfer are next in importance because they are needed for accurately measuring the bulk temperature of the fluid. Because outlet temperature is usually sensed at one point in the outlet cross section, it may not be representative of the bulk temperature of the fluid. For example, a time-varying temperature distribution across the fluid cross phenomenon is often the factor determining the ultimate precision of the temperature control. The third design objective is a fast transient response. The heat exchanger dynamics limit the bandwidth of the control system, which in turn determines the amount of disturbance rejection at any given frequency. Finally, the last two design objectives are carry-overs from standard heat-exchanger design practice.

Some of these design objectives conflict, but a cross-flow heat exchanger design satisfies them more than most other types of heat exchangers. They are simple to build, have well-defined flow patterns, and have a small pressure drop. Good mixing of the machine coolant occurs with it flowing on the shell side of the heat exchanger. Mixing is also promoted by avoiding oblong flow passages on the shell side and by arranging the tubes in a staggered pattern. The spacing of the tubes also influences heat transfer and mixing. To obtain a compact, fast responding heat exchanger with uniform heat transfer, a large number of small diameter tubes is necessary. The convection coefficients inside and outside the tubes and the heat load determine the size of the heat exchanger. To improve convection and enhance mixing, both the tube- and shell-side flows should be well in the turbulent flow regime. High velocities through the heat exchanger also reduce transit times of the two fluids flowing through it. This is often the dominant factor limiting the speed of response, particularly for air-to-water heat exchangers.

Methods of Heat Exchanger Control

With a flowrate-controlled heat exchanger, the chilled-water supply temperature must be stabilized by some means. If the temperature drifts from its set point, the flow controller can compensate for the deviation over a limited range. Therefore, a relatively crude, on/off controlled, commercial refrigeration unit can often be used. Typically, the water supply can be maintained to within ±1.0°C of its set point. If the fluctuations in the chilled-water-supply temperature are too large for adequate heat exchanger control, an in-line heater can be used to reduce them. Only shown in Fig. 2a, a heater can be used with any of the other heat-exchanger control techniques in Fig. 2. For precise control, however, a mixing tank should be placed in the line after the heater. Experience has shown that heaters do not uniformly transfer heat, producing slugs of hot and cold water. Heaters are typically controlled in a rapid on/off mode that simulates proportional control. The controller, heater, and additional temperature sensor are available commercially.

Manipulating flowrate allows fine control of a heat exchanger's exit temperature, but precise flowrate control is now required. The most commonly used technique at LLNL for controlling flow in heat exchangers is illustrated in Fig. 2a. Two on/off solenoid valves are employed, one normally open and one normally closed. The bypass valve opens as the supply valve closes, thus avoiding water hammer problems. The solenoid valves are operated by a controller that senses the outlet temperature of the machine coolant. Even though liquid heat exchangers are illustrated in Fig. 2, all three control techniques are equally applicable to chilled-water-cooled air heat exchangers.

If a flow controller is not precise enough, an in-line heater is often used to improve the temperature control of the machine coolant. The heater is mounted after the heat exchanger, as shown in Fig. 2a. Besides the additional hardware that is required, a heater has the disadvantages of less uniform heat transfer and a slower response time than most heat exchangers.

An alternative flow control technique using valves is shown in Fig. 2b. A manual valve has been placed in parallel with an automatic valve. The manual valve is adjusted so that the heat exchanger always provides some cooling. This technique is particularly useful if the heat loads on the system are fairly constant. With such a manual valve, the on/off transients of a solenoid valve have a smaller effect on the heat exchanger, allowing a more precise control. In addition, the response time of the heat exchanger is faster.

A manual valve is not always necessary when a proportional modulating valve is used in place of the solenoid valve in Fig. 2b. However, it may permit the use of a smaller modulating valve in its most linear range. For many proportional valves, such as gate, globe, diaphragm, or ball valves, flowrate is nonlinearly related to valve position. This nonlinearity degrades the flow controller performance. Gate valves typically provide the finest flow control and can be made linear, or even purposely nonlinear to cancel heat exchanger nonlinearities. For fast response, a motor-driven valve is used, but slower pneumatically-driven valves are also in common use. However, these valve drive mechanisms often suffer from backlash and stiction, which limits their flow control capability. These can be compensated for by sensing valve position and closing a control loop on it. However, it is even better to sense the flowrate directly and close a control loop on that. Then valve wear and all the other valve imperfections can be compensated for, nonlinearity on the heat exchanger control loop.

By using a variable speed pump, as shown in Fig. 2c, the flow control problem becomes one of speed control. This is one of the fastest, most precise ways to control flowrate. There are no backlash problems with a pump, and many commercially available motor controllers have very precise speed control. Positive displacement pumps are often used, because the flowrate is directly proportional to pump speed, independent of pressure. Typically, positive displacement pumps pumping water use rubber impellers or nylon gears. However, a centrifugal pump can be used when a small nonlinearity between speed and flow can be tolerated, but then the flowrate becomes sensitive to changes in the downstream flow resistance. However, a flowmeter can be used for closed loop flow control, thereby minimizing the deficiencies of a centrifugal pump. A variety of variable speed motors are available. To be cost effective, the response time and the precision of speed regulation must be appropriate for the application. Inexpensive controllers often use only current feedback from the motor, but more precise control is possible when feedback from a tachometer or flowmeter is used. Typically, tachometers are less expensive and more accurate than flowmeters, so their use is preferable if flowrate feedback is not required for the pump deficiencies. To attain high performance, machine-tool servomotors, such as DC servomotors or variable frequency AC motors, should be used to drive the pump. These motors have response times of tens of milliseconds, which is at least an order of magnitude faster than the response time of most heat exchangers. The dynamics of the flow controller can therefore be ignored when designing the control loop for the heat exchanger.

Design of a Liquid Coolant System

An operational water temperature-control system at LLNL is described here. It serves to illustrate how such a system, shown schematically in Fig. 3, can be implemented. A constant-speed 5 HP, AC motor is used to drive a centrifugal pump that supplies 500 l/min of temperature-controlled water to a precision machine tool. The water flows through a cross-flow heat exchanger and through a 20 m long pipeline to a mixing tank 7 m above the machine. The 1000 l tank and its baffle plate encourage mixing, which tends to average out temperature fluctuations of short duration. Its thermal time constant is 120 s. The water flows by gravity out of the tank to the machine tool. After passing through the tank, the water is much more homogeneous in temperature than before it entered. The water then drains to a 2500 l sump beneath the machine from which the pump recycles it through the system.

A 2.5-HP, 2200-rpm, variable-speed DC servomotor drives a rubber-impeller pump, which can supply up to 120 l/min of chilled water to the tube side of the heat exchanger. Using tachometer feedback, the pump speed can be controlled to an accuracy of ±1 rpm, over a 25 Hz bandwidth. The pump draws from a 1600 l water tank that is maintained at 10° ±1°C by a commercial, on/off-type water-chiller unit. A pump within the chiller circulates the refrigerated water to the tank and back.

The single-pass heat exchanger contains sixty 0.63-cm diameter, brass tubes, each 7.6-cm long. They are mounted in eight rows, each row staggered from its neighbor, to form a 7.6- by 7.6-cm flow passage for the machine coolant. The tube-side manifold, shown in Fig. 4, allows each half of the heat exchanger to have opposite tube flow. This tends to balance the temperature gradient across the heat exchanger. This design can transfer 2.7 kW of heat from a water flow of 400 l/min while maintaining a 20°C outlet temperature. The corresponding chilled water flow at 10°C was 100 l/min. This heat exchanger also responds quickly to changes in flowrates; its bandwidth is 0.8 Hz.

As shown in Fig. 4, a 2.5-cm thick piece of aluminum honeycomb is mounted in front of the thermistor at the outlet of the heat exchanger. The honeycomb breaks up large vortices into small eddies and smoothes the temperature fluctuations by convecting heat to and from the honeycomb walls. To encourage a uniform flow of water over the heat exchanger tubes, a plate with four cutouts is installed at the inlet to spread the flow. A similar plate is also used on the tube side to promote uniform heat transfer throughout the heat exchanger. By using these devices, a single thermistor measurement is more representative of the bulk outlet temperature of the heat exchanger.

At LLNL, thermistors are commonly used for precision temperature measurements. They have high sensitivity, excellent long-term stability, fast response, and ease of use. The thermistor beads in this system are only 0.3 mm in diameter and are bonded to the tips of glass rods which are then mounted in aluminum tubes. These thermistors respond very quickly to temperature changes in flowing water. Their thermal response bandwidth is between 10 and 15 Hz. The temperature induced change in resistance of the thermistor is measured by an AC Wheatstone-bridge circuit with a 0.00001°C resolution. The more common DC Wheatstone bridges provide resolutions ranging from 0.001 to 0.01°C.

Closed-Loop Control Techniques

A number of techniques can be used to close feedback control loops on heat exchangers. Most are commercially available, so the type of controller chosen must be cost effective for the desired level of temperature control. The simplest and most inexpensive controller is an on/off type with an adjustable deadband. The deadband is adjusted so that the nominal on/off cycle time is short compared to the response time of either the machine being cooled, or of a mixing tank located after the heat exchanger. However, this method provides precise temperature control only if the chilled water temperature and the system heat loads are relatively constant. A duty-cycle offset occurs in the machine-coolant temperature if the ratio of on to off periods changes greatly. This type of control, in conjunction with in-line heaters (Fig. 2a), has achieved a ±0.01°C temperature stability in a 160 l/min oil flow.[1,2] This temperature was averaged over a 30 s period, because the cycle time of the solenoid valves was nominally 6 s, which is much faster than the response time of the machine being cooled.

Duty-cycle offset can be corrected if a proportional plus integral (PI) controller is used. The output of a PI controller is typically a voltage signal that is proportional to the amount of flowrate required for temperature control. For an on/off device like a solenoid valve, this voltage signal can be converted to a time-proportioned output that cycles the valve's on and off periods in proportion to the controller output. If the cycle time is short compared to the response time of the heat exchanger, this technique effectively mimics the proportional control of flowrate. However, very durable solenoid valves are required to withstand the frequent on/off cycles. Microprocessor-based PI controllers are commercially available with time-proportioned outputs. Some other digital controllers offer more flexibility than analog controllers, such as nonlinear compensation and averaging of multiple temperature sensors.

The inner feedback loop of the block diagram in Fig. 5 shows how a PI controller can command a motor-speed controller to obtain flow control. Because the temperature-control set point is fixed, the sole purpose of the feedback control is to provide disturbance rejection. Most system disturbances, such as chiller cycling or changing heat loads, are ramp functions. These induce a steady-state following error in a PI-controlled system. To maximize disturbance rejection, the PI controller must be tuned to maximize the open-loop gain of the system at low frequencies, while an adequate stability margin is maintained. Most heat exchangers respond slow enough that many commercially available digital controllers can provide precise temperature control. However, analog controllers are still widely used.

In the system shown in Fig. 3, the machine and the heat exchanger are separated by a 35 m long pipeline and a 1000 l tank. The coolant is well mixed at the machine, but it is subject to additional environmental heat loads. Therefore, closing a feedback loop with temperature sensed only at the heat exchanger was not adequate, and sensing temperature at the machine makes the control system too slow for disturbance rejection. Therefore, an inner control loop was closed at the heat exchanger, and its set point was commanded by another control loop sensing temperature at the machine. A block diagram of this configuration is shown in Fig. 5. This cascaded outer loop compensates for inner-loop following errors and for errors in the inner-loop bulk-temperature measurement. The outer loop is much slower than the inner loop, therefore, it is only capable of low frequency corrections. The final closed-loop performance of the system is recorded in Fig. 6. This temperature record was made at the machine after being filtered by a single-pole filter. The filter had a time constant of 160 s, emulating the fastest thermal response of any machine component. This ±0.0002°C temperature stability is typical and is independent of the 1.5°C variations in chilled water temperature. Without the outer loop closed, the inner loop thermistor could be stabilized to ±0.00002°C, but the coolant temperature at the machine only indicated ±0.002°C stability. This demonstrates the sensitivity of the single-loop control system to chiller cycling, errors in the bulk temperature measurement, and to environmental heat loads occurring after the heat exchanger.

Design of an Air Conditioning System

Many of the principles of liquid-temperature control systems apply to air conditioning systems as well. However, air is more difficult to control than liquids because of its lower heat capacitance, and the larger volumes of coolant. Air cooling systems typically operate with higher heat loads than liquid systems because of the energy required to move large volumes of air. Air also does not convect heat as well as liquids, necessitating larger, more difficult to control heat exchangers. It is also more difficult to control the temperature of a machine using air showers, as opposed to oil showers. Careful attention must be given to the flow distribution over the machine, which must be uniform, of high velocity, and time invariant. Radiation heat transfer to the surroundings also degrades the temperature control of the machine.[7]

An example of a high-performance air-conditioning system at LLNL is shown in Fig. 7. The air is drawn in through a filter box and passes through a cross-flow, air-to-water heat exchanger. A centrifugal fan, driven by a 25 HP motor, is placed after the heat exchanger to deliver an air flow of up to 570 m^3/min. This fan is the largest heat load in the system, raising the air temperature by 1°C. As a consequence, only cooling is required throughout the year. A duct connects the fan to a 0.5-m deep plenum covering the entire ceiling of the 6- by 6-m machine enclosure. With a nominal velocity of 15 m/min over the machine, the flow of 20°C air is laminar, and is also comfortable for the operator. The air exits through gratings at the perimeter of the enclosure. This enhances cooling of the heavily insulated walls and minimizes the thermal coupling of people to the machine. A raised floor allows the air to pass into a pit below the machine, from which it returns to the filter box through a stairwell. The enclosure is slightly pressurized to avoid inward leakage of outside air, and it force cools the fluorescent ceiling lights with air leaving the machine enclosure. Make-up air is drawn into the system from a point near the stairwell.

The variable-flow, chilled-water supply for the air-to-water heat exchanger is identical to the system shown in Fig. 3. The heat exchanger was designed to minimize its water volume, thereby decreasing its response time. The 1.2-m tall, by 1.5-m long, by 6.4-cm thick heat exchanger has 4 parallel aluminum fins per centimeter. These are attached perpendicularly to twentyeight 1.0-cm diameter copper tubes running down and back through the fins. This two-pass heat exchanger design tends to even out the temperature gradient across its face. The ends of these tubes are connected to a supply and return manifold, respectively. The large number of small-diameter tubes encourages uniform heat transfer. Locating the fan after the coil serves to mix the air flow, and compensates for any uneven cooling that may result from uneven air flow across the heat exchanger. A cascaded control technique similar to that shown in Fig. 5 is used. The air temperature for the inner loop is measured at a single point in the 1.2- by 1.5-m duct downstream of the fan. The large duct cross section makes bulk temperature measurements more difficult. The air temperature for the outer loop is measured in the machine enclosure directly above the machine. PI controllers are used for both the inner and outer loops, but the inner-loop bandwidth is only 0.01 Hz, because of the large heat exchanger. The performance of the air system is measured with 10 thermistors distributed throughout the machine enclosure. The unfiltered, 10-thermistor average over 36 hours is shown in Fig. 8, demonstrating the ±0.002°C temperature stability of the system.

Conclusions

This paper highlights the practical techniques and principles of obtaining precision temperature control in liquids and air. These include:

(1) Minimizing heat load variations and decreasing their rate of change
(2) Increasing the thermal response time of critical machine components
(3) Using large flowrates of coolant
(4) Selecting the proper variable-flowrate heat exchanger for the application
(5) Using a method of flow control that is appropriate to the application
(6) Encouraging mixing to improve bulk temperature measurements of the machine coolant
(7) Using a proportional plus integral action controller for improving temperature control precision
(8) Employing cascaded control loops for improving disturbance rejection

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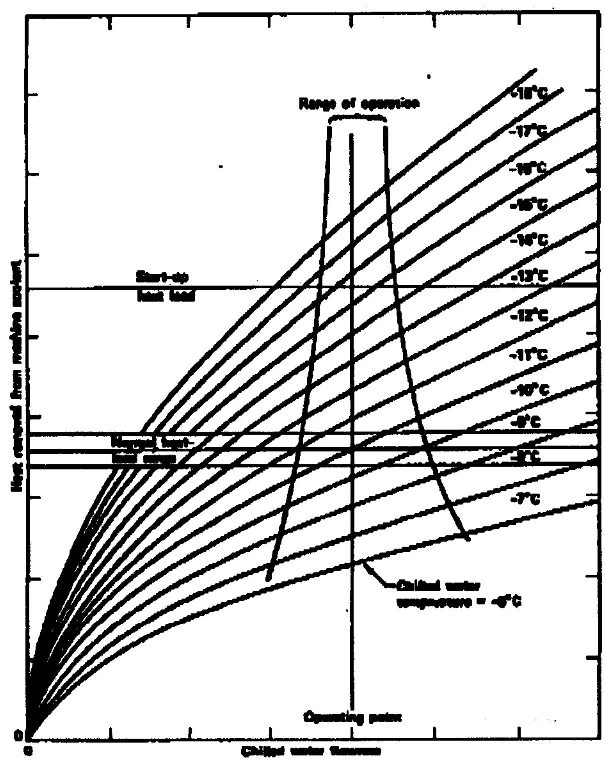
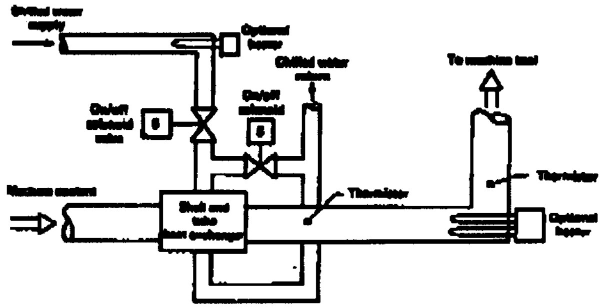
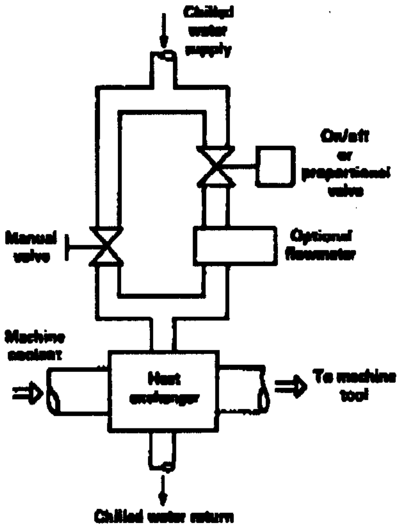


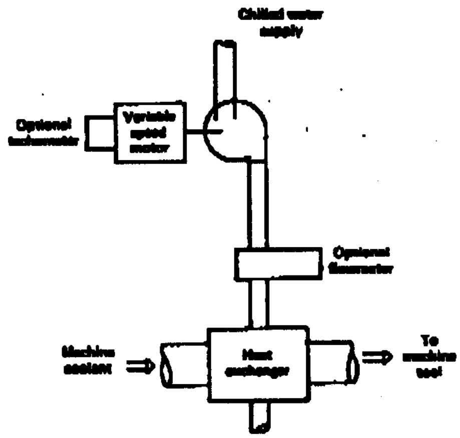
Fig. 1. Steady-state heat transfer characteristics of a heat exchanger.



a) Most common technique.



b) More precise technique.



c) High precision technique.

Fig. 2. Temperature control techniques using heat exchangers.

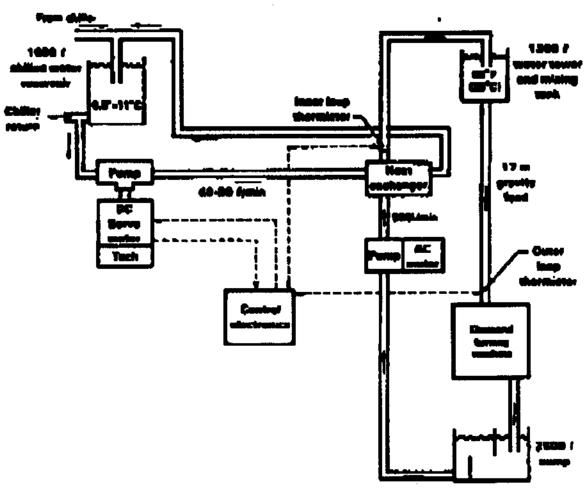


Fig. 3. A water temperature control system.

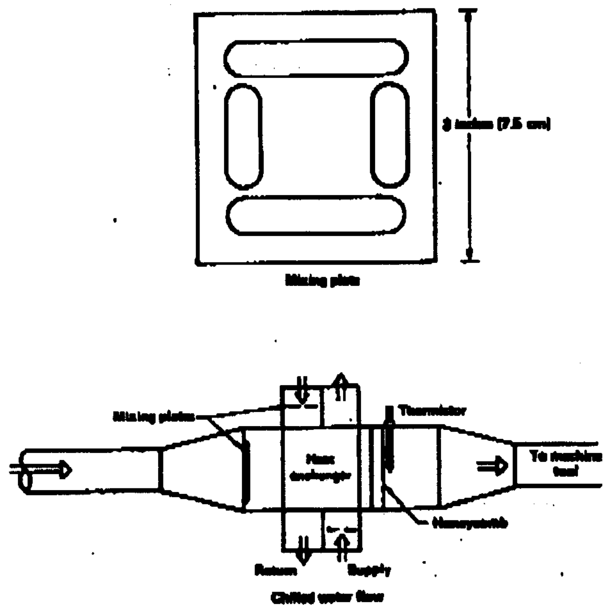


Fig. 4. Schematic diagram of water flow paths through the heat exchanger.

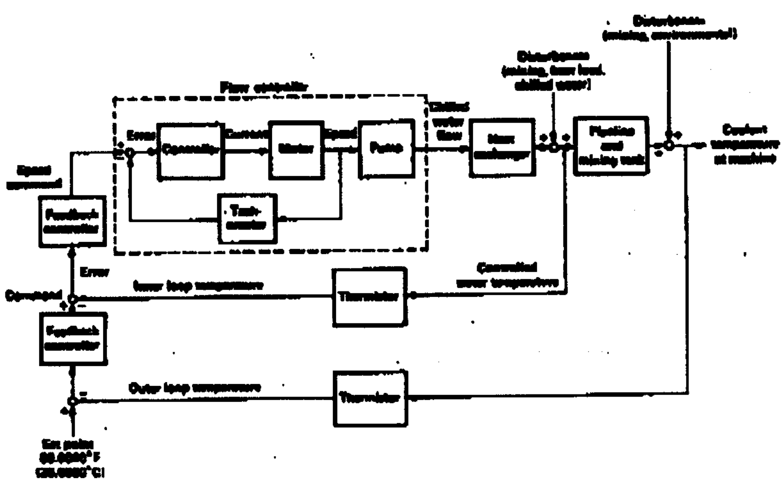


Fig. 5. Block diagram of the feedback control loops.

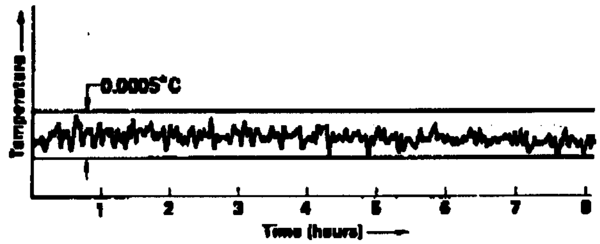


Fig. 6. Stability of the water temperature control system.

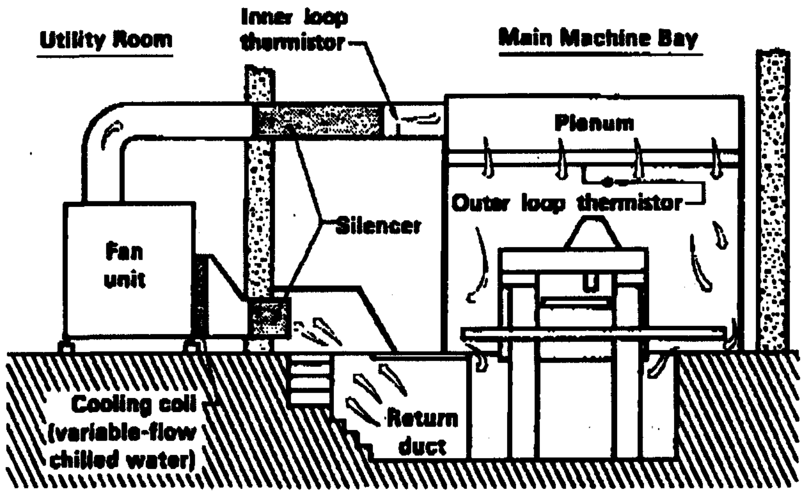


Fig. 7. A machine enclosure and air conditioning system.

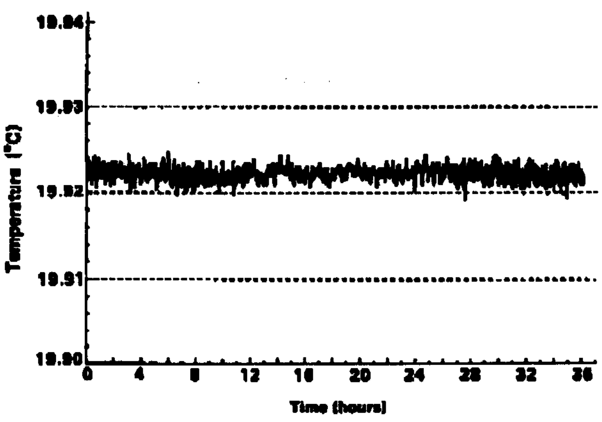


Fig. 8. Ten thermistor average temperature in the machine enclosure.

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TECHNIQUES FOR PRECISION AIR TEMPERATURE CONTROL

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Abstract

High quality air temperature control can provide an excellent means for minimizing the thermal drift of machine tools and inspection instruments when other means are not practical. Improvements in air temperature control has been improved in several machining areas by as much as ten to one by the careful identification of heat loads and the application of some fundamental classical control theory.

Introduction

The kinds of errors which govern the accuracy to which a machine tool can produce a part are usually broken into two categories: repeatable and nonrepeatable errors. Donaldson (1.) has reported that frequently the repeatable errors when compared to the nonrepeatable errors are very small, and as a result, large improvements in machine performance can often be obtained by trying to decrease the nonrepeatable errors. He also indicates that the most common cause of nonrepeatability is changing temperature and the consequent thermal deformation of the machine structure.

Thermal distortion has long plagued the areas of precision machining and gaging. Past solutions to the problem have included careful scheduling of processes with respect to day/night ambient temperature variations, construction of subterranean inspection facilities (2.) and machine temperature control by liquid flooding. In many cases the thermal distortion problem could have been minimized adequately if air temperature could have been controlled well enough. Examples of successful applications of flood cooling for temperature control have been proven by Bryan at LLNL on Diamond Turning Machines #1 and #2 (3.) and on a modified Moore #3 measuring machine which was improved in thermal stability by a factor of nearly 10 to 1 (4.). These solutions, although sufficient have caused both inconvenience in production and increased cost of both the product and the facilities required. In addition, many of the solutions have utilized on-off controls and have been consequently sensitive to heat load changes. An alternate approach provides for the use of both liquid and air temperature control. In this scheme, closed liquid cooling is used to remove localized heat from the machine, while a light air shower is used to maintain the temperature of the superstructure. Localized heat would include heat produced by the spindle drive motor and slide drive motors. The reason that this approach is sometimes desirable is that if there are heat loads on the machine which are substantial, high air velocities are required to provide the necessary heat transfer if air alone is used. The high air velocities can be unpleasant as well as unhealthy for the operator. The combined approach is being implemented on several precision turning machines at LLNL.

The main thing that makes precision air temperature control appear to be difficult is that insufficient attention is paid to the analysis of the heat sources present, the dynamics of the temperature control system and to the mode of control implemented. Most air temperature control systems are designed on static criteria such as maximum heating and cooling loads with no regard to whether the loads are steady or variable. Very little attention is ever paid in many cases to the type of controller used since the majority of systems are not designed for high precision applications and as a result simple on-off type controls will suffice.

Heat load considerations

A machine tool will be at static thermal equilibrium as long as nothing which affects the net heat transfer to it changes. Things which supply or remove heat from a machine at a constant rate will not cause the temperature distribution of the machine to change after it has been allowed to reach thermal equilibrium in the presence of those sources or sinks. On the other hand, things which supply or remove heat at variable rates will inevitably cause the temperature distribution to vary in time with a resulting thermal drift.

The primary conclusion is that it is not necessary to worry about constant heat sources or sinks beyond ensuring that the static heating or cooling capacity of the room is adequate to handle the net heat input or removal from the room. The secondary conclusion is that if a constant temperature distribution is to be maintained throughout the machine, variable sources or sinks must be either removed or their effects reduced to a level which produces a tolerable change in machine temperature distribution.

Constant sources of heat typically include things like CNC units, lights (which are left on day and night) and any other equipment within the same space as the room which operates constantly. Variable sources include people, open doors, varying wall temperatures, varying air temperature, and machine drive units which operate at varying speeds. A well designed air temperature control system should be able to provide air at sufficiently constant temperature, and minimize the effects of personnel, varying wall temperatures and open doors.

A useful technique for representing heat load information is the generation of a heat load table for the space occupied by the machine in question. It is a list of all of the heat loads present and an indication of their nature (i.e. steady or variable). In addition it includes totals for heating and cooling loads which are useful for determining necessary system capacity. An example for a typical machining area is given in Table 1.

Table 1. Sample Heat Load Table

		Winter			Summer		
	Load Type	Latent	Sensible	Total BTU/HR	Latent	Sensible	Total BTU/HR
Conduction To Inside Spaces	DN	0	0	0	0	0	0
Conduction Thru North Wall	DN	0	-15092	-15092	0	14063	14063
Conduction Thru Roof	DN	0	-31040	-31040	0	69134	69134
Conduction Thru Floor	S	0	0	0	0	0	0
Personnel Heat Load	DN	6132	3000	9132	6132	3000	9132
Air Handler 7-1/2 BHP	S	0	19100	19100	0	19100	19100
Infiltration	S	0	-10780	-10780	0	0	0
Make-Up Air (7000 SCFM Max)	V	0	-332640	-332640	0	279720	0
Lighting	S	0	28700	28700	0	28700	28700
Machine Tools	V	0	12804	12804	0	12804	12804
	S	0	59633	59633	0	59633	59633
TOTALS		6132	-266315	-260183	6132	486154	492286
		6132	*66325	*72457	6132	*206434	*212566

* Includes Effect of Separate System for Makeup Air

The idea of making a heat load table of this kind is not uncommon to the HVAC profession; however, most tables do not present the nature of the loads. Time history information is useful in the design of the control system for determining the kind of disturbance rejection which will be necessary. In addition the results of the heat load table will indicate what modes of conditioning are required, whether heating or cooling or both. It is desirable from a controls point of view to only have heating or cooling since it is difficult to find controls for precision applications which can control both processes.

Control system considerations

There are two common control problems one or both of which many systems are plagued with. The first problem is steady oscillation of air temperature with respect to time. The most common cause of this is the on-off controllers frequently used. Their very nature is destabilizing from a controls point of view. To be more specific, the on-off action introduces a non linearity to the system which usually results in a limit cycle condition. Almost all heat transfer devices are nonlinear to begin with and an on-off controller just makes the whole system more so. Most adjustments made to the system will only vary the amplitude and frequency of the oscillation. The discussion should not be construed as that no oscillation of any kind is acceptable. McClure (5.) demonstrated that machine tools do demonstrate a thermal frequency response, and that for ambient temperature variations above some frequency characteristic of the machine, it's response in terms of drift is negligible.

The second problem which appears quite often in air temperature control systems is a day/night temperature variation. This is an indication that the system has a relatively poor disturbance rejection at low frequency, and more obviously, that some low frequency disturbance is present. The changes in disturbance are caused by both changes in outside air temperature and also by changes in the temperatures and pressures of the cooling and heating fluids used in the control system.

Both of the above problems can often be dealt with by being more attentive to the interaction of the controller with the heating and/or cooling device and in general by being more careful in the selection of a controller. In the case of the heating and/or cooling devices it is desirable, whenever possible to consider not only their static heat transfer characteristics, but their dynamic characteristics as well. Sometimes it leads to devices which are inherently easier to control.

Controllers in most cases need not be much more sophisticated than to have proportional and integral actions although it usually costs no more once proportional and integral are present to get derivative action as well. Proportional action is linear in contrast to on-off action and the integral action is all that is normally required to remove day/night temperature variations given adequate system capacity. With the advent of the microprocessor, many control manufacturers are offering digital controllers with PID action, suitable for air temperature control systems for less than $1000.

Application of the principles

BODTM enclosure

The BODTM (Baby Optics Diamond Turning Machine) (6.) is a small t-based machine which has shown the capability of producing parts to 20 microinch size and contour tolerances using post-process gaging with feedback. A major factor contributing to the capability of the machine is its high repeatability which is made possible by minimizing thermal distortion. Two efforts are made to control the distortion. One is the application of closed flow cooling to the spindle drive motor and the other is maintaining good control of the ambient air temperature. The machine was originally housed in an inspection laboratory with ± .20° F temperature control. When programmatic needs demanded that the machine be moved to a shop with ± 2.0° F air temperature control, the problem of how to provide adequate air temperature control materialized.

In response, a small temporary enclosure was constructed to house the machine which would be capable of providing air temperature control commensurate with what existed in the machine's prior habitat. The control scheme involved drawing air from the shop which had ± 2.0° F variations and heating it at variable rates in order to provide the necessary control accuracy. The resulting conditioned air was blown over the machine and then returned to the surrounding shop. Figure 1 shows the configuration of the enclosure.

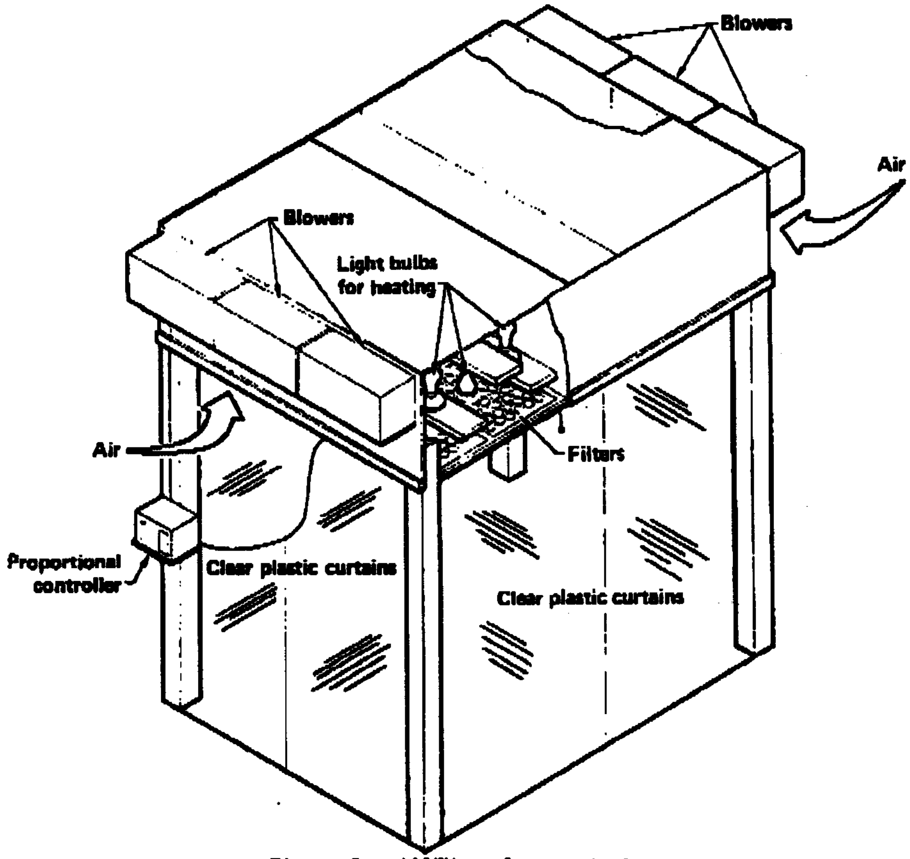

Figure 1. BODTM enclosure design

The air temperature was heated by arrays of standard 100 W light bulbs which received varying amounts of power depending upon the output signal of the temperature controller. The latter was a basic commercial proportional control which had as it's input a thermistor type temperature probe located within the enclosure near the slides of the machine. Light bulbs were chosen for their fast thermal response as well as availability.

Several efforts were made in order to attempt to reduce the effects of the surroundings on the temperature within the enclosure. The first thing that was done was to equip the enclosure with transparent plastic curtains to contain the conditioned air and minimize the infiltration of outside air. The net positive pressure provided inside by the blowers also helped to inhibit the entry of outside air. The second thing which was done was to design the enclosure to limit the entry of the operator to the minimum requirement of arms and hands, and hence reduce the amount of variable heat contribution by the operator.

The enclosure, although crude, is capable of maintaining the air temperature inside to ± 0.3° F while the temperature in the surrounding shop varies as much as ± 2.0° F. A sample temperature recording from within the enclosure is given in Figure 2. The main caveat of the design is that since the enclosure can only provide heating, the temperature inside can only be held higher than the outside shop temperature. The consequences of this are twofold. The first is that if the parts are inspected in an area with a different temperature, size discrepancies will no doubt be present. However, this problem can generally be accommodated by simply taking into account the coefficient of thermal expansion of the part material and the temperature difference between the machine and inspection environments. The second consequence is that since the enclosure temperature is linked to the outside shop temperature, if the latter should depart widely from the set point temperature within the enclosure, control will be either lost or severely degraded. On a more positive note, the simplicity of the design provided for a rapid, low cost (under $5000) solution for the temperature control problem on BODTM. In terms of reliability, downtime of the enclosure has been less than one week in the one year of continuous operation since it's completion.

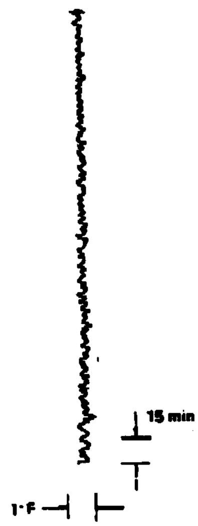

Figure 2. Sample temperature recording from BODTM enclosure

PERL enclosure

The Precision Engineering Research Lathe (PERL) now under construction at LLNL is a small turning machine similar in many ways to the aforementioned BODTM machine. One of the major differences between the two machines is that the PERL has a design accuracy an order of magnitude higher than BODTM's. When it is finished it is slated to operate in the same shop as the BODTM and therefore some means will be required to maintain the air temperature around it at constant temperature. The design air temperature variation for the machine is specified as ± 0.25° F.

During the design of the enclosure all of the pitfalls of the BODTM enclosure were considered and the following decisions were made to improve the subsequent design:

1. The dependence of the enclosure on outside shop air temperature would be decreased.

2. The enclosure would be configured so that the operator would not have to work through curtains and would not be exposed to the flow of escaping enclosure air past his or her face.

3. More emphasis would be placed upon the aesthetics of the enclosure to improve acceptance by operators.

The resulting enclosure features closed air flow and is designed so that the operator can stand inside next to the machine. Construction features extruded aluminum structural members and formica covered panels. Some wood was used in the design for it's thermal insulation and noise attenuation characteristics. The walls of the enclosure are made of plastic curtains similar to what one commonly finds on refrigeration cases in supermarkets. They provide for ease of operator access while inhibiting the infiltration of loosely controlled outside air. The design of the enclosure is depicted in Figure 3.

Air is distributed evenly over the top of the machine by two large plenums with face areas almost as large as the area of the enclosure itself. The air is returned through a U-shaped return duct which exists along both sides of the machine and across it's back. A 1/2 HP blower draws the air at the rate of about 700 CFM and forces it through a conventional furnace filter and through a small water to air heat exchanger before the air arrives at the supply plenums.

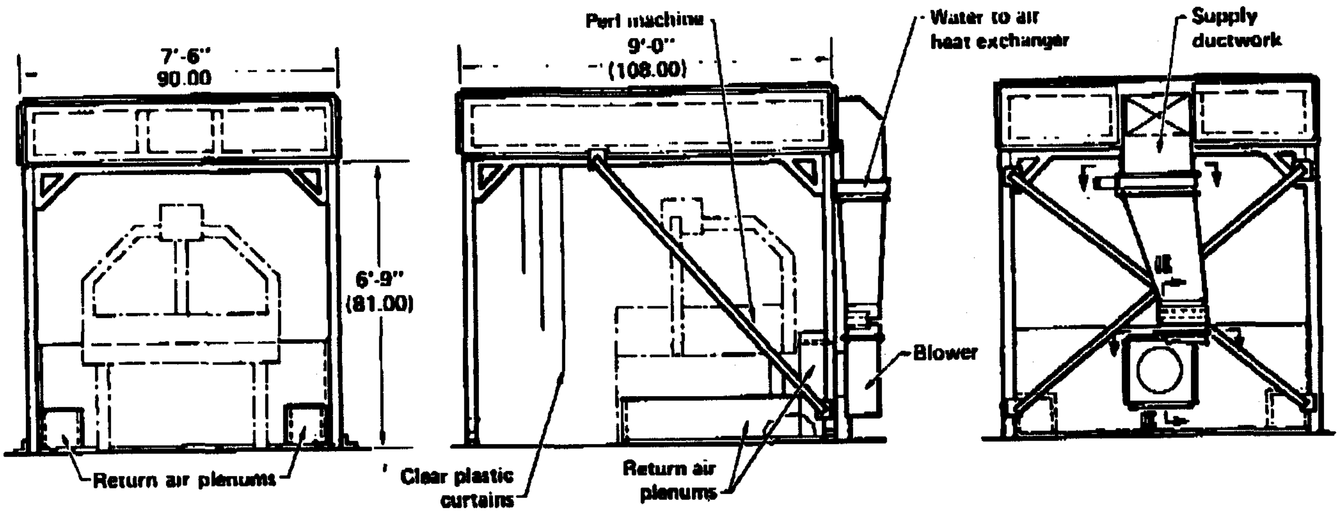

Figure 3. PERL enclosure design

The temperature of the air is sensed by a thermistor type sensor located near the machine slides. The signal from the sensor, after conditioning, is sent to a commercially available digital PID controller. The controller has a time proportioning output and operates solenoid valves which in turn modulate the flow of 50° F chilled water through the water-to-air heat exchanger. A schematic diagram showing the interconnections of the sensor, conditioner, controller, valves and the heat exchanger is shown in Figure 4. Two solenoid valves are used to first ensure that fresh cold water is always available right at the exchanger and second to minimize the effect of water hammer.

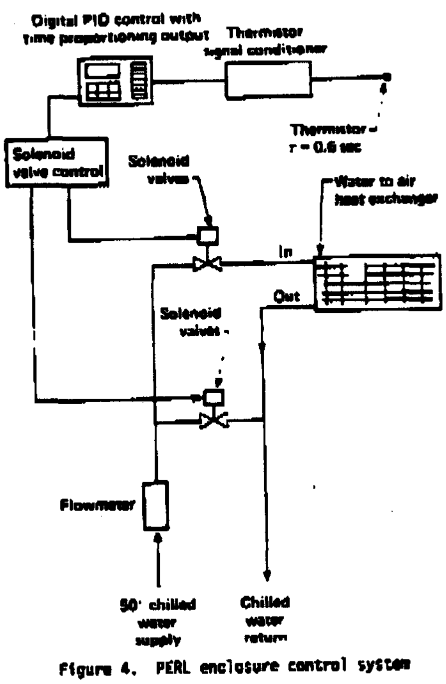

Figure 4. PERL enclosure control system

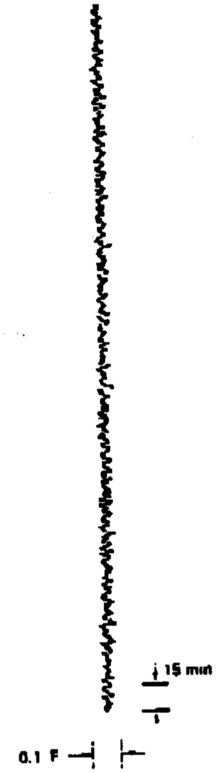

Figure 5. Sample temperature recording from the PERL enclosure

Proportional and integral actions have been used on the control to provide ± 0.08° F air temperature control with no visible day/night variation. A sample temperature recording is given in Figure 5. The variation in temperature has an oscillatory nature which appears at this time to be as a result of the nonlinearity introduced by the solenoid valve action. Efforts are now under way to find a low cost means of modulating the flow with less introduction of nonlinearity.

CBN room

The Cubic Boron Nitride (CBN) machine is a t-based precision turning and boring machine currently under construction at LLNL. It has a design disc check accuracy on the same order as LLNL's existing diamond turning machines but is designed to be stiffer and hence more suitable for use with cubic boron nitride cutting tools. The machine is built around the base of a 920 EXCELLO for an indication of the size of overall machine size.

Due to the accuracies involved, close air temperature control was deemed necessary. The design control level was 68 $\pm$ 0.1° F. Since the machine was fairly large it was more reasonable to put it in a room with good temperature control as opposed to constructing an enclosure around it as in the cases of BODTM and PERL. No room with the necessary temperature control existed within MFD's machining complex, so one was specially designed.

The room was designed with the objective of providing close air temperature control by minimizing the effects of variable heat sources. Since the variable heat sources on the machine all have closed flow liquid cooling, the primary sources of variable heat remaining are conduction and radiation from the walls, heat generation by the operator and infiltration of air by leaks and the makeup air system.

In order to minimize the contribution from the walls supply and return registers were constructed around the perimeter of the room and adjusted to provide a relatively high velocity curtain of air around the perimeter. Velocities are on the order of 100-150 FPM along the walls. Air is also distributed over the top of the machine by two large plenums with exit areas slightly larger than the projected area of the machine. Air velocities out of the plenums averages between 30-40 FPM. This air returns along with the high velocity perimeter air through the perimeter registers. The air distribution scheme is shown in Figure 6.

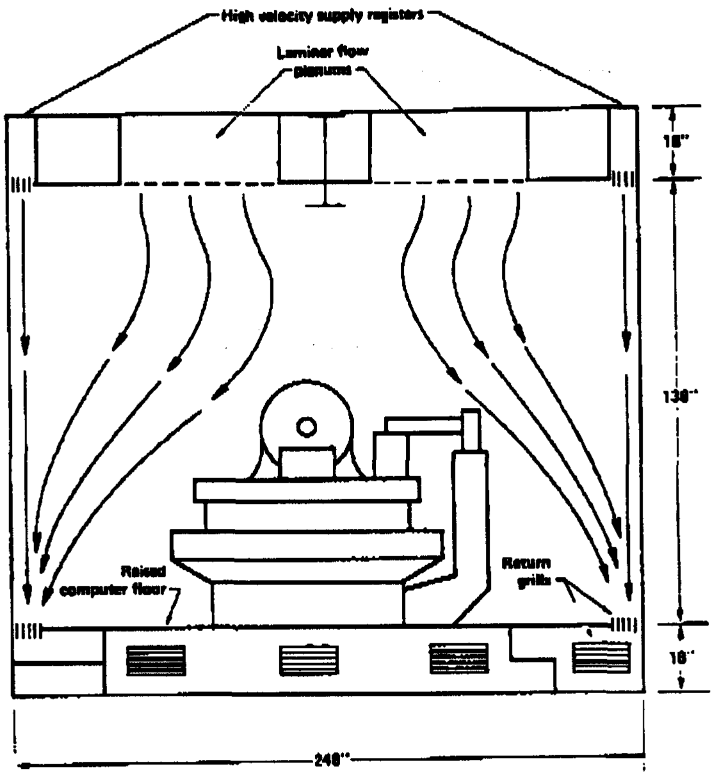

Figure 6. CBN room air distribution scheme

A unique feature of the design is the configuration of the return ducting. The height of the machine spindle centerline made it necessary to construct a raised floor around the machine to facilitate operator access. The floor is actually a computer floor and provides for about 18 inches of clearance between the bottom of the raised floor and the floor on which the machine rests. The clearance space serves as the return duct, providing some cost savings in sheet metal work.

The control scheme implemented is the same as that implemented on the PERL enclosure with the exception that the air flow is increased to 10,000 CFM. Only cooling is required since there are enough heat sources present throughout the year to maintain 68° F.

Thus far the system has been capable of providing 68 ± 0.05° F control for the 6 months that it has been in operation. A sample temperature recording is given in Figure 7. The later temperature variation produces a ± 2.0 microinch thermal drift between the two slides of the machine measured at tool height, even when two or three machinists are entering and leaving the room. Closed flow liquid cooling around the base of the machine is expected to reduce the drift even further.

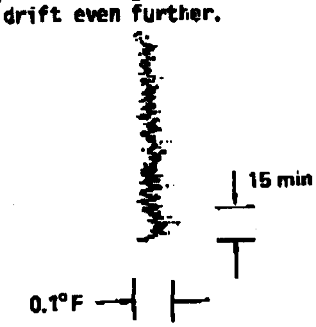

Figure 7. Sample temperature recording from CBN room

Numerical control shop upgrade

Long range planners within MFD have projected that in order to continue to meet programmatic needs, the air temperature control in the shops which house it's precision turning and boring machines will have to be improved from its current ± 2.0° F to ± 0.25° F. A study was conducted in order to determine the best way to meet the almost order of magnitude improvement requirement. The study included preparing a heat load table for the shops, identification of the current control scheme, reviewing present air temperature control technology and conducting some experiments with the current system.

A brief review of the existing system revealed that the system operated as follows: Air temperature was sensed by a thermistor type sensor located about 20 feet above where the turning machines are located. The signal was fed into an on-off controller with about 0.3° F of deadband. The controller switched a solenoid valve which modulated the flow of air to a pneumatically operated chilled water valve. The chilled water valve was used to modulate the flow of chilled water through a water-to-air heat exchanger located in ductwork on the roof of the shop. The heat exchanger was used to modulate the air temperature of the shop below.

A simple experiment which was conducted was to replace the on-off control and replace it with a digital PID control the same as those used in the PERL enclosure and in the CBN room. Without any other modification, the air temperature fluctuation in the shop went from ± 2.0° F to ± 0.5° F. The total cost of the change was under $1000.

Another discovery resulting from the study was that there was a 60 second transport delay between the heat exchanger and the temperature feedback sensor. This is a destabilizing effect which meant that in order to obtain system stability, the loop gain of the system had to be decreased. The result was that disturbance rejection had to be compromised and therefore the quality of the temperature control was limited to ± 0.5° F.

In order to meet the temperature control requirement of ± 0.25° F a control system is being built for the shop which has two temperature feedback probes. The first feedback probe will be located only a few feet downstream from the heat exchanger and the second will be located down near the turning machines. The control scheme is shown in block diagram form in Figure 8. The purpose of the first sensor and its loop will be to provide fairly tight control over the immediate outlet air temperature from the heat exchanger. The purpose of the second sensor and loop is to detect changes in the temperature of the air as it travels from the first sensor to the second. Measurements made have shown that the latter temperature changes occur fairly slowly and hence will not be that difficult for the second sensor loop to correct for.

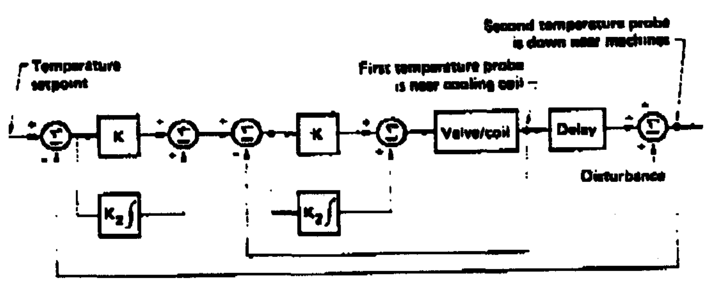

Figure 8. Control scheme for NC shop upgrade

The control system is being configured around a small personal computer, not because the control scheme requires significant computing power, but because it will also be used to log temperatures at various locations in the shop simultaneously. The latter function will be performed so that the performance of the air temperature control system can be continually and conveniently observed by shop supervisors. The system will be capable of displaying temperature time histories graphically on its monitor at the discretion of whoever is operating the keyboard. It will also be able to store several days worth of temperature data on disk. A layout of the system is shown in Figure 9.

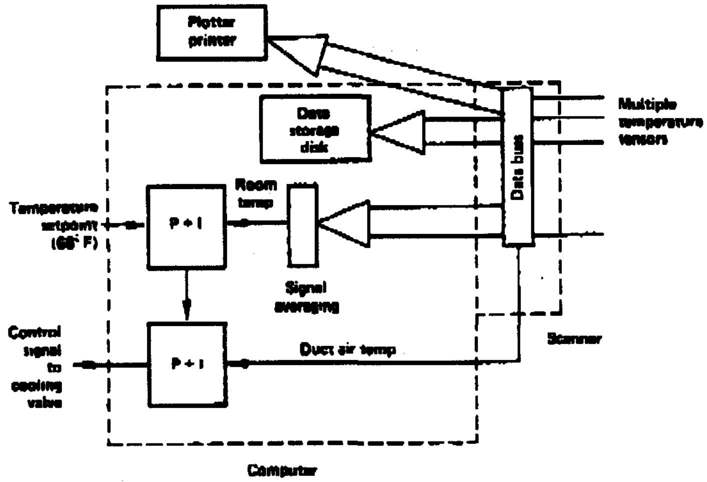

Figure 9. Computer based temperature control and data logging system

Conclusions

Precision air temperature control can be an excellent way of improving the accuracy capability of machines by removing one of the largest sources of nonrepeatability. Work at LLNL has demonstrated that achievement of close temperature control can be obtained by careful identification of heat sources and the application of some fundamental classical control theory. The work has also demonstrated that large improvements in the performance of temperature control systems do not necessarily have to be complicated or expensive.

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