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VARIABLE PRIMARY FLOW CHILLED WATER SYSTEMS: POTENTIAL BENEFITS AND APPLICATION ISSUES

Final Report
Volume 1

March 2004

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EXECUTIVE SUMMARY

The use of variable primary flow pumping (variable flow through chiller evaporators) in chilled water systems is increasing due to its perceived potential to reduce energy consumption and initial cost relative to more conventional pumping arrangements. Neither the conditions under which significant energy savings are realized nor the likely magnitude of savings are well documented.

To characterize current thinking on the use of variable primary flow chilled water systems, literature review; surveys of designers, owners, and chiller manufacturers; and additional correspondence were synthesized into a composite portrait of prevailing practices and attitudes.

To quantify the energy use and economic benefits of variable primary flow, an extensive parametric simulation study was conducted that compared variable primary flow system energy use with that of other common system types. System types included in the study were constant flow/primary-only, constant primary flow/variable secondary flow, and primary/secondary with a check valve installed in the decoupler. Parameters varied included load type, number of chillers in the central plant, temperature difference vs. part load characteristics, and climate.

State of the Art Review Findings

There is growing support for variable primary flow among chiller manufacturers and system designers, owners, and operators. Modern chiller controls are capable of practical variable primary flow operation. Advances in capacity controls, freeze protection, and flow detection have increased chiller stability—a particular concern in variable primary flow applications because evaporator flow rates can change abruptly during chiller staging. Manufacturers are providing more detailed variable flow application guidance than in the recent past, including recommended chilled water tube velocity ranges and maximum rates of flow variation for most chiller models.

Variable primary flow systems are perceived to be more complicated than comparable primary/secondary systems. This is partly because chiller staging requires more care in order to achieve stable operation and realize anticipated energy savings. Chiller isolation valves should open and close at a rate that is consistent with the response time of the chiller's capacity control. The low flow bypass control required in most variable primary flow systems adds further complexity. The bypass and valve should be sized for the minimum required flow rate of the largest chiller and should be located close to the plant. Flow measurement devices must have sufficient turndown to measure flow throughout the anticipated range.

Over half of the survey respondents had designed or operated variable primary flow systems. Those who had no variable primary flow experience identified lack of guidance as a key reason why they had not. Owners cited reduced operating costs, lower first cost, smaller space requirement due to fewer plant components, and ability to improve chiller loading in systems experiencing low chilled water ÷T as advantages of variable primary flow systems over primary/secondary systems. While most claims of variable primary flow superiority over other system alternatives revolve around energy and first cost savings, there is little quantitative evidence in the open literature. Most arguments in favor of variable primary flow are anecdotal. Designers and system owners with variable primary flow experience generally are willing to consider the use of variable primary flow for future projects.

Parametric Study Findings

Variable flow, primary-only systems reduced total annual plant energy by 3 to 8-percent, first cost by 4 to 8-percent, and life cycle cost by 3 to 5-percent relative to conventional constant primary flow/variable secondary flow systems. Several parameters significantly influenced energy savings and economic benefits of the variable primary flow system relative to other system alternatives. These included the number of chillers, climate, and chilled water temperature differential. The following factors tended to maximize variable primary flow energy savings relative to other system alternatives:

- Chilled water plants with fewer chillers
- Longer, hotter cooling season
- Less than design chilled water temperature differential

Load type had little impact on variable primary flow energy savings. The magnitude of savings was much larger for greater cooling loads, but when savings were standardized on a per design ton basis the differences were relatively small.

Chilled water pumps and chiller auxiliaries accounted for essentially all savings. Differences in chiller energy use were not significant from system type to system type. Variable flow, primary-only systems chilled water pump energy use was 25 to 50 percent lower than that of primary/secondary chilled water systems. In systems with two or more chillers configured in parallel, chiller auxiliary energy savings were 13 percent or more relative to primary/secondary.

The addition of a bypass check valve to the constant flow primary/variable flow secondary system resulted in total plant energy savings of up to 4 percent and a life cycle cost savings of up to 2 percent. Savings occurred only when chilled water ÷T's were less than the design value. Chilled water pump savings were 5 percent or less and chiller auxiliary savings were 13 percent or less.

Conclusion

In view of both the state-of-the-art review and parametric study results obtained in this project, it can be concluded that variable primary flow is a feasible and potentially beneficial approach to chilled water pumping system design. However, the magnitude of energy and economic benefits varies considerably with the application and is obtained at the cost of more complex and possibly less stable system control. The literature on effective application of variable primary flow is growing and should promote its appropriate and effective use in the future.

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ABBREVIATIONS AND ACRONYMS

BTU	British thermal unit
CAPFT	Fraction of available to design capacity
CHW	Chilled water
CHWP	Chilled water pump
CHWR	Chilled water return
CHWS	Chilled water supply
EIRFT	Fraction of full load to rated power consumption
EIRFPLR	Part load performance
FT	Feet
GPM	Gallons per minute
HP	Horsepower
H	Hour
kW	Kilowatt
kWh	Kilowatt hour
LCC	Life cycle cost
LBM	Pounds (mass)
MAX	Maximum
MBH	1,000 x Btu
MIN	Minimum
NA	Not applicable
O&P	Overhead and profit
PLR	Part-load ratio
RPM	Revolutions per minute
Tchw	Chilled water temperature
Tcw	Condenser water temperature
TEL	Total equivalent length
TON	12,000 Btu per hour
VFD	Variable frequency drive
VPF	Variable primary flow
VS	Versus

NOMENCLATURE

c_p =	Specific heat, $\left(\frac{Btu}{lbm \cdot {}^\circ F}\right)$
÷T =	Chilled water temperature difference, ˚F
ξ_{drive} =	Drive efficiency
ξ_{motor} =	Motor efficiency
ξ_{pump} =	Pump efficiency
F_{HP} =	Fraction of nameplate motor horsepower
F_N =	Fraction of nominal speed
$H_{control}$ =	System control pressure, ft of water
H_{design} =	System design pressure, ft of water
H_{pump} =	Pump head, ft of water
$H_{pump,nom}$ =	Pump head at nominal pump speed, ft of water
H_{system} =	System operating pressure, ft of water
HP_{pump} =	Total pump power, hp
q |	Heat transfer rate or cooling load, Btu/h
$\dot{Q}$ |	Chilled water flow rate, gpm
$\dot{Q}_{design}$ =	System design flow rate, gpm
$\dot{Q}_{pump}$ =	Pump flow rate, gpm
$\dot{Q}_{pump,nom}$ =	Pump flow rate at nominal speed N_0, gpm

Q_{ref} = Capacity at the reference evaporator and condenser temperatures where the curves come to unity, tons

$\dot{Q}_{system}$ = System operating flow rate, gpm

R = Condenser water range, ˚F

ψ = Water density, lbm/ft^3

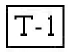
T_{chws} = Chilled water supply temperature, ˚F

T_{cw} = Entering condenser water temperature, ˚F

T_{cws} = Condenser water supply temperature for water-cooled, ˚F

T_{wb} = Ambient wet-bulb temperature, ˚F

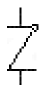
SYMBOLS



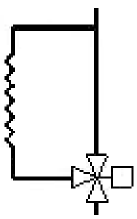
Temperature sensor

Differential pressure sensor

Pump

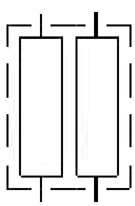


Check valve



Cooling coil and three-way control valve

Cooling coil and two-way control valve



Water-cooled chiller

1. INTRODUCTION

Chilled water systems provide cooling for many air-conditioning and industrial processes. Regardless of size or complexity, every chilled water system is comprised of cooling loads, cooling equipment, a distribution system, pumps, and control valves. Heat is added to a circulating stream of water by cooling coils, radiant panels, process heat exchangers and other loads and is removed by cooling equipment such as chillers, heat pumps, or heat exchangers. The distribution system is a piping network that transfers chilled water between loads and cooling equipment at rates determined by pumps and control valves.

Chilled water system loads may have both sensible and latent components, but heat transfer within the chilled water system is purely sensible. The rate of heat transfer is proportional to both the flow rate and the temperature rise of chilled water as it passes through a load (Equation 1-1).

$$q = \rho \cdot c_p \cdot \dot{Q} \cdot \Delta T \qquad (1\text{-}1)$$

where:

$q =$ heat transfer rate or cooling load, Btu/h

ρ = water density, lbm/ft^3

c_p = specific heat, Btu/lbm-°F

$\dot{Q} =$ chilled water flow rate, gpm

ΔT = chilled water temperature difference, ˚F

The relationship between flow and the temperature differential in Equation 1-1 is of critical importance. The smaller the temperature differential that exists across a cooling load, the larger the flow rate required to meet the load. Flow rate requirements determine the size of components in a chilled water system as well as the amount of pumping energy a system consumes. The chilled water supply temperature for air-conditioning loads is typically between 39 and 45 degrees F and the design temperature difference is typically on the order of 10 to 20 degrees F.

1.1 Chilled Water System Types

Cooling loads are often highly variable. In order to track changes in cooling load, chilled water systems must respond by varying chilled water flow rate, chilled water temperature differential, or both, in accordance with Equation 1-1. Typical chilled water system design practice is based on either a constant flow/variable temperature difference or variable flow/constant temperature difference concept. These are called, respectively, constant volume and variable volume systems. Systems can also be classified according to the levels of pumping present. Systems with a single level of chilled water circulating pumps are called primary-only systems and those with both circulating pumps for chillers and distribution pumps are called primary/secondary systems. Some systems may have load circulators as well, which are sometimes called tertiary pumps.

A description of common chilled water system types and a historical account of the progression of chilled water pumping strategies from the most basic comprehensive constant flow to all-variable-flow systems follow. For purposes of discussion, chilled water pumping systems are divided into three categories:

- Constant flow chilled water systems
- Variable flow chilled water systems with constant evaporator flow (constant primary flow)
- Variable primary flow chilled water systems

In this section, a comparative overview of the system types is provided to highlight the essential differences between chilled water system types and indicate the reasons for their use.

1.1.1 Constant Flow Chilled Water Systems

A constant flow system is perhaps the simplest chilled water system type. A constant flow, primary-only system with two chillers in parallel is shown in Figure 1-1. A single set of constant speed pumps distributes water throughout the entire chilled water system. Three-way control valves at each load allow the chilled water that does not flow through the cooling coil to return to the chiller so that flow remains approximately constant (in reality, there is some degree of variation in total flow through a three-way valve as it modulates, but the flow variation in a properly balanced system will be small relative to that in "variable flow" systems discussed in sections 1.1.2 and 1.1.3). Constant flow though the evaporator ensures a stable chilled water supply temperature and prevents freezing in the evaporator tube bundles—a potential effect of sudden changes in flow rate.

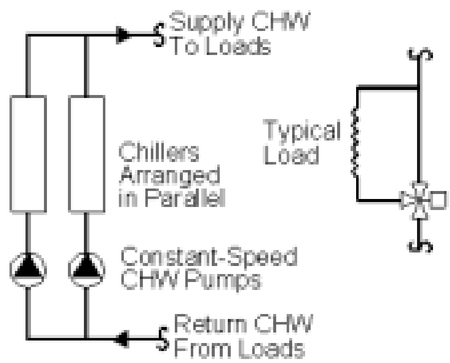


Figure 1-1: Constant flow, primary-only pumping arrangement with two chillers configured in parallel.

In a constant-flow parallel pumping system it may be difficult or impossible to stage chiller capacity without either increasing the chilled water supply temperature or adversely affecting three-way valve performance. If a chiller is de-energized while its pump continues to operate, warm return water will bypass through the chiller and mix with cold supply water, thereby raising its temperature. If both chiller and pump are de-energized, the resulting drop in chilled water flow rate may cause inadequate flow to some loads. For these reasons a series chiller configuration is preferred in most cases (Figure 1-2) because it permits the staging of chillers and associated auxiliaries (i.e., the cooling towers and condenser water pumps) with varying load while maintaining constant flow. However, increased pressure drop that results from having the additional chiller evaporator in series adds to pump energy consumption.

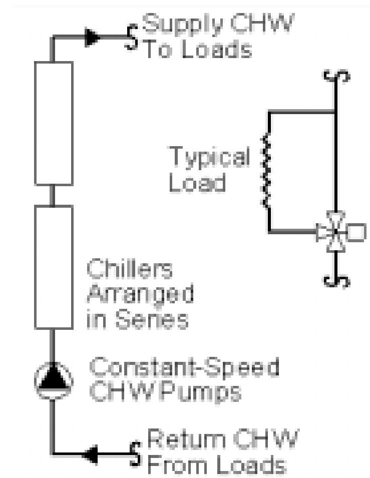


Figure 1-2: Constant flow, primary-only pumping arrangement with two chillers configured in series.

1.1.2 Variable flow chilled water systems with constant primary flow

The most common type of variable flow chilled water system combines a primary (plant) loop in which each evaporator receives a constant flow of chilled water with a variable flow secondary (distribution) loop.(Figure 1-3). Primary pumps are typically constant speed and the secondary pumps may be constant or variable speed. A decoupler pipe, also called a bypass, separates the primary and secondary loops. The flow through each evaporator on the primary side of the system is constant and the flow on the secondary side varies in response to the cooling load.

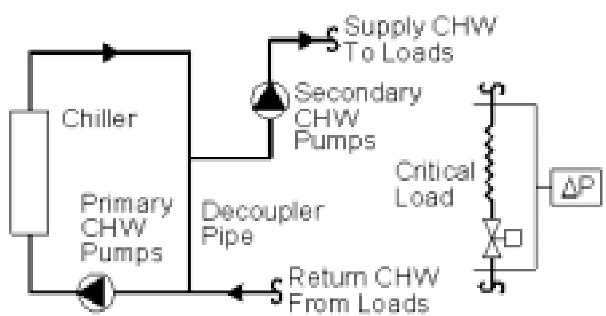


Figure 1-3: Constant primary flow/variable secondary flow pumping arrangement.

Other variations of the basic constant primary flow/variable secondary flow chilled water system include primary/secondary/tertiary systems, which have an additional level of pumping at the loads, and distributed or zone pumping arrangements, which have decentralized secondary pumps. System designers use economic and engineering criteria to determine which arrangement best suits the application.

1.1.3 Variable primary flow chilled water systems

An increasingly common way to apply variable primary flow is by using a single set of pumps equipped with variable frequency drives to serve both the production and distribution loops (Figure 1-4). This arrangement is called a variable flow, primary-only system. The function of the bypass line in Figure

1-4 should not be confused with that of the decoupler in Figure 1-3. The bypass in this case is a smaller pipe sized for the minimum flow of the largest chiller .It contains a normally closed control valve that modulates open only when the low flow limit is reached. Chilled water pumps equipped with variable frequency drives operate to maintain a minimum differential pressure at the critical load. Chillers may be staged on and off based on calculated cooling load.

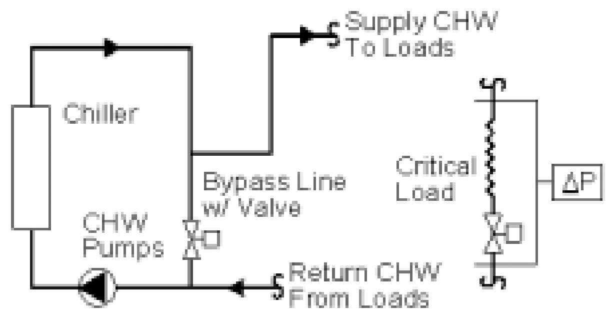


Figure 1-4: Variable flow, primary-only pumping arrangement.

Other variable primary flow arrangements that have been used include primary/secondary with variable speed primary pumps and primary/secondary with a check valve in the decoupler. The schematic of the variable flow primary/variable flow secondary system resembles the constant flow primary/variable flow secondary system discussed previously (Figure 1-3). However, in the former, the large decoupler pipe is replaced by a small bypass and normally closed control valve (Figure 1-5). Otherwise, the primary pumps will track the secondary flow by minimizing the flow through the decoupler. In addition, the operation of the system is such that chillers are staged in a manner that optimizes chiller energy.

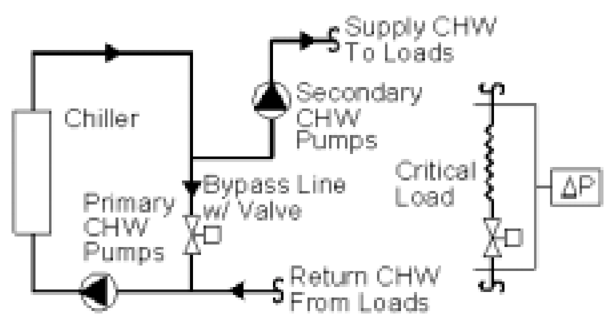


Figure 1-5: Variable primary flow/variable secondary flow pumping arrangement.

Addition of a check valve to the decoupler pipe of the constant flow primary/variable flow secondary system prevents flow from return to supply (Figure 1-6). This puts the primary and secondary pumps in series whenever the secondary flow demand exceeds the design flow of the active primary pumps. Assisted by the secondary pumps, the constant flow primary pumps move to the right on their characteristic curves to accommodate the additional flow and operate at a head lower than design while the pressure drop through the primary circuit, because of increased flow, is greater than design. If the variable speed secondary pumps are not capable of handling the increase in both head and flow, this is not an acceptable option as a retrofit.

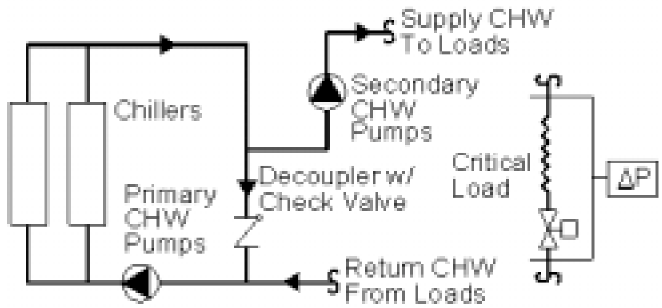


Figure 1-6: Consant flow primary/variable flow secondary pumping arrangement with a check valve in the decoupler.

1.1.4 Comparative overview of chilled water pumping systems

Although constant flow rate design is a low capital investment approach for chilled water systems, it wastes energy. This is because flow during periods of low load is much greater than would be necessary with variable volume two-way valve control. The most appropriate applications of constant flow systems have nearly constant loads or minimal distribution systems.

Variable flow systems have become the standard for larger systems due to their lower operating costs. The constant flow primary/variable flow secondary arrangement (primary/secondary) is presented in HVAC Textbooks (McQuiston et al. 2000) and industry Handbooks (ASHRAE 2000) as the model architecture for variable flow chilled water plants. In general, the design community accepts this view and considers these systems standard practice.

Drawbacks of constant flow primary/variable flow secondary systems are the additional cost and space required for separate plant and distribution pumps. This can be an issue particularly in retrofit projects where space is limited. Also, it can be difficult to economically justify a second set of pumps in a building that has a small cooling load.

Variable flow can be achieved with constant speed or variable speed pumps, although variable speed pumping usually results in significantly lower energy use. Because of technology advancements and decreasing cost of variable frequency drives, variable speed pumping has become the norm for variable volume chilled water systems. Energy saved by controlling the pump speed to match system head requirements can offset the cost of variable frequency drive equipment and provide significant energy savings.

Although the intent of variable distribution flow is to maintain a constant chilled water temperature differential, this rarely occurs. For a variety of reasons, most systems do not achieve design chilled water temperature differential at either design load or part load. This is referred to as "low ÷T syndrome" in the literature. Problems at the cooling load cause most low temperature differential problems. Examples include

- two-way valves that do not close against system pressure differential
- air-side temperature set points that are too low
- coil fouling.

among others. With proper design, operation, and maintenance most causes of low temperature differential can be prevented (Fiorino 1996, Taylor 2002), but some may be unavoidable. Taylor suggests that reduced coil effectiveness caused by water-side and air-side fouling, and lower-than-design entering air temperatures, common in systems with outside air economizers, are in the latter category.

The view that low temperature differential problems are endemic to chilled water systems has led some to conclude that chilled water systems must be designed to tolerate them (Kirsner 1996, Avery 2001). Proponents of this perspective contend that the conventional decoupled constant flow primary/variable flow secondary scheme with constant evaporator flow cannot respond effectively to low ÷T syndrome. The most serious effect of low ÷T syndrome on a constant flow primary/variable flow secondary system is inability to load chillers. Because evaporator flow rate is constant, full cooling capacity can be achieved only when the chilled water temperature difference across the evaporator is at its design value.

Recent publications suggest (Avery 2001, Hartman 2001) that the obvious successor to the constant flow primary/variable flow secondary system is the variable primary flow system. Constant evaporator flow plants respond to low ÷T syndrome by bringing more chillers and their auxiliaries on line to increase primary flow. Variable primary flow plants can avoid the need to start chillers by allowing evaporator chilled water flow rates to exceed the design value to compensate for low ÷T. Variable primary flow systems also match evaporator flow to system demand eliminating excess flow in the plant.

Advocates of variable primary flow consider the constant evaporator flow constraint of constant flow primary/variable flow secondary system design to be a fatal flaw that is unnecessary given the characteristics of contemporary chiller technology. This approach is gaining support among both chiller manufacturers and design professionals. Variable flow, primary-only systems are particularly attractive because they have lower equipment costs than primary/secondary systems.

1.2 Objective and Scope

Arguments supporting chilled water plant design based on variable primary flow have been advanced by influential figures in the industry. However, published literature provides little persuasive proof of performance benefits or detailed application guidance based on the performance of real applications. Such information is necessary to help designers and owners decide whether, and when, this new design approach should be adopted.

The objective of the research described in this report was to quantify the potential benefits of variable primary flow and to generate guidance for its use. The scope of the analysis was confined mainly to water-cooled electric motor-driven chilled water plants with parallel, equal sized chillers, but it can be generalized to other cases.

The work plan consisted of two major tasks :

1) a state-of-the-art review including extensive literature review and a survey of chiller manufacturers, chilled water system designers, and system owners to summarize current industry design criteria, application experience, and attitudes
2) a parametric study of simulated variable primary flow system performance, including energy use and economic comparisons with other pumping system types.

Chapter 2 summarizes the state-of-the-art review. Chapter 3 describes the parametric study. Conclusions and recommendations are presented in Chapter 4.

2. THE STATE OF THE ART

The "state of the art" in variable primary flow system design includes knowledge distributed among researchers, designers and manufacturers as well as open literature. Literature review; surveys of designers, owners, and chiller manufacturers; and correspondence were synthesized into a composite portrait of prevailing practices and attitudes. This information is divided into the following sections:

2.1 Information Sources
2.2 Variable Flow Operation of Chillers
2.3 Variable Primary Flow Design Practice
2.4 Comparisons of Variable Primary Flow with Other System Types
2.5 Survey of Experiences with Variable Primary Flow
2.6 Attitudes Toward Variable Primary Flow

Literature, survey responses and personal communications are referenced throughout this report as necessary to address each of these areas.

2.1 Information Sources

Published sources consulted for information on chilled water plant equipment and systems included archival journals, trade magazines, textbooks, handbooks, design guides and manufacturers' literature. Chilled water system designers, chilled water system owner/operators and chiller manufacturers were surveyed to obtain answers to questions found in published sources. These surveys were not intended to provide statistical information, but rather, to provide a cross-section of opinion and experience from the individuals who design, construct and operate chilled water systems. Surveys, responses, and correspondence with selected respondents are provided in Appendix A. Some questions were included in each survey while additional questions were specific to the group addressed. Questions to manufacturers addressed the suitability of a chiller for variable primary flow service and guidelines for application. Design professionals and system owners were asked about their attitudes toward variable primary flow, experiences with variable primary flow, application considerations and design practices.

Surveys were posted from May 2001 through April 2002 on a web site hosted by the Pennsylvania State University Department of Architectural Engineering. Respondents were obtained from solicitation of design firms in a database maintained by Penn State, a sidebar notice to an article on variable primary flow in a widely-read trade magazine (Bahnfleth and Peyer 2001), personal contacts by the investigators, and "walk-in" hits on the survey web site. A total of 52 responses were obtained, approximately 20 percent of those solicited. Forty-three respondents were designers, eight were system owners, and one was a chiller manufacturer. Four chiller manufacturers and several survey respondents participated in follow-up discussions.

Survey participants were assured anonymity, so names of respondents are not cited unless permission was granted. Responses are cited using a letter code to identify the category of respondent ("D" for designer, "O" for owner and "M" for manufacturer) followed by a number corresponding to the respondent in that category. If a response to a specific question in the survey is cited, the question number is added to the survey identifier.

2.2 Variable Flow Operation of Chillers

The ability to vary water flow rate in the evaporator of a chiller has always existed. However, until the mid-1990's, chiller manufacturers did not publicly support or encourage variable flow operation because of risks related to the limitations of on-board chiller controls. Improved chiller control technology has lessened these concerns to the point that most manufacturers consider variable primary flow to be acceptable when properly applied. Because increased interest in and application of variable primary flow are tied directly to the capabilities of modern chillers, it is appropriate to begin this summary by examining chiller characteristics under variable flow conditions and guidelines on variable flow application provided by manufacturers.

2.2.1 Chiller Controls

Variable flow presents a challenge to chiller stability. Internal chiller controls must maintain refrigerant and water temperature set points over the desired range of chilled water flow and respond acceptably to rates of change. Consequences of inadequate control include nuisance faults leading to frequent shutdown or even damage to the chiller. Chiller capacity modulation, freeze protection, and flow detection are key control issues in variable primary flow applications discussed in this section.

Capacity Control

The basis of chiller capacity control is typically regulation of leaving-chilled-water-temperature. A deviation from set point stimulates a response from the capacity controller, which modulates inlet guide vanes on a constant-speed centrifugal compressor, the steam valve on an absorption chiller or other capacity control devices (ASHRAE 1998).

The cooling load on the evaporator of a chiller is a function of both temperature differential and flow rate (Equation 1-1). When the evaporator water flow rate is constant and entering water temperature is not varying wildly, there is a linear relationship between leaving-chilled-water-temperature and cooling load. This makes the gain (rate of load change a function of leaving-chilled-water-temperature) of the system essentially constant, which promotes stable capacity control. In a variable primary flow system, the rate of load change as a function of leaving-chilled-water-temperature change varies with the chilled water mass flow rate. A given change in leaving water temperature at a low flow rate represents a smaller change in load than the same deviation at a higher flow rate.

With simple linear proportional controls found on older chillers, this could lessen chiller stability or limit system performance. Under low flow conditions, proportional control would overcompensate because the perceived load change would be greater than the actual load change, which could drive the chiller into instability. Under high flow conditions, the change in load would be underestimated and deviation from the leaving temperature set point would persist or grow (Eppelheimer 1996).

Present-day chiller controls are resistant to the problems described above. Manufacturers have abandoned slower, less responsive pneumatic control systems for microprocessor-based controls (Feduik 2002). Proportional-integral-derivative (PID) controllers have replaced proportional and floating action controllers. Integral control can reduce or eliminate proportional offset, while derivate control provides a faster response to transients that can reduce maximum offset (M2-9).

Control improvements have extended to absorption chillers, for example, by modulating solution flow rate as well as heat input (Schwedler 2002). The solution concentration and temperature are monitored, while heat input and solution flow are varied to maintain the leaving-chilled-water-temperature. The results are increased chiller stability and fewer constraints on flow and load variation (M3-5).

Freeze Protection

All chillers are equipped with safeties to prevent low refrigerant temperature from damaging the chiller by freezing water in the evaporator (ASHRAE 1998). Chiller control panels monitor either refrigerant evaporating pressure or leaving-chilled-water-temperature.

The risk of freezing the evaporator is greatest during a sudden drop in flow rate. Such changes can occur in the course of normal operation, as illustrated by the following variable primary flow example (Kirsner 1996). Consider a plant comprised of two identical parallel chillers with one chiller on line and fully loaded while the other is on standby with no flow through its evaporator. System load increases slightly causing leaving-chilled-water-temperature to rise and the plant controls bring the second chiller on line. If the flow demand of the system only slightly exceeds the capacity of one chiller, the new equilibrium flow rate in each chiller will be roughly 50 percent. If the decrease in flow through the first chiller occurs rapidly due to fast opening of the isolation valve on the second chiller, the first chiller will be placed in a condition of full refrigerant load at half chilled water flow. As Equation 1-1 dictates, this results in a doubling of the chilled water temperature difference. The chilled water temperature could plunge below the low limit before chiller controls are able to adapt to the new operating conditions forcing the chiller into a protective shutdown.

The best way to prevent this type of shut-down is to design system controls that prevent rapid excursions (see Section 2.3). However, the same problem can result from accidents such as a valve failure, so even good design does not preclude occurrence of this scenario. To protect the chiller from this failure mode, manufacturers use integral control to keep the chiller on line during a short lapse below freezing. Instead of shutting down the chiller immediately when a freezing temperature is sensed, the controller sums the degree-seconds below freezing and initiates shut down only when the total rises to a critical level (Eppelheimer 1996). This gives the capacity controller an opportunity to stabilize.

Flow Detection

The flow detection safety is another form of freeze protection that ensures that chilled water flow is present while the chiller is operating. They come into play in the control sequence during start-up or if there is a real failure that causes loss of flow. Flow detection devices are meant mainly to distinguish between no flow and design flow. Sensing a small fraction of design flow or of remaining stable in a rapidly fluctuating flow is not normal duty. In variable flow applications, flow detection methods better suited to the application should be used (Eppelheimer 1996, Hubbard 2002, D34-13). One solution is to monitor signals from the flow detection device at frequent intervals and base control action on multiple readings. One manufacturer's equipment samples flow indication at one-second intervals and requires five consecutive signals before the chiller control panel acknowledges the proof-of-flow circuit and action is taken (Eppelheimer 1996). Another solution is to use a differential pressure transmitter selected for variable flow duty (Hubbard 2002). Either of these measures will improve the reliability of chillers in variable primary flow applications.

2.2.2 Evaporator Tube Velocity and Flow Rate Recommendations

Variable primary flow system design is constrained by the range of flow rate permitted in the evaporator, how rapidly flow in the evaporator can vary without causing instability, and the system chilled water mass available to damp transients in load. Manufacturers' recommendations in each of these areas are examined in this section.

Velocity Limits

The range of flow rate for a given evaporator is a function of the high and low velocity limits of the evaporator tubes and the design tube velocity. Low velocity limits are established to prevent laminar flow from occurring; high velocity limits are set at levels that will prevent tube erosion. Manufacturers' catalogs provide minimum and maximum flow rates that correspond to these velocity limits. The velocities associated with catalog flow rate limits are essentially the same for all flooded evaporators and in general agreement with values found in the literature (Table 2-1). Developments in tube technology are tending to drive the minimum velocity lower. Values as low as 1.5 ft/s for special tubes can be found in current catalogs (Trane 2001).

The ASHRAE Handbook gives a velocity range for flow perpendicular to evaporator tubes of 2 to 10 ft/s for direct-expansion evaporators (ASHRAE 2000). The lower velocity limit is necessary to keep tubes clean and the upper limit to avoid erosion. Velocity ranges for direct-expansion evaporators vary greatly depending on construction and size and cannot be generalized.

Table 2-1: Recommended water velocity limits for flooded evaporator chillers.

Source	*Minimum velocity, ft/s*	*Maximum velocity, ft/s*
Eppelheimer 1996	3	11
Schwedler & Bradley 2000	3	11
Hubbard 2001	3	12

System Turnover Time

System turnover time is the time required to circulate one system volume at the system flow rate. It is a measure of the system water mass relative to the cooling load and indicates how rapidly temperature disturbances will propagate through the system. Longer turnover times increase stability of chiller control. Turnover time will be relatively small for small volume, close-coupled systems and large for systems such as district cooling systems. The volume contained in the distribution piping of larger systems is typically more than sufficient to prevent turnover from being a problem (Feduik 2002).

System turnover time limits are recommended by chiller manufacturers to ensure that capacity controls can react stably to variations in load. Minimum turnover times vary greatly depending on the chiller controls employed. A recent manufacturer's newsletter discussing air-cooled chiller application recommended that the minimum turnover time should be 7 minutes at design flow rate (McQuay 2001). Another chiller manufacturer suggests that system volume be at least 6 gallons/ton of installed chiller capacity (M2-11). This equates to a 3-minute turnover at design flow with a 12˚F chilled water temperature difference (i.e., 2 gpm/ton). Turnover time has additional significance for variable primary flow because some manufacturers tie it to their recommended rate of chilled water flow variation through the evaporator.

Rate of Chilled Water Flow Variation

The rate of chilled water flow variation (typically expressed as percentage of design flow rate per minute) should not exceed the rate at which the chiller can stably maintain the leaving-chilled-water-temperature. More rapid flow variations can cause control instability and compressor flood-back or shutdown (Redden 1996).

Recommended flow rate variation ranges from less than 2 percent per minute to 30 percent per minute depending on the chiller type, controls, and system turnover time. Vapor compression chillers equipped with continuous capacity control (e.g., water-cooled centrifugal chiller with inlet guide vanes, air-cooled screw chiller with continuous slide valve, etc.) are generally capable of larger rates of flow variation than equivalent absorption chillers. Table 2-2 provides manufacturer recommendations for rate of chilled water flow variation for vapor compression and absorption chillers. The range of values shown suggests that either rate of flow variation is very sensitive to chiller type or that some recommendations are considerably more conservative than others.

Table 2-2: Recommended maximum rate of flow variation for vapor compression and absorption chillers.

Source	*Vapor Compression Maximum rate of ÷#flow, % design flow/min.*	*Absorption Maximum rate of ÷#flow, % design flow/min.*
Manufacturer 1	4 - 12	Not provided
Manufacturer 2	20 - 30	2 - 5
Manufacturer 3	Not provided	30
Manufacturer 4	2	Not provided
Manufacturer 5	Not provided	1.67
Dietrich 1999	1.7 - 3.3*	Not provided
Schwedler 2001	10 - 30	Not provided
McQuay 2001	5 - 10	Not provided

* Maximum value based on a system turnover rate of 15 minutes or greater.

2.2.3 Energy Use Characteristics

If variation in evaporator flow has a significant effect on chiller energy consumption, it would affect the economics and applicability of variable primary flow. However, published data (Redden 1996) and data provided by a manufacturer for use in this study (Berry 2000) indicate that the impact of flow variation on chiller energy use is small.

Berry (2000) provided performance data generated by selection software for constant and variable flow operation under varying loads of a chiller with the design conditions shown in Table 2-3. Energy consumption (kW/ton) as a function of varying load and chilled water flow rate was calculated and is plotted in Figures 2-1 and 2-2. It is evident that the variable flow (i.e., constant temperature differential) chiller had nearly the same kW/ton for all flows examined over a wide range of part load. In all cases the kW/ton differed by no more than 2 percent relative to the value at design flow.

Table 2-3: Design criteria for study centrifugal chiller.

Cooling load, tons	*CHW flow rate, gpm*	*Leaving CHW temperature, ˚F*	*CW flow rate, gpm*	*Entering CW temperature, ˚F*
500	1200	44	1500	85

These results substantially agree with the previously test stand data of Redden (1996). Redden measured motor power consumption during test stand experiments in which the load on a centrifugal chiller was varied with constant flow and variable temperature difference or constant temperature difference and variable flow. Flow in the constant flow case was the design flow rate. In variable flow tests, flow rate varied in proportion to load from the design velocity of 5.3 ft/s down to a minimum tube velocity of 2.4 ft/s. The difference between constant flow and variable flow unit power consumption (kW/ton) at any load varied less than 1 percent.

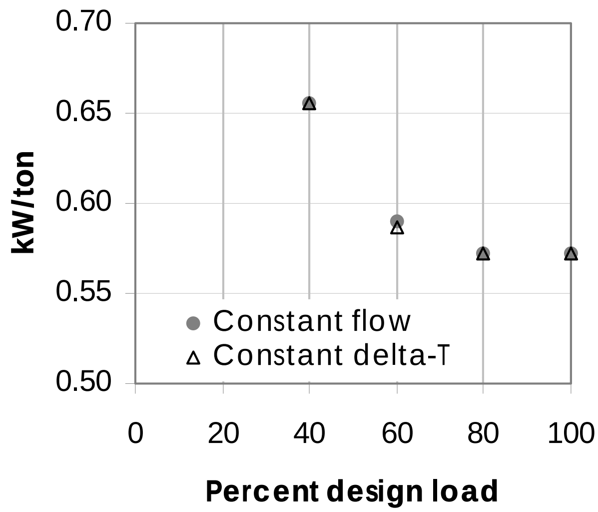


Figure 2-1: Chiller part-load performance (kW/ton) for constant chilled water flow with variable temperature difference and variable chilled water flow with constant temperature difference (Berry 2000).

Reduction in chilled water flow rate might be expected to permit COP improvements because a closer approach could be obtained between entering refrigerant and leaving-chilled-water-temperatures. However, low evaporator tube approaches (1 to 2˚F), typical of current chiller technology do not allow for significant energy savings potential through increased saturated evaporating temperature (Hubbard 2002).

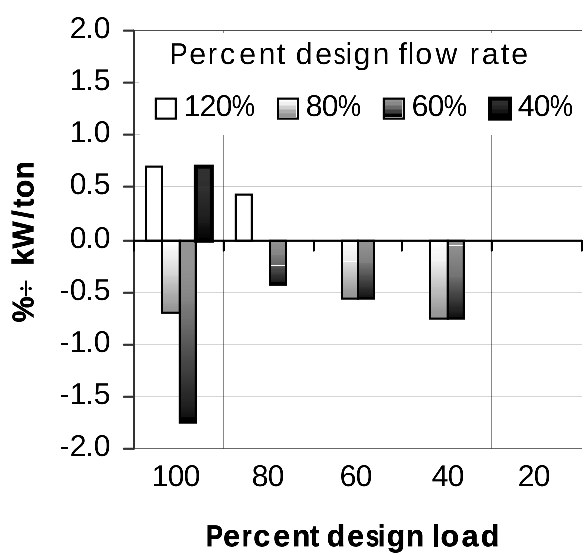


Figure 2-2: Change in chiller performance relative to design flow rate for various flow rates and part loads. (Berry 2000)

2.3 Variable Primary Flow Design Practice

Twenty of the 43 designers surveyed had prior experience with variable primary flow. Information obtained from their survey responses, published literature, and follow-up correspondence has been merged in this review of variable primary flow design practice. A summary of application considerations is followed by design issues; pumping arrangements, chiller selection, chiller staging, and low flow control.

2.3.1 Application Considerations

One of the advantages claimed for primary-only variable primary flow systems is that they not only consume less energy, but are also lower in first cost than conventional primary/secondary systems. On this basis, it might seem that they would always be the best choice. However, a variable primary flow application is more likely to be successful and economically beneficial under the following conditions (Schwedler and Bradley 2000, 2003, Taylor 2002):

- Cooling load varies and most loads are controlled by two-way valves. This is an obvious requirement since operating cost savings for variable primary flow result from flow variation. Schwedler and Bradley note that having some three-way valve controlled loads on a system does not preclude use of variable primary flow. A fifty percent reduction in system flow has the potential to reduce pumping power by as much as 80%.
- Slight variations in supply water temperature are acceptable. When both flow and load on a chiller vary, control response will be less stable than when flow is constant with the result that leaving temperature will vary more.
- Low flow measurement instrumentation can be maintained and calibrated regularly. Protecting chillers against low flow conditions is a critical monitoring and control system task that cannot be performed reliably if the primary flow rate is not accurately known.
- Designers and operators understand the need to operate chillers within recommended limits. The greater demands placed on chillers in a variable primary flow system reduce the margin for error and create more possibilities for system faults. These can be minimized by not staying within operating boundaries established by the equipment manufacturer and design engineer.

2.3.2 Pumping Arrangements

Variable primary flow has been applied to all of the common chilled water pumping system types, including primary-only, primary/secondary, and distributed pumping systems. Both the literature (Schwedler and Bradley 2000, Taylor 2002) and the designer surveys identified primary-only pumping as the most common system architecture (Figure 1-4). Nineteen of the 20 designer survey respondents with variable primary flow design experience have used the primary-only pumping arrangement at least once and 15 have used it exclusively (Table 2-4). Respondents cited lower first cost and less required space than comparable primary/secondary and distributed systems and owner request as reasons for using the primary-only configuration (Table 2-5).

Table 2-4: Variable primary flow pumping arrangements used by surveyed designers.

Pumping arrangement	*# of respondents*
Primary-only	19
Primary/secondary	3
Distributed	4

Table 2-5: Reasons that surveyed design professionals chose to use the variable primary flow primary-only pumping arrangement.

Reasons for primary-only arrangement	*# of respondents*
Lower first cost	7
Less space	4
Owner request	2
Had not considered other alternatives	2

Existing constant primary flow decoupled chilled water systems (primary/secondary or distributed pumping) have been converted successfully to variable primary flow (D34-11). One conversion method is to install variable frequency drives (VFDs) on the primary pumps and control them to minimize chilled water flow through the decoupler pipe. Another approach, illustrated in Figure 1-6, is to install a check valve in the decoupler pipe of the primary/secondary system (Kirsner 1996; Avery 1998, 2001; D34-11). When system demand for flow exceeds the design flow rate of the constant speed primary pumps that are operating, the check valve closes, the primary and secondary pumps are placed in series, and primary flow will increase to match secondary flow. There has been considerable discussion in the literature regarding whether the use of a check valve in the bypass is good design or simply a patch for a system that is controlling its loads poorly (Coad 1998, Hegberg 2001a, Luther 1998a, Rishel 1998, Kirsner 1998, Avery 1998, 2001, Taylor et al. 2000).

2.3.3 Chiller Selection

This section reviews chiller selection issues for variable flow applications. These include a number of criteria applicable to any chiller and considerations related to chiller type.

Selection Criteria

Chiller selection criteria important for variable primary flow operation are:

- Chiller type/control characteristics. The importance of chiller controls has already been discussed extensively. Variation of control capability across different kinds of chiller must be taken into account in the design process.
- Tube velocity limits. These manufacturer-imposed limits determine the maximum and minimum flow rates for a given evaporator.
- Nominal tube velocity. The design tube velocity, together with the maximum and minimum velocities recommended by the manufacturer, determines the range of flow that is permitted as a percentage of design flow rate.
- Evaporator water-side pressure drop. An objective of variable primary flow operation is to reduce chilled water pumping energy use, so the design pressure drop of the evaporator is an important system parameter.

Designer survey respondents were asked to identify the selection issues they considered most important. Their responses (Table 2-6) clearly singled out flow rate limits as their major concern.

Table 2-6: Most important variable primary flow chiller selection criteria identified by survey of designers.

Chiller selection criteria	*# of respondents*
Velocity or flow limits	14
Evaporator water-side pressure drop	6
Internal chiller controls capable of VPF	4
Rate of change for flow rates	2
No particular criteria specific to VPF	1

Chiller Type Considerations

Seventeen of the 20 respondents with variable primary flow design experience used water-cooled centrifugal or screw chillers in their systems (Table 2-7). Most said that they used these chillers not because of their ability to perform well in variable primary flow applications, but because of their efficiency (Table 2-8). Several designers noted that centrifugal chillers have superior control systems that enable them to operate with greater stability when evaporator flow varies.

Table 2-7: Chiller types used by surveyed designers for variable primary flow applications.

Chiller types	*# of respondents*
Water-cooled centrifugal and screw chillers	17
Air-cooled (reciprocating and screw) chillers	3
Absorption chillers	2

Respondents were apprehensive about variable flow application of absorption chillers because of the lack of absorption chiller guidance available. One respondent stated that a manufacturer he consulted recommended nearly constant flow (D10-12).

Three respondents (D6-12, D27-12, D34-12) successfully used air-cooled rotary screw chillers in variable primary flow applications. Others, however, warned of potential problems with air-cooled chillers because they are typically found in small systems that lack the system thermal mass to allow sufficient control response time for variable flow through the chiller bundle (D20-12, D27-12). Clearly, the concern in this case was not the chiller, but the turnover time of the system as discussed previously.

Table 2-8: Reasons why surveyed designers preferred electric motor-driven, water-cooled centrifugal or screw chillers for variable primary flow application.

Reasons	*# of respondents*
Energy efficiency	6
Energy costs	3
Availability of energy source	2
Lowest life-cycle cost	2
Superior internal chiller controls	1
Ability to vary flow	1
Lack of VPF application guidance for alternatives	1

2.3.4 Chiller Staging

Maintaining the leaving-chilled-water-temperature set point is the primary objective of chiller staging, with the secondary objective being to consume as little energy as possible (Avery 2001, Kirsner 1996). In a typical chilled water plant, most system auxiliaries operate at constant speed, including cooling tower fans, condenser water pumps, and primary chilled water pumps. In many systems, cooling tower cells, condenser water pumps and primary chilled water pumps are matched one-to-one with chillers. Aside from cooling tower fans, which may cycle as load or ambient conditions vary, these auxiliary components operate at full power whenever a chiller is on line. At small part loads, the parasitic power they consume can become a very large fraction of the total energy consumption of the system. Consequently, most chiller staging strategies take into consideration the total of chiller and auxiliary power.

Chiller staging strategies require comparison of chiller capacity with cooling load. This can be done in a variety of ways.

- Chilled water temperature leaving the evaporator indicates a capacity shortfall when it remains above set point for a specified length of time, but is not a good indicator of excess capacity
- In a primary/secondary system, bypass flow direction (indicated by temperature in the bypass) can indicate a capacity shortfall
- Calculation of load from measurements of flow and temperature can be useful both for deciding when to start or stop chillers
- Measurement of current drawn by the compressor motor is also useful for controlling chiller starts and stops
- Models of chiller and chilled water plant operation can, in principle, be used off-line or in real time to support optimal control strategies. This is not currently the norm, but may be the future direction of chilled water plant controls.

These methods can be applied with varying degrees of effectiveness to many different system types.

This section reviews recommendations for staging of constant speed chillers and variable speed chillers in primary/secondary and variable primary flow applications with additional discussion of methods for avoiding nuisance trips that may result during staging.

Constant Speed Chiller Staging

For a given condensing condition, the efficiency of typical constant speed vapor compression chillers typically varies little from full load down to a part load of perhaps 30 percent, below which efficiency decreases rapidly. Per unit of cooling produced, the total of chiller and auxiliary power is typically lowest at full load. Consequently, constant speed chillers are typically staged to minimize the number operating to meet a given load.

In primary/secondary systems with constant primary flow, staging must keep the flow through the decoupler from supply to return at a rate less than the flow of the smallest operating chiller. This can be done relatively simply using temperature measurements at the bypass. The situation is more complex for variable primary flow chillers because they may be limited either by high or low evaporator flow rate limits or by maximum capacity, which varies with chilled water and condensing conditions. Variable primary flow chillers can be staged based on measurement of leaving-chilled-water-temperature, cooling load, chiller current draw, chilled water flow rate, or a combination of two or more of these methods. Staging methods used in variable primary flow systems by survey respondents are summarized in Table 2-9.

Table 2-9: Methods for staging chiller capacity used by surveyed designers.

Chiller staging	*# of respondents*
Calculated load	7
Chilled water flow rate	3
Combination of leaving CHW temperature and chiller current or calculated load	2
Chiller current	1

Available chiller capacity can be calculated using a simple model of chiller performance if cooling load and the temperatures of leaving chilled water and entering condenser water are measured. Details of empirical chiller models are discussed in Section 3 of this report. Nominal chiller capacity can be used for load-based staging, but this is not optimal and may create a situation in which insufficient chiller capacity is activated because the capacity of operating equipment is overestimated. This could occur whenever actual condensing conditions are more severe than design condensing conditions.

Control panels of electric motor-driven chillers monitor the current drawn by the compressor. Compressor current can be used to stage chillers. If the leaving-chilled-water-temperature set point is satisfied and the chilled water flow rate is within the high and low limits, then current can be increased until it hits its high limit. If the maximum flow limit is reached, another chiller must be brought on line. If the peak amperage is attained and chilled water temperature exceeds the set point then another chiller must be added. Knowing when to subtract chiller capacity may be the most difficult decision with this strategy. One way of accomplishing this is by comparing the peak and actual operating current. If the difference is greater than the peak amps of the smallest chiller in operation and the flow rate will not exceed the maximum of the remaining chillers, then the chiller can be taken off line.

Variable Speed Chiller Staging

Variable speed chillers have higher off-design efficiencies than constant speed chillers. Savings occur because the compressor is essentially a refrigerant fan that can realize the same energy use benefits under part load operation as variable speed pumps and fans in variable flow systems. Variable speed chiller energy savings occur whenever the lift on the compressor is lower than design, even at design load. Energy savings can be very substantial over an annual cycle. The use of variable speed chillers to date has been rather limited because of their cost, but can be expected to increase as energy costs increase and the cost of drives decreases.

Variable speed chillers typically are staged like constant speed chillers, i.e., to minimize the number of operating chillers and their associated auxiliaries. Plants with multiple chillers typically include one variable speed chiller and one or more constant speed chillers to reduce first cost. In such plants, the variable speed chiller modulates to handle variations in load while constant speed chillers are fully loaded.

Because the constant speed chillers and auxiliaries are subject to part load efficiency loss as described previously, it is generally reasonable to minimize the number of constant speed machines operating. However, when multiple variable speed chillers are available, this adoption of the constant-speed paradigm may not maximize energy savings.

Compressor motors are the largest power consumers in a chilled water plant. When they can operate at variable speed at reduced load, large motor power savings are possible relative to constant speed operation. Affinity laws applicable to fans and compressors indicate that it is generally more efficient to meet a given load with multiple compressors operating at reduced speed than with a single compressor operating at full speed. The performance of a variable speed machine will deteriorate at low part loads as drive and motor efficiency fall rapidly, but over a wide range of load the principle of operating as many variable speed chillers as possible may give the lowest energy consumption. One source of design guidance recommends using as many variable speed chillers as possible provided that they are loaded at least 20 – 35% (Taylor et al. 2000).

Hartman (2001) takes this concept to its limit in recommending that all pumps, fans, and compressors in a chilled water system should have variable speed drives and be optimally dispatched. He argues that variable condenser and evaporator flow must be permitted in order to achieve optimal chilled water plant energy savings. Otherwise, condenser and evaporator water pumping energy and cooling tower fan energy will detract from the benefits of operating many variable speed chillers simultaneously.

Avoiding Nuisance Trips During Chiller Staging

Any chilled water plant may experience faults during chiller staging that cause running chillers to shut down while an attempt is being made to bring another chiller on line. The cause of such problems is often the response of chiller controls to sudden changes in water flow or other system conditions. Primary/secondary systems, which allow for free bypass of excess primary flow, may experience smaller transients in the flow through operating chillers when a new chiller and its chilled water pump are started and are relatively fault resistant (Eppelheimer 1996). A different situation exists for variable flow, primary-only systems, as previously described in detail section 2.2.1. In the worst case, the flow through one chiller may suddenly be cut in half, virtually guaranteeing a freeze protection shutdown.

The control stability issue is the same for constant primary flow and variable primary flow systems, but the risk is substantially greater for variable primary flow. Likewise, the remedies/preventive measures are the same, but it is even more important that they work correctly in a variable primary flow application. Recommended practices include using slow opening isolation valves and specifying control sequences that unload active chillers prior to initiating flow through a standby chiller. A typical sequence of operation (Taylor 2002) is:

- Unload all active chillers.
- Slowly open the isolation valve for the chiller to be activated.
- Activate the chiller after flow is confirmed.
- Load all active chillers together.

2.3.5 Low flow control

Methods for Low Flow Control

Every chiller has a minimum permissible evaporator flow rate fixed by the manufacturer's low water velocity limit. A variable primary flow system must have either:

- a low flow bypass, so that it is possible to operate with a single chiller when the system flow rate is less than the chiller minimum (see Figure 1-4), or
- a sufficient number of three way valves or wild coils to guarantee that the system flow will never fall below this limit.

Eighteen of the 20 survey respondents with variable primary flow design experience use a bypass with a modulating valve in order to maintain minimum flow, while the other two respondents use three-way valves. System flow is controlled by a differential pressure sensor at or near the most hydronically remote coil that regulates pump speed to meet a minimum set point. Independently, flow measuring devices on each evaporator open the normally closed bypass valve whenever the flow through an active evaporator falls below its minimum. Flow measuring methods include both the use of flow meters and the measurement of pressure differential across the evaporator, from which flow rate is inferred. Survey respondents showed a slight preference for the pressure differential method, probably because it adds no cost to the system (Table 2-10). The drawback of this method is its lower accuracy.

Table 2-10: Evaporator flow measurement device preferences of surveyed designers.

Flow measurement	*# of respondents*
Monitor flow with ÷P across evaporator	8
Flow meter type depending on application	5
Either flow meter or ÷P depending on application	4

Three-way valves have both advantages and disadvantages as an alternative to a controlled bypass. The advantages are that the system is turned over regularly and minimum flow control is simplified. The disadvantages are that pump energy is wasted continuously and bypassed water lowers the system temperature differential (Schwedler and Bradley 2000). The significance of these disadvantages is in direct proportion to the fraction of system flow affected and essentially in proportion to the number of chillers in the plant. For example, if a given chiller has a minimum flow rate that is 40 percent of its design flow rate, a plant that uses ten such chillers to meet the peak load requires a 4 percent bypass through three-way valves to be protected against low flow. It will experience minimal adverse consequences compared to a single-chiller plant, for which a 40 percent minimum flow rate is quite large.

Low Flow Bypass Control

The following guidelines have been suggested for low-flow bypass design and control (Taylor 2002):

- Locate the valve close to the plant. If bypassed water has a shorter distance to travel it will have less impact on chilled water pumping energy.
- Modulate the valve to maintain minimum flow rate of the largest chiller on line.
- Size the valve for the minimum flow rate of the largest chiller. The maximum quantity of bypass flow required at any given time is the minimum flow rate of the largest chiller.
- Size the valve to operate properly with a pressure drop less than the set point of the system differential pressure sensor under all operating conditions.
- Flow measured with flow meter or pressure differential across the chiller evaporator.

It should be noted that selecting a single valve to meet all of the criteria above may be difficult in some applications. If controllability over a wide range of differential pressure is of concern, alternatives such as the use of parallel valves may be considered. This, of course, adds to both the cost and complexity of the system.

2.4 Comparisons of Variable Primary Flow with other Pumping Systems

Table 2-11 summarizes primary-only variable primary flow advantages and disadvantages relative to primary/secondary systems as noted in a variety of sources (Taylor 2002, Luther 1998b, Kirsner 1996, Hartman 1996). This section considers some of these issues in greater detail, including capital cost, space requirements, and energy use.

Table 2-11: Advantages and disadvantages of a variable flow, primary-only system relative to a primary/secondary system.

Advantages	*Disadvantages*
Lower first cost	Increased chiller staging complexity
Reduced peak demand	Increased bypass control complexity
Reduced energy use	Requires greater operator competency
Less mechanical room space	Requires greater designer expertise
Ability to cope with lower-than-design CHW ÷T	

2.4.1 Capital cost

The variable flow, primary-only pumping arrangement has been promoted as a lower cost alternative to the constant flow primary/variable flow secondary system. Table 2-12 summarizes the distinctions between these systems that lead to cost differences.

Table 2-12: Sources of cost difference between variable flow, primary-only and primary/secondary systems (Schwedler and Bradley 2000, Taylor 2002).

CHW system type	Primary/secondary	Variable flow, primary-only
CHW pumps	Two sets of lower head pumps	One set of higher head pumps
Services for constant flow chiller pumps	Pipe/fittings, electrical service, control wiring, and starter	N/A
Variable frequency drives (VFD's) for CHW pumps	VFD's and electrical service for distribution pumps	VFD's and electrical service for larger pumps
Bypass/decoupler line	Sized for the design flow rate of the largest chiller	Sized for the minimum flow of the largest chiller
Bypass control valve	NA	Modulating control valve
Flow measurement	NA	Flow meter (or ÷P sensors)
Mechanical room space	Space required for two sets of low head pumps	Space required for one set of high head pumps

While savings in pump, piping, fitting and electrical costs favor the primary-only system, the need for larger variable frequency drives, modulating bypass control valve, and flow measurement device decrease the net first cost savings (Taylor 2002). Also, although primary-only systems have fewer chilled water pumps than comparable primary/secondary systems, the primary-only pumps must handle the pressure drop through both the distribution and plant circuits. These higher-head, larger-motor primary-only pumps may not be much less expensive than the total cost of equivalent primary/secondary pumps (Hegberg 2001b) although this effect is very application-specific.

An example of cost differences between representative equivalent variable flow, primary-only and primary/secondary systems was constructed to provide a point of reference for the generic statements found in the literature. Chilled water pumps, variable frequency drives, motor starters, piping, and fittings were selected for the system design conditions shown in Table 2-13. Two 10 hp chilled water pumps and two 15 hp secondary distribution pumps configured in parallel were selected for the primary/secondary system.

Two 30 hp pumps configured in parallel were selected for the primary-only system. An equipment vendor (Anzelone 2002) supplied pump prices, and mechanical contractors estimated the pump installation costs (Hayden 2002, Tressler 2002). Installed costs of piping and other accessories were estimated using standard construction cost data (Means 2001).

Component and total costs are compared in Table 2-14. Use of a variable flow, primary-only chilled water system resulted in a substantial pumping system savings of $15,960 relative to the roughly $44,000 cost of the primary/secondary system pumping system. In the larger picture of total plant costs, however, this savings is a relatively modest $3.25 per peak ton, and could be less than 3 percent of the cost of a complete plant. Most of the benefit of the primary-only system in this example comes from reduced pipe and fitting costs. While pump and pump installation costs were smaller for the variable primary flow system, the primary-only drive cost was significantly greater than the cost of drives for secondary pumps in the primary/secondary system. The larger decoupler of the primary/secondary plant offsets a small portion of the cost of the modulating control valve included in most variable primary flow plants. The choice of flow measurement strategy could impact the savings as flow meter costs range from $1,050 for a venturi-type flow meter (Means 2001) to $4,000 for an electromagnetic flow meter (Neville 2002).

Table 2-13: Design conditions for example first cost comparison between primary/secondary and variable flow, primary-only system.

CHW system type	Primary/ Secondary	Variable flow, primary-only
Cooling load, tons	500	500
Total CHW flow rate, gpm	1,000	1,000
Primary pump head, feet	50	120
Secondary pump head, feet	70	NA

Table 2-14: Example component and total pumping system cost comparison of primary/secondary and primary-only variable primary flow.

CHW system type	Primary/ Secondary	Variable flow, primary-only
CHW pump equipment cost, $	10,516	7,358
CHW pump installation cost, $	2,857	1,486
Piping and fittings installed cost, $	19,070	NA
VFD/starter installed cost, $	9,860	14,550
Bypass/decoupler installed cost, $	1,328	929
Bypass valve installed cost, $	NA	1,548
Flow meter installed cost, $	NA	1,800
Total installed cost, $	43,631	27,671
÷ *Total installed cost, $*	Base	-15,960
÷ *Total installed cost, %*	Base	-37

Variable flow, primary-only plants require less space than primary/secondary plants because they eliminate one set of pumps. An estimate using pump frame sizes available on a manufacturers web-site (ITT 2002) and clearances of 2 to 3ft for piping connections and pump maintenance resulted in a space savings of approximately 0.05 ft^2/gpm, or 50ft^2 in the 1000 gpm example (Table 2-13), for the variable flow, primary-only plant. This difference may not amount to a great deal of first cost savings for a low tonnage construction project. However, the amount of space savings would be greater for larger applications, as larger pumps would require greater clearances. A variable flow, primary-only plant may also be advantageous when there is not enough space available for a conventional decoupled pumping arrangement, as is the case in the retrofit of some constant volume systems (Luther 1998b).

2.4.2 Energy Use

A number of published sources claim and, in some cases, document energy savings achieved in theory or practice by variable primary flow systems due to reduced chilled water pump, chiller, and chiller auxiliary energy costs (Schwedler and Bradley 2000, Bahnfleth and Peyer 2001, Bellenger 2003, Peterson 2004). Comparison of these claims and data reveals points of agreement, but also raises questions about the significance of some operational differences.

Chilled Water Pump Peak Demand

It has been suggested (Taylor 2002, D36-8) that the peak electrical demand of a variable flow, primary-only system may be less than that of a primary/secondary system because flow in the primary/secondary system must pass through two sets of pump trim rather than one, which creates a greater total pressure loss. A representative header-to-header calculation based on data from a widely used piping handbook (Ingersoll-Dresser 1998) for a 500 gpm flow in a 6 inch pump circuit is summarized in Table 2-15. The estimated head loss is 6 feet. Designer 36 estimated a similar or greater head reduction of 6 to 15 feet for the primary-only system.

Table 2-15: Pressure drop through pump circuit with a design flow rate of 500 gpm.

Quantity	*Description*	*Pressure drop, feet*
20 LF	6-inch Std. weight schedule 40 steel pipe	0.33
4	6-inch 90˚ elbow (long radius)	1.12
2	6-inch Tee (branch flow)	1.05
2	6x4 Reducer (flow in direction of increasing area)	0.25
2	6x4 Reducer (flow in direction of decreasing area)	0.16
2	6-inch Butterfly valve (wide open)	0.69
1	6-inch Swing check valve	0.66
1	6-inch Y strainer	1.73
2	4-inch Flexible connection	0.05
-	Total pressure drop, feet	6.04

For a given flow rate, pumping power is approximately proportional to pump head. As a fraction of total pumping power, the impact of this additional pressure drop depends on the percentage of the total pump head it represents. Additional friction losses of 6 to 15 feet could cause a 12 to 30 percent increase in pumping power for a pump selected at a design head of 50 feet, or as little as 4 to 10 percent for a pumps selected at 150 feet.

However, this argument ignores the larger context of the chilled water plant, in which there are potentially head increases associated with variable primary flow. The most significant of these is the pressure drop through the evaporator, which is quite sensitive to the design velocity selected. If a system designer chooses a design velocity of 7.5 ft/s for tubes with a 3 ft/s minimum in order to permit reduction of flow to 40 percent of design while an alternative selection for constant flow with more tubes of the same dimensions has a design velocity of 6 ft/s, the higher velocity case will have a pressure drop roughly 56% greater than the lower velocity case. This could easily negate the 6 to 15 ft savings realized by deleting additional pump trim. The total head may be greater for some primary/secondary systems, but it does not seem generally true that this is a significant effect and, in some cases, a primary-only system might actually have a higher design head. A further issue brought out in the analysis of Bahnfleth and Peyer (2001, 2003) is that when chillers in a variable primary flow plant operate beyond their design point the pressure drop through the chiller can negate savings in auxiliary power. For two chillers with the same flow rate and head loss at design flow, operation at 120 percent of design will cause head loss to increase by 44 percent in variable primary flow.

Another proposed peak electrical demand savings for a primary-only variable primary flow system relative to a primary/secondary system is that pumps selected for lower head duty generally are less efficient (Taylor 2002, Rishel 2000). Table 2-16 gives typical selection data for 12 different pumps. All three pumps selected for 1,500 gpm at 170 feet of head are roughly 5 to 9 percentage points more efficient than the pumps selected for 50 feet of head. However, this is not true for all of the selections. Two of the pumps selected for 500 gpm flow with 50 feet of head have higher efficiencies than those with 120 feet of head. As in the case of the preceding argument, this advantage is application specific and cannot be taken as generally true.

Table 2-16: Typical chilled water pump selections (ITT 2002).

Nominal flow rate, gpm	*Nominal head, feet*	*Motor speed, rpm*	*Motor size, hp*	*Impeller size, inches*	*Pump efficiency, %*
500	50	1750	10	8.375	77.4
500	50	1750	10	7.75	81.0
500	50	1750	15	7.625	69.0
500	120	1750	25	12.25	72.7
500	120	1750	30	11.25	77.0
500	120	1750	40	11	69.8
1500	50	1765	30	10.8	71.2
1500	50	1765	30	10.8	71.2
1500	50	1750	30	9.125	72.8
1500	170	1780	100	14.8	78.1
1500	170	1780	100	14.8	78.1
1500	170	1780	125	14.6	80.5

Chilled Water Pump Energy Use

According to affinity laws, brake horsepower varies as the cube of the pump speed. Unlike constant primary flow systems, variable primary flow provides comprehensive variable frequency drive pumping for both distribution and plant chilled water circuits. Two published case studies (Schwedler and Bradley 2000, Bahnfleth and Peyer 2001) show that the energy savings potential of pumps in variable primary flow applications depends greatly on the amount of frictional loss in the primary circuit. Systems with a higher ratio of primary to secondary circuit pressure drop will provide greater savings potential for variable primary flow with respect to constant flow primary/variable flow secondary systems.

Using commercial energy and economic analysis software, Schwedler and Bradley (2000) compared three pumping alternatives: constant and variable flow primary-only systems and a constant flow primary/variable flow secondary system serving a medical office building in Atlanta, GA. The design conditions for this example are given in Table 2-17. Each alternative used two air-cooled screw chillers piped in parallel with a pump dedicated to each chiller. A summary of results is given in Table 2-18. Variable primary flow provided 50 and 20 percent pump energy savings, respectively over the constant flow, primary-only and primary/secondary systems.

Bahnfleth and Peyer (2001) compared the same three pumping alternatives for a similar building but made different pump head assumptions that are reflected in certain differences between their results and those of Schwedler and Bradley. A simulated chilled water plant was exercised using actual hourly load data from an office building in Ithaca, NY. Each alternative used two water-cooled centrifugal chillers piped in parallel with dedicated condenser water pumps and cooling towers. Chilled water pumps were piped in parallel and headered together. Design conditions for their example are also shown in Table 2-17 and results are summarized in Table 2-18. Variable primary flow provided 66 and 40 percent pump energy savings over the constant flow, primary-only and primary/secondary systems, respectively.

Table 2-17: Design conditions for published variable primary flow case studies.

Source	*Cooling load, tons*	*Flow rate, gpm*	*Primary circuit pressure drop, feet*	*Secondary circuit pressure drop, feet*	*Ratio of primary to total pump head, %*
Schwedler & Bradley 2000	470	1200	20	60	25
Bahnfleth & Peyer 2001	500	1000	50	70	42

Table 2-18: Pump energy savings for two-chiller plants

Source	*Relative to constant flow, primary-only, %÷*	*Relative to primary/secondary, %÷*
Schwedler & Bradley 2000	50	20
Bahnfleth & Peyer 2001	66	40

Differences in pump head assumptions in the two case studies (Table 2-17) largely explain the percentage differences in variable primary flow pump energy savings shown in Table 2-18. The factor of two difference in savings relative to a primary/secondary system (40 vs. 20) is matched by a nearly factor of two difference in the percentage of primary pump head (42 vs. 25).

Chiller and Auxiliary Energy Use

As discussed previously, chiller performance is affected little by variation in flow rate (section 2.2). However, constant flow chillers operating with below-design chilled water temperature differentials may not be loaded to full capacity, which can cause more chillers than necessary to be on line. When this happens, constant flow chillers operate with low efficiencies and consume greater energy. Additional energy is consumed by the increased operation of constant speed condenser water pumps and cooling tower fans.

Variable primary flow operation of chillers can permit evaporator flow rates to exceed design so that chillers can be fully loaded during times when chilled water ÷T drops below design. As a result, variable primary flow chillers tend to achieve higher efficiencies and constant speed auxiliary equipment operates fewer hours. However, if a constant primary flow system is able to maintain a near-design chilled water temperature difference, the benefit of above-design flow operation of chillers may be small, as was noted in the case study by Bahnfleth and Peyer (2001).

2.4.3 Controllability and Maintainability

Variable primary flow systems are acknowledged to be more difficult to control and maintain than comparable constant primary flow systems even by their proponents. In addition to the controls required in constant primary flow systems, variable primary flow requires an accurate means of measuring chilled water flow rate and a method for adding units of chiller capacity without abruptly reducing the flow rate through active chillers.

2.5 Survey of Experiences with Variable Primary Flow

The consensus of surveyed designers and system owners who have variable primary flow experience is that it is a feasible approach to chilled water pumping system design. Most reported no significant problems with their projects. All surveyed system owners would consider variable primary flow for future projects, and only one of the 20 designers (D32-10) would not design another variable primary flow system.

Designer 34 (D34-12) reported having successfully applied variable primary flow with chillers from four major manufacturers--including chillers with centrifugal, reciprocating, and helical screw compressors and with both flooded and direct-expansion evaporators. He also stated that he had successfully applied variable primary flow as a retrofit to chillers built in the 1970s that do not have modern digital controls.

Designer 34 (D34-8) also reported resolution of low ÷T problems in a constant flow primary/variable flow secondary system that was converted to variable primary flow. Peak loading of chillers increased by 70 percent after the retrofit.

Other survey respondents indicated similar success (D18, 20, 27, 30) and several (D17-8,10, D37-8,10, O6-7, O8-6) claimed that their variable primary flow systems have saved energy. Problems noted were typically related to initial tuning of controls and were said to be minor (D10-10, D12-9).

In justification of his preference for constant flow primary/variable flow secondary systems, Designer 32 noted flow control problems and inadequate support from manufacturers (D32-7) as the main reasons he would not choose to design variable primary flow systems in the future. Designer 10, while not opposed to the use of variable primary flow on future projects, indicated that he would not apply variable primary flow in the future where there either was no building engineer or, in his judgment, the building engineer lacked the sophistication to operate such a system.

Commissioning and maintenance costs were higher and operating costs were lower for variable primary flow systems relative to equivalent variable primary flow systems. Survey respondents believed that commissioning was more difficult, and therefore more expensive, because variable primary flow controls are more complex and startup takes longer (O1-11, O5-11). Estimates of maintenance costs, based on system owner experience, ranged from no difference (O1-12) to 2 percent higher due to added controls and equipment (O5-12).

2.6 Attitudes toward Variable Primary Flow

2.6.1. Designers without Variable Primary Flow Experience

Although variable primary flow has strong proponents, many designers remain skeptical of its feasibility. Nearly half (20 of 43) of the design professionals who returned surveys had never designed a variable primary flow system. Responses from three who claimed to have done so suggest that they may not have understood the distinction between a variable primary flow system and a primary/secondary system (D3, D11, D16). Table 2-19 summarizes reasons why surveyed designers have not applied variable primary flow in their projects.

Table 2-19: Survey respondents' reasons for not having applied variable primary flow.

Description	*# of respondents*
Lack of guidance/support from manufacturers and literature	13
Recent technology/unproven	7
Concerned about chiller performance	5
Have not found right client/application	4
Complexity	4
Unfamiliarity of those involved with the project.	3

Thirteen respondents identified lack of guidance and technical support from manufacturers and the literature as a reason for not employing variable primary flow on a project. Designers expect manufacturer representatives to provide flow and temperature limitations, acceptable rates of chilled water flow variation, control sequences, and other guidelines for chillers in variable primary flow applications and have not always been able to obtain it. Designer 42 stated that only one vendor is providing this information, while others (D2-7, D22-7) suggested that more should be done by manufacturers. Two survey respondents (D8-4, D16-4) believed that some chiller manufacturers "require" constant evaporator flow, while three others (D7-7, D14-4, 7 and D28-7) stated that manufacturers are not advocates of variable primary flow. Designer 14 noted that if HVAC sales and service representatives are not confident of variable primary flow that design professionals will be less likely to attempt to apply it.

These are interesting responses, given that several chiller manufacturers not only provide such design guidance upon request, but also have published articles and technical papers on the subject (Schwedler and Bradley 2000, Eppelheimer 1996, Redden 1996) and include variable primary flow application guidelines in newsletters and catalogs (McQuay 2001, Trane 2001). Numerous recent publications suggest that chiller manufacturers have become supportive of variable primary flow application of their equipment (Kirsner 1996, Luther 1998b, Rishel 1998, Avery 2001, Taylor 2002).

Five respondents (D4-7, D19-7, D23-7, D31-7, D35-7) found the literature lacking in guidance, confusing, and contradictory. Designers would like to see more case studies and detailed design information before attempting variable primary flow. Designer 31 (D31-7) would like more discussion of the feasibility of variable primary flow for various plant capacities and units of chiller capacity. Designer 23 (D23-7) would like to see more presentation of the reasons for using specific design strategies, including anecdotal information from users.

Several designers pointed out that their clients (building owners and owner representatives), contractors, service technicians, and testing agencies are not familiar with variable primary flow. Because variable primary flow is perceived to be more complex than constant primary/variable secondary, there is more confusion and less support from these parties during the design, construction, start-up, and operation of the system. As a result, many designers decline to attempt variable primary flow.

2.6.2 Designers with Variable Primary Flow Experience

Designers with variable primary flow chilled water system experience gave the reasons summarized in Table 2-20 for choosing to use it. Most use variable primary flow because they believe it will provide energy and operating cost savings, lower first cost, less required space, or a combination of these benefits.

Table 2-20: Survey respondents' reasons for designing variable primary flow systems.

Description	*# of respondents*
Energy savings/reduced operating costs	12
Lower first cost	11
Less space required	8
Minimize number of chillers on line and kW/ton	6
Simplicity	4
Owner preference	2

2.6.3 System Owners and Operators

Most system owners were willing to consider the use of variable primary flow in their systems. Five of the 8 survey respondents own or operate a variable primary flow system. Of the others, two confused variable primary flow with variable secondary flow in a primary/secondary system (O2, O7) and another (O4-5) would consider variable primary flow in future applications but has yet to operate a variable primary flow system.

Owners with variable primary flow experience (O1, O3, O5, O6, O8) believe their systems to be successful and would consider variable primary flow for future chilled water systems. Their attitudes range from enthusiastic (O1-8) to cautious (O2-8,O6-8) about finding the right application for before attempting variable primary flow in future projects . Owner 8 has already begun planning for another major variable primary flow project.

3. PARAMETRIC STUDY

An extensive parametric study was performed to investigate the effect of potentially significant variables on chilled water plant energy consumption, operating cost, and economics. Computer models were developed to simulate chilled water plant performance using hourly load data as input. This summary of the parametric study has five major sections:

3.1 Study Parameters
3.2 Chilled Water System Simulation
3.3 Simulation Matrix
3.4 Simulation Results
3.5 Economic Analysis

3.1 Study Parameters

Parameters with the potential to affect chilled water system performance include chilled water system type, chilled water plant equipment type, cooling load type, climate, chilled water temperature difference (÷T), and number of chillers.

3.1.1 Chilled Water System Type

Four chilled water system types were modeled:

- Constant flow, primary-only
- Constant flow primary/variable flow secondary
- Constant flow primary/variable flow secondary with bypass check valve
- Variable flow, primary-only

The constant flow, primary-only system, although wasteful of pumping energy, is widely used in systems driven by considerations of simplicity and low first cost. The constant flow primary/variable flow secondary system represents conventional variable flow chilled water system design. The constant flow primary/variable flow secondary system with a bypass check valve was included in the study because it has been discussed in the literature as a retrofit solution to low ÷T problems in constant flow primary/variable flow secondary systems (Avery 2001). As noted in section 2.3.2, primary-only is the most widely used variable primary flow system architecture.

3.1.2 Equipment Type

Only central plant equipment, including chillers, pumps, and cooling towers was modeled in the parametric study. Components not modeled were assumed to perform identically in each scenario. Component models were based on commonly used equipment types. This section presents an overview of equipment alternatives. Component models are described in detail in section 3.3.2.

Prior studies (Redden 1996) indicated that chiller energy consumption characteristics do not change significantly as flow in the evaporator varies as long as leaving chilled water temperature remains constant. Therefore, constant-speed, electric motor-driven, water-cooled centrifugal chillers were used in all simulations because of their common use in chilled water plants ranging from medium-sized commercial buildings to large campus chilled water plants.

Electric motor driven, single-stage, centrifugal pumps are the most common pumps found in hydronic systems (ASHRAE 2000) and were used in both chilled water and condenser water system models. End-suction or double-suction, flexible-coupled pumps were selected depending on the design flow rate and head.

Induced draft, cross-flow cooling towers were used because they have lower capital and operating costs than comparable forced draft towers. Each tower had a single cell equipped with a two-speed fan motor (full and half-speed operation).

3.1.3 Cooling Load Type

Three cooling load types were modeled:

- 500-ton office building
- 1500-ton medical facility
- 4500-ton district chilled water plant serving a campus of five buildings

Load types were selected to represent a range of load distribution and size. A range of load sizes was necessary in order to investigate a representative group of plant configurations. A variety of load types was considered because of the potential affect of the load profile on the ability of variable primary flow operation to generate energy savings.

3.1.4 Climate

Three climates were considered:

- A relatively long, hot, and humid cooling season (Houston, TX)
- A relatively long, hot, and dry cooling season (Phoenix, AZ)
- A relatively short and humid cooling season (Syracuse, NY)

The cities chosen to represent the generic climate types were selected because they are TMY (Marion and Urban 1995) weather data sites. In addition, Syracuse was chosen because of its proximity to Ithaca, NY. Measured load data from Ithaca were available and could be used to validate modeled cooling load data and simulation procedures, as described in section 3.2.

Figures 3-1 and 3-2 are dry-bulb and wet bulb temperature duration curves, respectively for the three climates. Table 3-1 gives the ASHRAE (2001) design temperatures for these locations. These data were used to size cooling towers, as described in section 3.2.1.

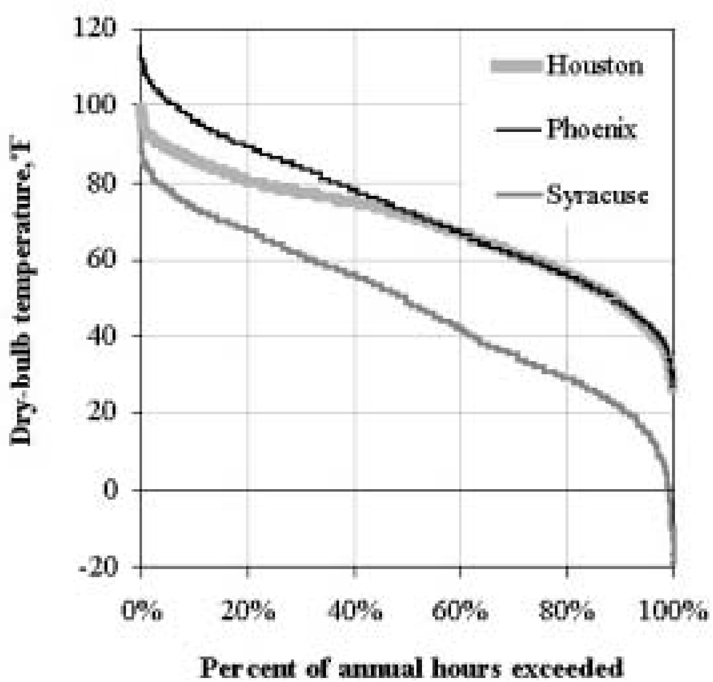


Figure 3-1: Dry-bulb temperature duration curves based on TMY2 data.

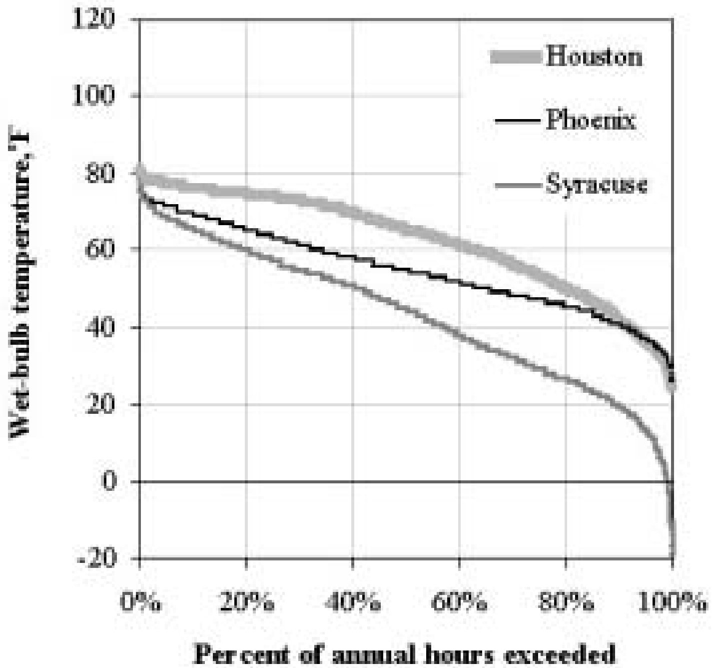


Figure 3-2: Wet-bulb temperature duration curves based on TMY2 data.

Table 3-1: Design temperatures for study sites (ASHRAE 2001)

Location	Syracuse	Houston	Phoenix
0.4% dry bulb with mean coincident wet-bulb, ˚F	88/72	96/77	110/70
0.4% wet bulb with mean coincident dry-bulb, ˚F	75/85	80/90	76/97

3.1.5 Chilled Water ÷T

For simplicity, chilled water ÷T was assumed to be a linear function of cooling load. The validity of this approach was demonstrated by comparison with actual operating data, as discussed in Section 3.3. Three scenarios were considered:

- Favorable ÷T
- Constant ÷T
- Unfavorable ÷T

Representative chilled water ÷T vs. load models are shown in Figure 3-3. The "favorable" scenario is so named because ÷T increases as load decreases. This reduces pumping energy in variable flow systems and ensures that chillers can be fully loaded. In the "unfavorable" scenario, ÷T decreases as load decreases, which increases pump energy consumption and may causes other system problems such as inability to load chillers in a primary/secondary plant (Fiorino 1996, Taylor 2002a).

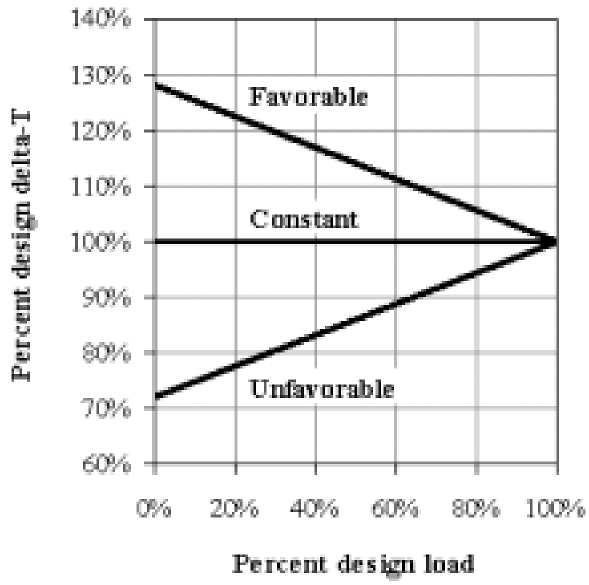


Figure 3-3: Chilled water ÷T characteristic curves.

3.1.6 Number of Chillers

The number of chillers in a plant affects its ability to match available capacity to load. The greater the number of chillers, the greater will be the average part load on operating machines. Office and medical facility models had from one to four equally sized machines in parallel. The district chilled water plant models had two to five equally sized chillers in parallel. The minimum number of chillers for the district plant cases was limited to two because of the low probability that a plant of more than 4,000 tons would have a single chiller.

3.2 Chilled Water System Simulation

The elements of a chilled water plant simulation are component models representing chillers, cooling towers, pumps; a system head vs. flow characteristic (system curve); component control algorithms approximating the control sequences of a real plant; and cooling load data. This section describes the equipment selection process and development of component models, control algorithms, load profiles, and ÷T vs. load models.

3.2.1 Equipment Selection

Major equipment was selected using the system design data in Table 3-2, ASHRAE cooling design weather data (ASHRAE 2001), and system head requirements dependent upon system configuration and facility type. Multiple components were equally sized and configured in parallel. Additional units delegated to standby capacity were not considered.

Circuit head loss estimate details used to size pumps are provided in Appendix C. A summary of the total value for each load type is included in Table 3-2 in the columns ‘CHW head’ and ‘CW head.’

Schedules of equipment selected for the various study load types, chiller configurations, and climates are also provided in Appendix C.

Table 3-2: Chilled water system design data.

Load, type	*Load, tons*	*CHW flow, gpm*	*CHW head, ft*	*CHW ÷T,˚F*	*Leaving CHW temp.,˚F*	*CW flow, gpm*	*CW head, ft*	*Range, ˚F*	*Entering CW temp.,˚F*
Office building	500	1,000	120	12	44	1,500	70	10	85
Medical facility	1,500	3,000	150	12	44	4,500	100	10	85
District plant	4,500	9,000	170	12	44	13,500	100	10	85

3.2.2 Component Models

Polynomial models were developed for key system components, i.e., cooling towers, pumps, and chillers. Model coefficients were determined by regression of manufacturers' performance data. System operating conditions are used to drive the models and compute component energy use.

Cooling Tower

The quantities of interest in a cooling tower model are the temperature of the water leaving the tower and the amount of power consumed by the fan. For multi-speed fan towers, the fan power is catalog information. The temperature of water leaving a cooling tower is a complex function of fill characteristics, air and water flow rates, ambient conditions, and heat load. Manufacturers customarily represent the performance of a specific tower with fixed airflow and water flow rates with characteristic curves that give leaving water temperature as a function of ambient wet-bulb temperature and range (fig. 3-4). For the required design conditions, tower curves were generated using manufacturer selection software (Marley 2000).

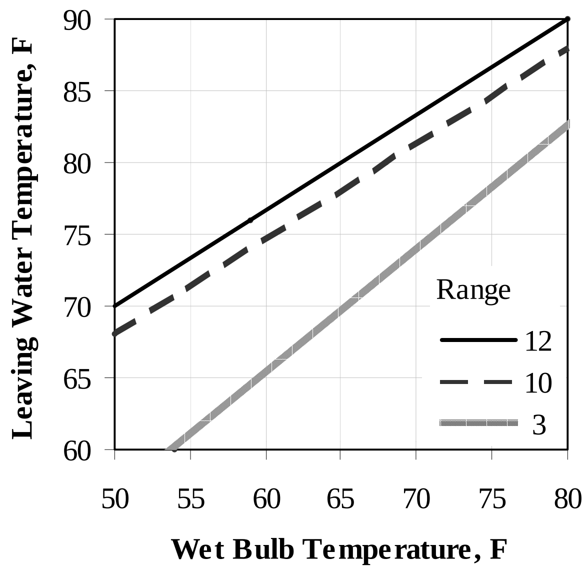


Figure 3-4: Cooling tower characteristic curves

Tower characteristic curves were modeled using a procedure described by Stoecker (1989) and summarized briefly here. For a given range, a tower characteristic can be approximated by a polynomial of low order, typically first or second order. In the present study, linear approximations were used:

$$T_{cw} = a_0 + a_1 T_{wb} \tag{3-1}$$

where

T_{cw} = entering condenser water temperature, ˚F

T_{wb} = ambient wet-bulb temperature, ˚F

a_0, a_1 = model coefficients based on range

Model coefficients are functions of range that can also be obtained by regression:

$$a_0 = b_0 + b_1 R + b_2 R^2 \tag{3-2}$$

$$a_1 = c_0 + c_1 R + c_2 R^2 \tag{3-3}$$

where

R = condenser water range, ˚F

b_0, b_1, b_2 = coefficients for function defining relationship of a_0 and range

c_0, c_1, c_2 = coefficients for function defining relationship of a_1 and range

For the two speed towers simulated, models of the above form were developed for both full and half speed operation. Tower model coefficients are tabulated in Appendix C. Manufacturers do not publish or certify tower performance with the fan off for the obvious reason that the airflow varies unpredictably in that operating mode. In order to simulate the operation of a typical condenser water system with fan cycling control, it was necessary to make an assumption about the fan-off performance of a tower. A tower manufacturer advised that it is reasonable to assume that the capacity of a tower with the fan off is five to ten percent of its capacity at full airflow (Mroch 2000). On the basis of this recommendation, it was assumed that the fan-off capacity of a tower was 5%.

Cooling towers frequently are controlled to produce a desired leaving water temperature by fan cycling. For given entering and ambient conditions, a tower will produce a unique temperature at each fan speed. In general, when water temperature exceeds set point, fan speed will increase and when water temperature falls below set point, fan speed will decrease. Cycling will occur between adjacent speeds. The fraction of time spent in each mode can be approximated by the proportions of water at the temperatures corresponding to the higher and lower fan speeds that give the desired temperature when mixed. The fan power consumed in cycling mode can then be determined by multiplying the fan power at each speed by the total duration of operation and the fraction of time spent operating at that speed.

Pump

The pump model predicts the power consumed by a pump/motor/variable speed drive combination for given flow conditions. Power is a function of flow rate, pump head, pump efficiency, motor efficiency, and drive efficiency (for variable speed operation):

$$HP_{pump} = \frac{\dot{Q}_{pump} \cdot H_{pump}}{3960 \cdot \eta_{pump} \cdot \eta_{motor} \cdot \eta_{drive}} \tag{3-4}$$

where

HP_{pump} = total pump power, hp

$\dot{Q}_{pump}$ = pump flow rate, gpm

H_{pump} = pump head, ft of water

3960 = unit conversion based on 68˚F water, $ft \cdot gpm/hp$

η_{pump} = pump efficiency

η_{motor} = motor efficiency

η_{drive} = drive efficiency

For constant flow pumps, the drive efficiency in Eqn. 3-4 is unity. In order to compute values of power from Eqn. 3-4, the head/flow/speed characteristic of the pump and the three efficiencies must be known. Eqn. 3-5 gives the head vs. flow characteristic for a pump at a nominal speed, N_0.

$$H_{pump,nom} = d_0 + d_1\dot{Q}_{pump,nom} + d_2\dot{Q}^2_{pump,nom} \tag{3-5}$$

where

$H_{pump,nom}$ = pump head at nominal pump speed, ft of water

$\dot{Q}_{pump,nom}$ = pump flow rate, gpm

d_0, d_1, d_2 = model coefficients based on nominal pump speed

The coefficients of Eqn. 3-5 are easily obtained by regression of several points on a standard pump curve (Appendix C). For a constant speed pump, Eqn. 3-5 is sufficient. This model can be extended to approximate the performance of a variable speed pump through use of the affinity laws to scale the nominal-speed characteristic. According to the affinity laws, the flow of the pump described by Eqn. 3-4 when operating at some other speed is:

$$\dot{Q}_{pump} = \dot{Q}_{pump,nom}\left(\frac{N}{N_0}\right) \qquad (3\text{-}6)$$

where N is an arbitrary speed and N_0 is nominal pump speed . Likewise, the head at arbitrary speed is:

$$H_{Pump} = H_{Pump,nom}\left(\frac{N}{N_0}\right)^2 \qquad (3\text{-}7)$$

Substitution of 3-6 into 3-5 to eliminate $\dot{Q}_{pump,nom}$ and substitution of 3-7 into 3-5 to eliminate $H_{pump,nom}$ or simultaneous solution of these equations by software gives the desired variable speed model. In similar fashion, the efficiency as a function of flow for a given nominal pump speed can be modeled by an equation of the form:

$$\eta_{pump} = e_0 + e_1\dot{Q}_{pump,nom} + e_2\dot{Q}_{pump,nom}^2 \qquad (3\text{-}8)$$

where

η_{pump} = pump efficiency

$\dot{Q}_{pump,nom}$ = pump flow rate at nominal speed N_0, gpm

e_0, e_1, e_2 = model coefficients based on pump speed

Again, this is sufficient for a constant speed pump. A model applicable to variable speed operation is developed by adopting the assumption that the efficiency of a point on the nominal speed characteristic remains unchanged as that point is mapped to other locations by the affinity laws. Model coefficients are provided in Appendix C.

A representative motor efficiency curve relating efficiency to fraction of nameplate horsepower (ASHRAE 2000) was used as the basis of the motor model (Figure 3-5).

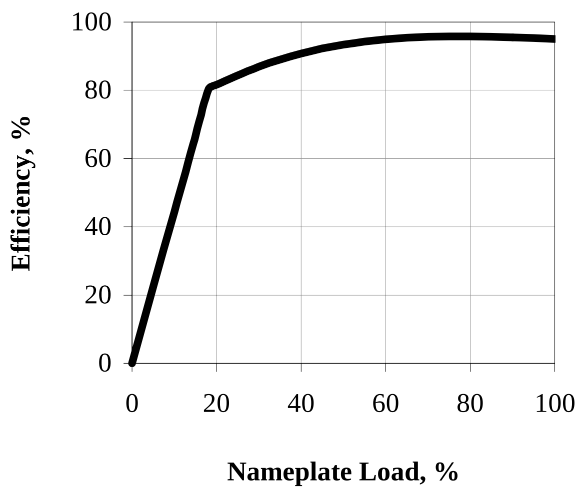


Figure 3-5: Model motor efficiency function (ASHRAE 2000).

The efficiency curve in Fig. 3-5 can be modeled by a piecewise continuous function:

$$\xi_{motor} = f_0 + f_1 F_{HP} + f_2 F_{HP}^2 + f_3 F_{HP}^3 \qquad \text{For } F_{HP} \geq 0.2 \qquad (3\text{-}9a)$$

$$\xi_{motor} = f_4 F_{HP} \qquad \text{For } F_{HP} < 0.2 \qquad (3\text{-}9b)$$

where

ξ_{motor} = motor efficiency

F_{HP} = fraction of nameplate motor horsepower

$f_0, f_1, f_2, f_{3,} f_4$ = model coefficients,

The generic variable speed drive efficiency curve (ASHRAE 2000) shown in Fig. 3-6 was taken as the basis for the drive model. This curve can be modeled by a second order polynomial with coefficients as given in Appendix C:

$$\xi_{drive} = g_0 + g_1 F_N + g_2 F_N^2 \qquad (3\text{-}10)$$

where

ξ_{drive} = drive efficiency

F_N = fraction of nominal speed

g_0, g_1, g_2 = model coefficients.

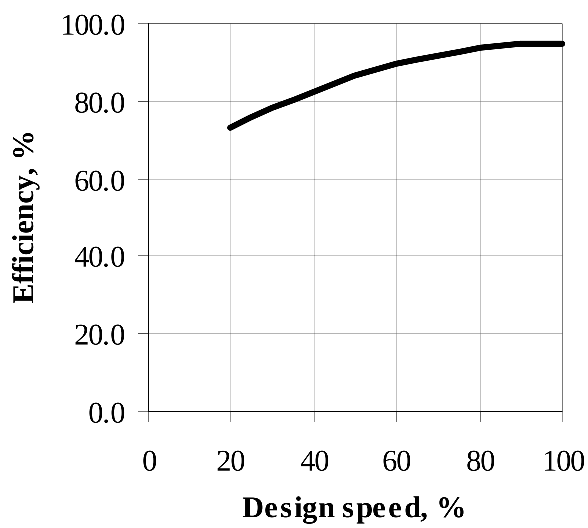


Figure 3-6: Model variable speed drive efficiency (ASHRAE 2000)

System Head

For simplicity, system head is modeled as a second order function of chilled water flow rate:

$$H_{system} = H_{control} + (H_{design} - H_{control}) \left(\frac{\dot{Q}_{system}}{\dot{Q}_{design}} \right)^2 \qquad (3\text{-}11)$$

where

H_{system} = system operating pressure, ft of water

$H_{control}$ = system control pressure, ft of water

H_{design} = system design pressure, ft of water

$\dot{Q}_{system}$ = system operating flow rate, gpm

$\dot{Q}_{design}$ = system design flow rate, gpm

The head and flow characteristic of a system actually forms a "head area" (Rishel 1996). However, for the purposes of estimating annual energy consumption, the simpler head curve approach, which does not require detailed modeling of the piping system and control devices, was adopted. There is no evidence in the literature that this simplification leads to significant error in the estimation of annual energy consumption.

Chiller

Chiller models predict compressor power based on given cooling load and entering condenser and evaporator water temperature. Models used in this study were empirical polynomial models of the form used in the DOE2 whole-building energy analysis program (DOE 1980). The full load chiller characteristics are modeled by functions CAPFT (Eqn. 3-12) and EIRFT (Eqn. 3-13). CAPFT is the available capacity of the chiller expressed as a fraction of the capacity at its rated condition. EIRFT is the full load power consumption expressed as a fraction of rated full-load power consumption. Both are dimensionless and have a value of unity at the chiller rating point. Both are functions of entering of condenser water temperature and leaving chilled water temperature. Typical EIRFT and CAPFT surfaces are shown in Figs. 3-7 and 3-8.

$$CAPFT = h_0 + h_1 \Delta t_{chws} + h_2 \Delta t_{chws}^2 + h_3 \Delta t_{cws} + h_4 \Delta t_{cws}^2 + h_5 \Delta t_{chws} \Delta t_{cws} \qquad (3\text{-}12)$$

$$EIRFT = j_0 + j_1 \Delta t_{chws} + j_2 \Delta t_{chws}^2 + j_3 \Delta t_{cws} + j_4 \Delta t_{cws}^2 + j_5 \Delta t_{chws} \Delta t_{cws} \qquad (3\text{-}13)$$

where

t_{chws} = chilled water supply temperature, ˚F

t_{cws} = condenser water supply temperature for water-cooled, ˚F

h_i, j_i = regression coefficients

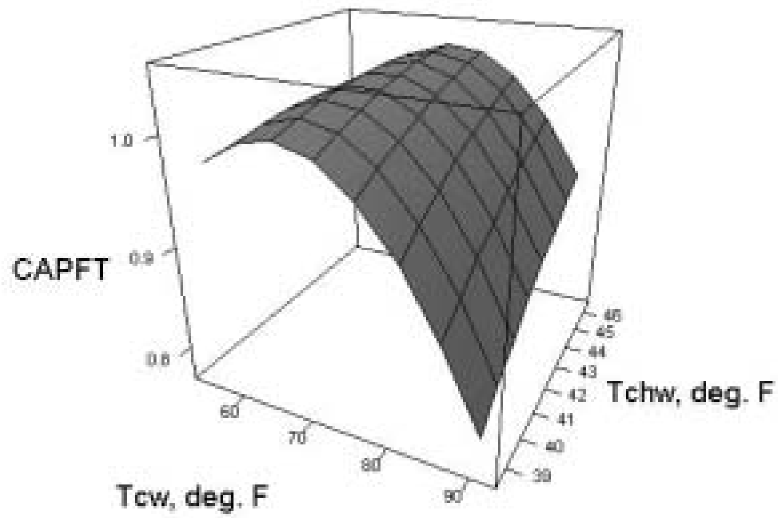


Figure 3-7: Typical available full load capacity fraction (CAPFT) surface

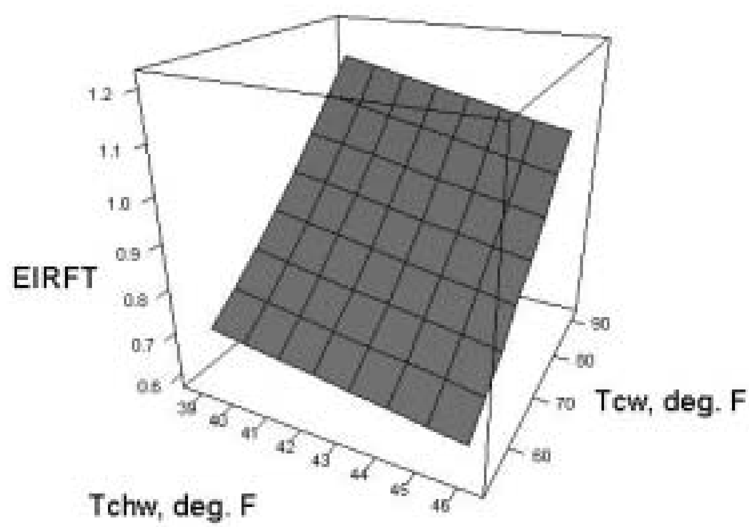


Figure 3-8: Typical full load chiller unit power fraction (EIRFT) surface

Part load performance of a chiller is modeled by the function EIRFLPR (Eqn. 3-14), which gives the fraction of full-load power as a function of the part load ratio, PLR, (Eqn. 3-15).

$$EIRFPLR = k_0 + k_1 \cdot PLR + k_2 \cdot PLR^2 \qquad (3\text{-}14)$$

$$PLR = \frac{Q}{Q_{ref} \cdot CAPFT(t_{chws}, t_{cws})} \qquad (3\text{-}15)$$

where

PLR = a function representing the part-load operating ratio of the chiller
Q = capacity, tons
Q_{ref} = capacity at the reference evaporator and condenser temperatures where the curves come to unity, tons
j_0, j_1, j_2 = regression coefficients.

Regression coefficients were generated with a spreadsheet program using manufacturers' data as input (PG&E 1998). Model coefficients are tabulated in Appendix C.

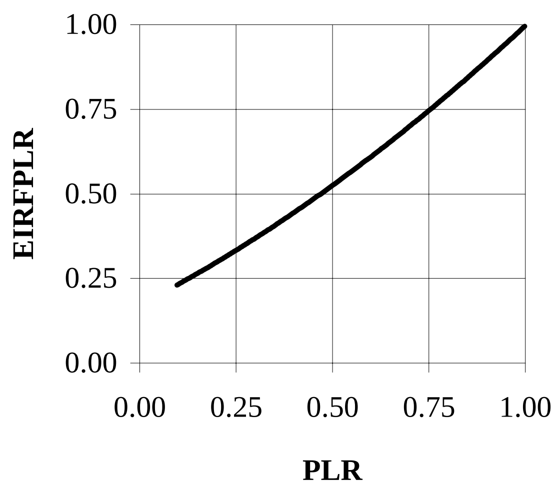


Figure 3-9: Typical part load power consumption (EIRFPLR) function

This model does not account for variable temperature differentials and flow rates in the evaporator and condenser. It was used because of clear evidence provided by manufacturers that the effect of these variations on power consumption is small. As discussed in Section 2.2.3, a chiller manufacturer's peer-reviewed analysis of simulation and test stand data (Redden 1996) as well as data provided by another manufacturer (Berry 2000) show that the effects of varying the evaporator flow rate on chiller efficiency are no larger than 2% for fixed entering condenser water and leaving chilled water temperatures.

3.2.3 Chilled Water System Models

Logic for sequencing and controlling chiller, cooling tower, and pump component models was developed for each system alternative using commercial equation solving software (Klein and Alvarado 2001). Programs listings may be found in Appendix D. Simulations are quasi-static, i.e., they are transient, but do not model true dynamic effects, i.e., the simulation proceeds from one hourly steady-state condition to the next. Hourly values of four parameters drive the simulation: ambient wet-bulb temperature, chilled water supply temperature, chilled water return temperature, and chilled water flow rate.

Control algorithms were based on conventional design practices as documented through literature review and interviews with the design engineers and other members of the HVAC industry. In addition to controlling dispatch of components, simulations calculate and save quantities of interest such as the hourly power consumption of plant components. A description of each algorithm follows.

Constant Flow, Primary-Only

Figure 3-10 shows a two chiller constant flow, primary-only chilled water system. Three-way valves at the loads bypass flow around cooling coils to control the supply air temperature. The simplest approach to control of this system type is to operate all chillers and their auxiliaries whenever there is a cooling load. In some cases, it is possible to stage chillers and pumps to reduce energy consumption, but this operating mode was not modeled. Chillers unload and load in order to maintain the chilled water temperature set point.

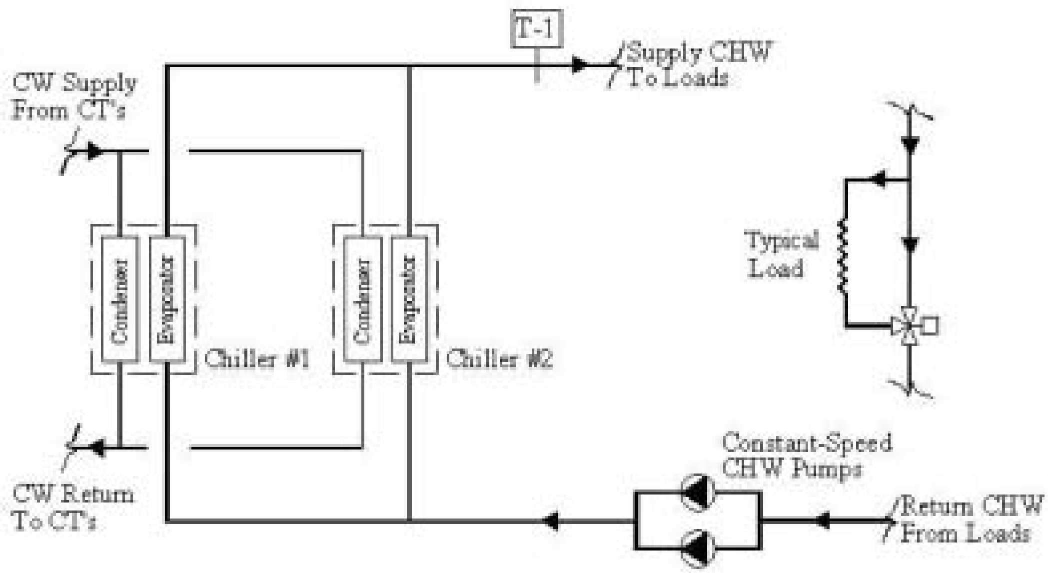


Figure 3-10: Constant flow, primary-only chilled water system

Constant Flow Primary/Variable Flow Secondary

Figure 3-11 shows a two-chiller constant flow primary/variable flow secondary system. The secondary chilled water pumps maintain a minimum differential pressure across the critical load. This is simulated by operating the secondary pumps at the system head (Eqn. 3-11) corresponding to the flow required for a given load and chilled water temperature difference.

Primary chilled water and condenser water pumps and cooling towers are sequenced with chillers. Each chiller in operation is equally loaded. Chillers are staged on in response to rising plant leaving chilled water temperature (T-1, Figure 3-11). A chiller is staged off when flow in the bypass exceeds the design flow of one chiller. In the simulation, this flow rate is calculated. In a real system, bypass flow can be calculated if bypass temperature (T-2), return temperature (T-3), mixed return temperature (T-4) and plant chilled water flow rate are known. Alternatively, a flow meter in the bypass line can be used to measure the excess primary flow directly.

Chiller staging is controlled by comparison of either the available chiller capacity with the calculated cooling load or of the chilled water flow through the primary circuit with that of the secondary circuit, whichever is critical. If the flow in the secondary circuit exceeds that of the primary circuit or if the load exceeds the available capacity then the number of chillers on line is increased. Once the number of chillers on line is determined, chiller, cooling tower, and pump energy is calculated using the appropriate component models (section 3.2.2).

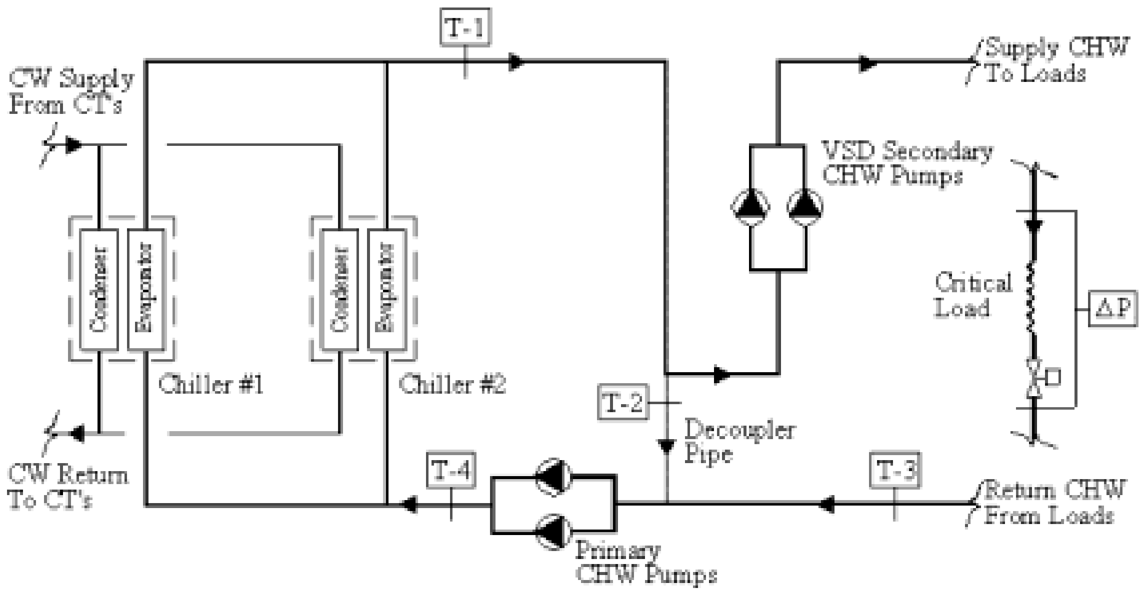


Figure 3-11: Constant flow primary/variable flow secondary chilled water system.

Constant Flow Primary/Variable Flow Secondary with Bypass Check Valve

The primary/secondary check valve system (Figure 3-12) is similar to the constant flow primary/variable flow secondary system. The addition of a check valve to the decoupler changes the control sequence for staging chillers and prevents return water from bypassing the chiller evaporators. Chillers are staged on if the chilled water temperature set point cannot be met or if the maximum chilled water flow rate is exceeded. Chillers are staged off when there is a surplus of chiller capacity online. When flow in the secondary exceeds flow in the primary, the check valve closes and the secondary and primary pumps are forced into series operation. In this condition, primary pumps operate beyond their design flow rate at a head lower than design. Secondary pumps provide the additional head needed to overcome plant head loss.

Variable Flow, Primary-Only

In the variable primary flow system (Figure 3-13) a single set of variable speed pumps serves both the chillers and load. Additional chiller capacity is brought on line if the chilled water temperature set point cannot be met or if the maximum chilled water flow rate through the evaporator is exceeded. Chillers are staged off when there is a surplus of chiller capacity online. This is determined by comparing the calculated cooling load and available chiller plant capacity. When the available capacity on line exceeds the calculated load by a quantity greater than or equal to one of the chillers in operation, then a chiller is staged off line. To permit operation of a single chiller below its low flow limit, a small bypass with a normally closed control valve is installed between the chilled water plant and the distribution piping. The bypass valve opens whenever flow through the evaporator of a chiller falls below the recommended minimum. The variable speed pumps act as distribution pumps, controlled by the same logic as secondary pumps in the primary/secondary system, i.e., pressure differential at remote cooling coils.

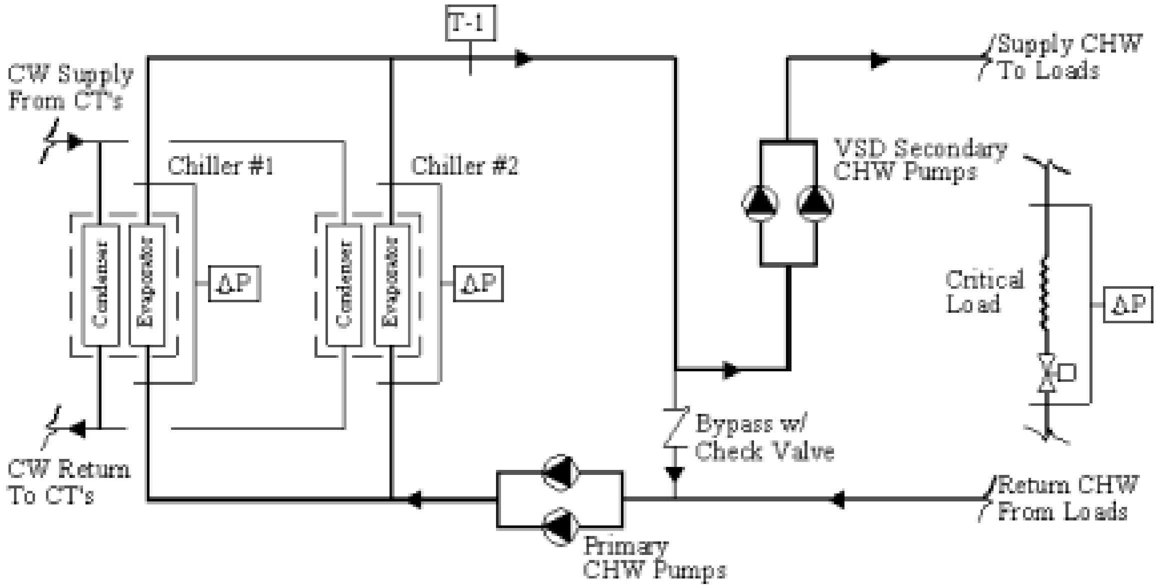


Figure 3-12: Constant flow primary/variable flow secondary with bypass check valve chilled water system

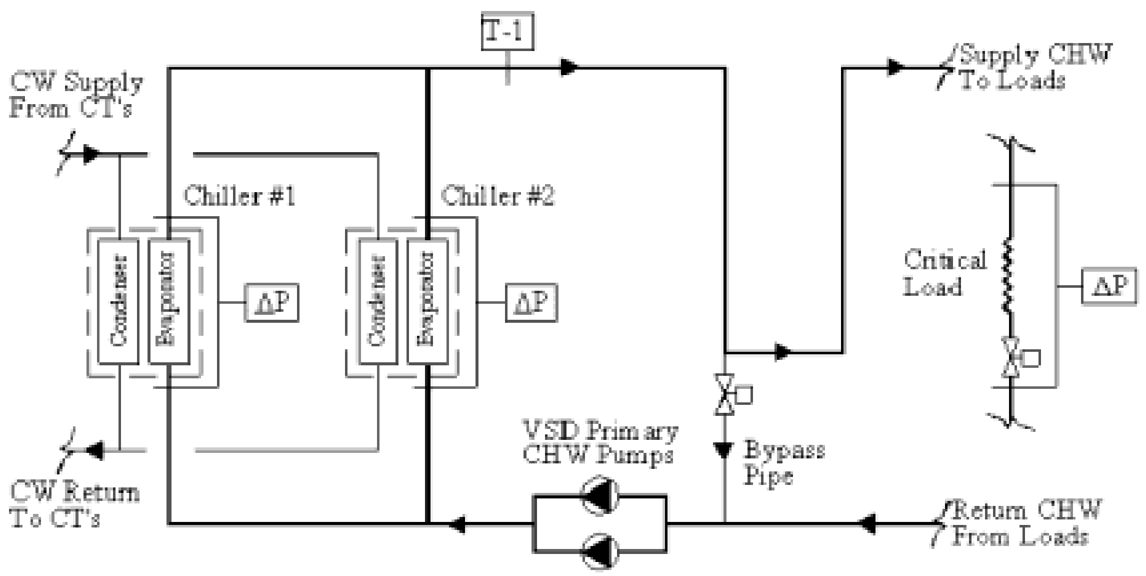


Figure 3-13: Variable flow, primary-only chilled water system

An algorithm of the type shown in Figure 3-14 controls each system simulation. In each simulation program the input data is read from and written to a table. The first step is to calculate the cooling load based on the data from the tabulated flow and temperature data. Calculated cooling load and the model inputs are used to determine the number of chillers on line. The logic in this subroutine varies with chilled water system type. The number of chillers on line is then used to determine the number of constant speed chiller auxiliaries on line. Chilled water flow rate determines the number of variable speed chilled water pumps on line. Subroutines implementing the component models defined previously are used to determine the power consumption.

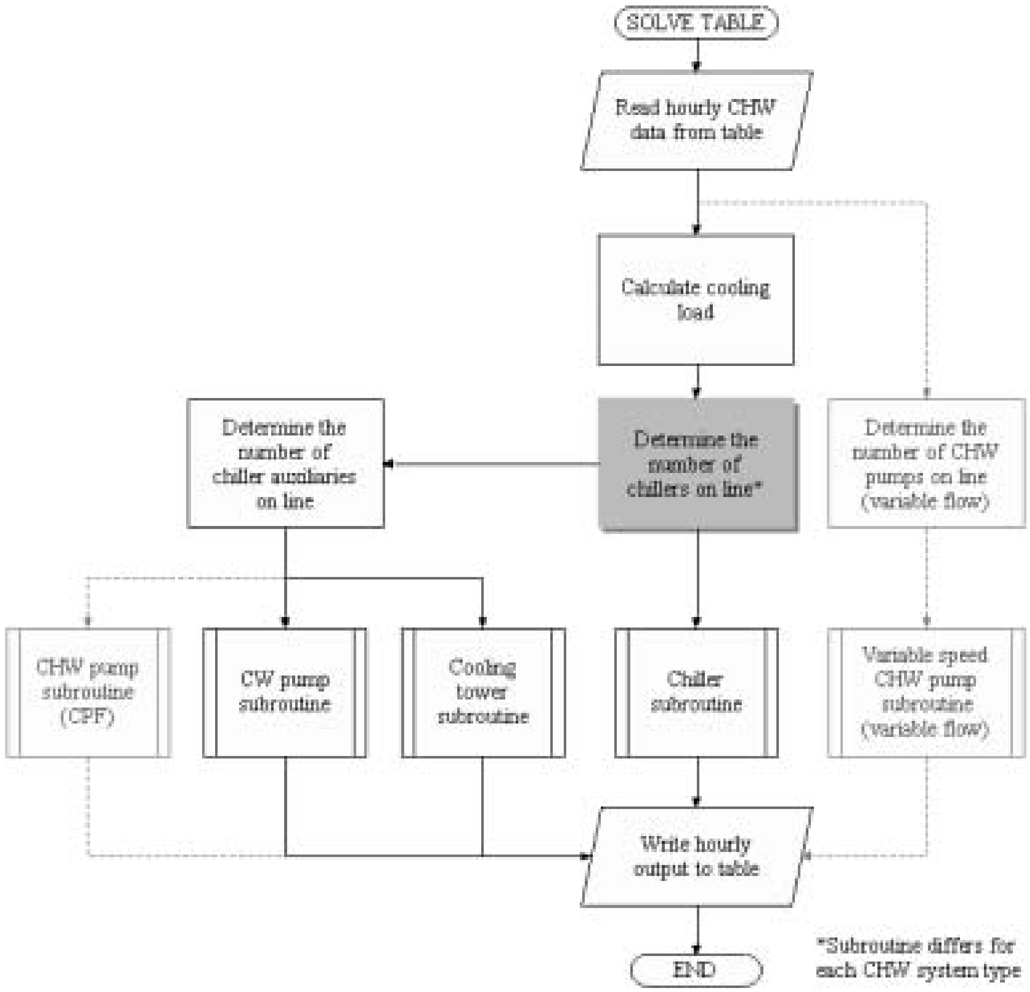


Figure 3-14: General chilled water system algorithm

As mentioned previously, the subroutine used to determine the number of chillers on line varies with system type. The variable flow, primary-only system staging algorithm is shown in Figure 3-15. Chilled water flow rate, cooling load, and entering condenser water temperature are input values. Chilled water flow rate is compared to the maximum and minimum rates of the chillers on line. If the flow rate is greater than the maximum, a chiller is added, and if the flow rate is lower, a chiller is removed. This process continues until the number of chillers on line can accommodate the flow rate. Once flow is established, the subroutine calls the cooling tower and chiller subroutines to solve for the range and calculate the chiller power.

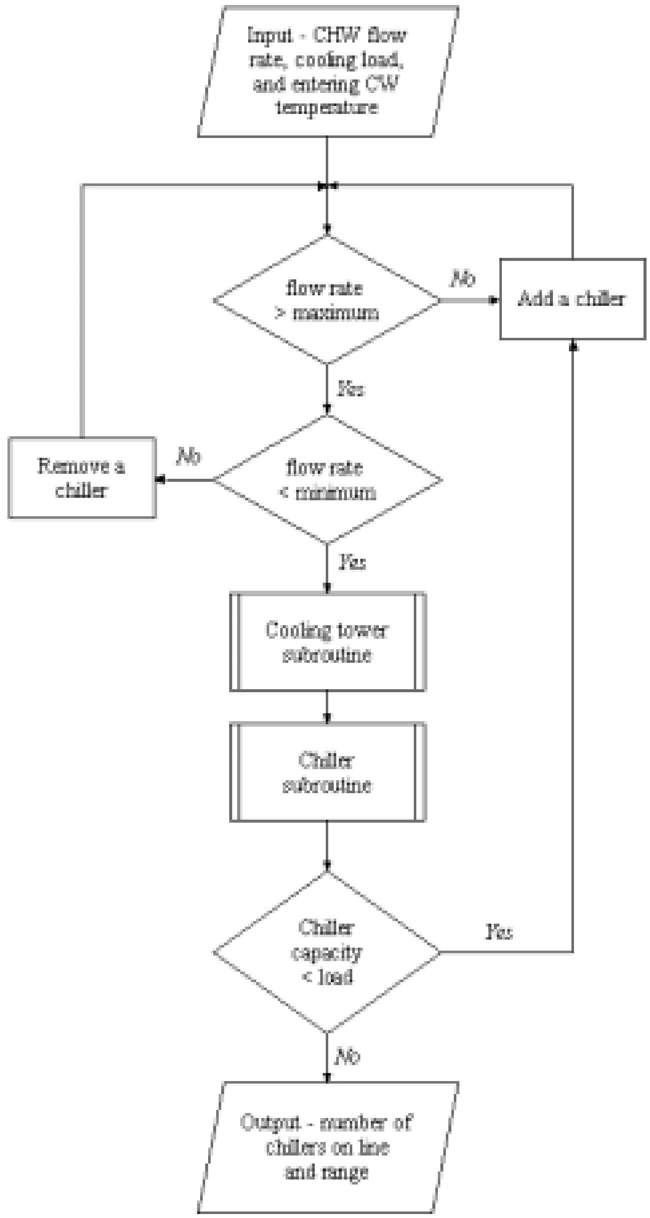


Figure 3-15: Algorithm used to determine the number of chillers on line for the variable flow, primary-only system

3.2.4 Load Data

Hourly load data used to exercise chilled water plant models were generated by computer simulations validated against actual measured load data. Loads were developed for the following cases:

- 500 ton Office building: Syracuse, NY; Houston, TX; Phoenix, AZ
- 1,500 ton Medical facility: Syracuse, NY; Houston, TX
- 4,500 ton District chilled water plant: Syracuse, NY; Houston, TX

This section discusses load generation and key load characteristics. Load details for each case can be found in Appendix B.

Cooling loads were generated using PowerDOE, an implementation of DOE-2.2 (PowerDOE 2000). Building and occupancy data were based on details provided by an engineering consultant (Banas 2000), templates in the PowerDOE program, and comparisons with the measured load data.

Two years of hourly data for five buildings connected to a district cooling system were obtained from the facilities department of a university located in Ithaca, NY(Little and Price 2000). The data included chilled water flow rate, supply and return water temperature, and ambient wet-bulb temperature for a veterinary science medical facility, a hotel, an office building, a chemistry laboratory building, and a physics laboratory building. Data from the office building and medical facility were used independently in the development of simulated loads for those types. Loads from all five buildings were added together to approximate a district cooling plant load.

As noted in section 3.1, Syracuse was selected as a study location in part because of its proximity to Ithaca. Before generating loads for other locations, each load type was modeled at the Syracuse location, compared with data from Ithaca, and tuned as necessary to give good agreement by adjustments to input parameters including internal loads, infiltration, quantity of outside air, and building dimensions. Differences between the weather data used in the simulation (Syracuse TMY2 data) and the actual weather for the study year sampled (Figures B-1 and B-2, Appendix B) resulted in differences in cooling load. Table 3-3 shows a comparison between the measured and simulated load data for the study load types.

Table 3-3: Comparison between cooling loads for load types using measured and simulated load data

Load type	Office building		Medical facility		District CHW plant	
Source	Measured	Simulated	Measured	Simulated	Measured	Simulated
Load, ton-hrs	865,081	946,593	1,522,855	1,629,305	11,839,311	11,868,037
÷ load, %	Base	9.4	Base	7.4	Base	0.2

Figures 3-16 through 3-18 are cooling load duration curves comparing the measured to simulated load data used to represent each of the three study load types. Although some differences are evident, values on the simulated load data duration curves are generally within 5-percent of the corresponding percent design load found on the curve representing the measured data.

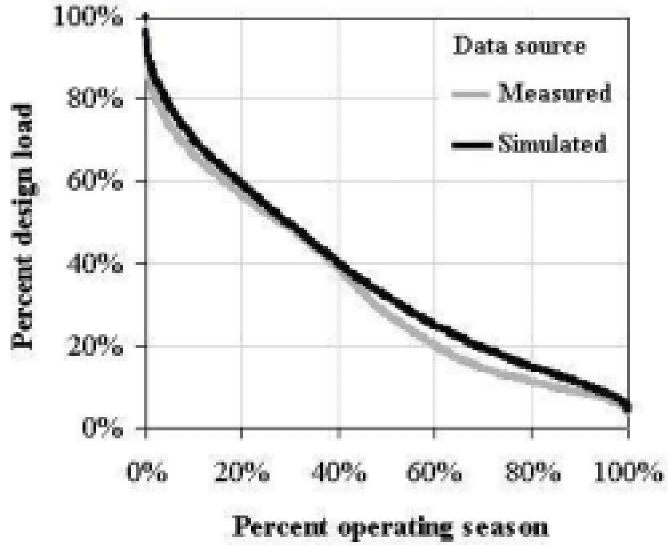


Figure 3-16: Measured (Ithaca NY) and simulated (Syracuse NY) cooling load duration curves for office building load type

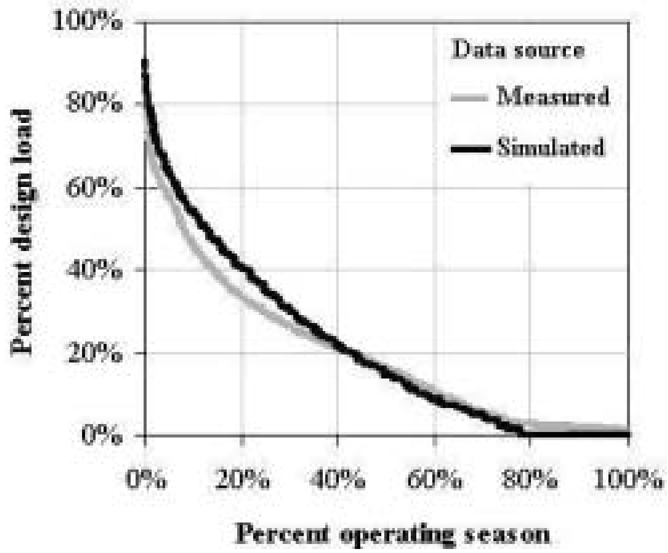


Figure 3-17: Measured (Ithaca NY) and simulated (Syracuse NY) cooling load duration curves for medical facility load type

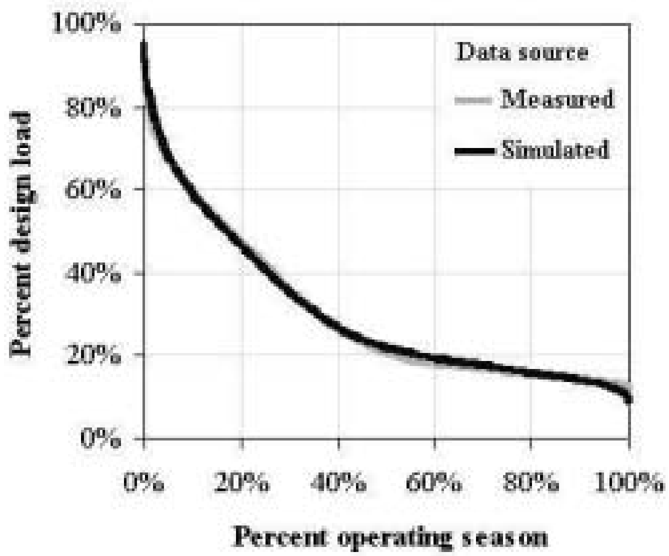


Figure 3-18: Measured (Ithaca NY) and simulated (Syracuse NY) cooling load duration curves for district chilled water plant load type

The design and peak cooling loads for each of the Ithaca data sets are shown in Table 3-4. Peak cooling loads are within 10-percent of the nominal tonnage. The building descriptions used to compare Syracuse simulations with Ithaca field data were also used to generate loads for the other study locations. Because of climate differences, this resulted in different values of peak load. Simulation results were scaled so that the same chilled water equipment could be used in each location and to facilitate comparisons between locations. This was accomplished by applying a multiplying factor to the load data from Syracuse, Houston, and Phoenix so that the peak load at these locations matched that of corresponding Ithaca load types.

Table 3-4: Design and peak cooling loads for the study load types from measured data

Load type	Office building	Medical facility	District CHW plant
Design cooling load, tons	500	1,500	4,500
Peak cooling load, tons	490	1,350	4,250

Table 3-5 summarizes simulated monthly cooling loads for Syracuse, Houston, and Phoenix. The Phoenix climate was used only to simulate load data for the office building. Despite differences in ambient humidity, the total loads imposed on the chilled water plant were very similar. Consequently, variable primary flow savings at the two locations were very similar in preliminary simulations using the office building model and it was deemed unnecessary to model the other load types at both locations.

Annual totals given in Table 3-5 show that, not surprisingly, equivalent loads had larger annual cooling requirements in Houston than in Syracuse. The medical facility annual cooling load was 2.8 times greater in Houston and the office building and district chilled water plant loads were both 1.6 times greater. The larger impact of climate on the medical facility cases is likely due to the greater quantity of outside air required. The other load types have higher ratios of internal to ventilation cooling load relative to the medical facility.

Table 3-5: Monthly cooling loads [ton-hrs] for Syracuse, Houston, and Phoenix

Load type	Office building			Medical facility		District CHW plant	
Location	Syracuse	Houston	Phoenix	Syracuse	Houston	Syracuse	Houston
January	42,004	100,004	102,746	13,807	183,020	504,110	1,025,151
February	43,534	92,973	95,894	15,043	102,066	489,869	837,549
March	49,883	122,883	133,922	27,222	220,372	580,359	1,179,911
April	80,557	152,382	149,844	31,105	362,163	719,312	1,504,087
May	114,059	186,917	180,998	113,888	523,541	1,019,043	1,906,967
June	153,941	214,436	223,402	297,661	632,531	1,470,121	2,165,442
July	181,753	225,525	241,359	440,880	735,012	1,833,174	2,384,868
August	177,091	216,102	232,457	430,758	696,869	1,798,099	2,286,708
September	144,301	201,556	208,050	250,717	559,779	1,360,506	2,009,424
October	94,891	169,322	169,897	64,296	414,154	860,730	1,674,664
November	64,134	133,160	122,660	42,296	257,615	666,714	1,273,353
December	51,416	102,937	102,139	26,468	149,622	566,000	989,338
Annual	1,197,564	1,918,197	1,963,369	1,754,141	4,836,744	11,868,037	19,237,462

Comparison of cooling load and outdoor air temperature data indicated that air handling units equipped with outside air economizer controls would have sufficient cooling capacity to meet the a majority of the load from November through March for the Syracuse office building and medical facility cases. Consequently, chilled water plant operation for these cases was limited to the period from April 1 through October 31 (5,136 hr). Other cases were assumed to have a year-long (8,760 hr) cooling season (Table 3-6).

Table 3-6: Chilled water plant operating season length (hours per year)

Location	*Office building*	*Medical facility*	*District CHW plant*
Syracuse	5,136	5,136	8,760
Houston	8,760	8,760	8,760
Phoenix	8,760	-	-

Plant cooling load duration curves for the office building (Figure 3-19) show the effect of economizer operation on the Syracuse load. In the hot Phoenix and Houston climates, plant cooling loads exist throughout the year, while economizer operation limits plant cooling loads in Syracuse to roughly 5000 hours per year. Duration curves for the medical center are shown in Figure 3-20. Changing climate from Syracuse to Houston also made significant changes in the medical facility load distribution, which went from having little to no cooling load for nearly 20-percent of the operating season to operating at 30-percent of the design load or greater for 50-percent of the operating season.

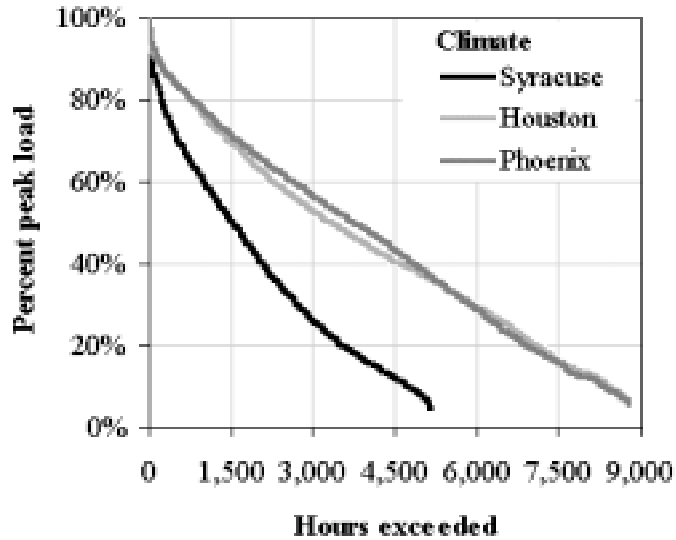


Figure 3-19: Plant cooling load duration curves for office building

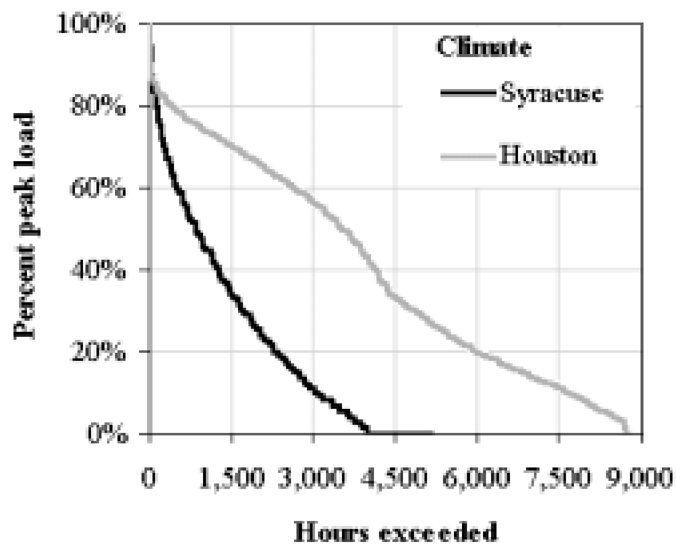


Figure 3-20: Plant cooling load duration curves for the medical facility

As noted in Table 3-6, the Syracuse and Houston cases for the district chilled water plant had the same cooling season length. However, the shape of the plant load duration curves for the two locations (Figure 3-21) are quite different. Since each system had essentially the same peak load, the difference is one of load factor (ratio of average to design or peak load). Monthly load factors based on design load are given for all study cases in Appendix B. Load factor is significant in this study because it is indicative of the extent to which flow in the system can vary and, therefore, the opportunity for pump energy savings in variable primary flow systems.

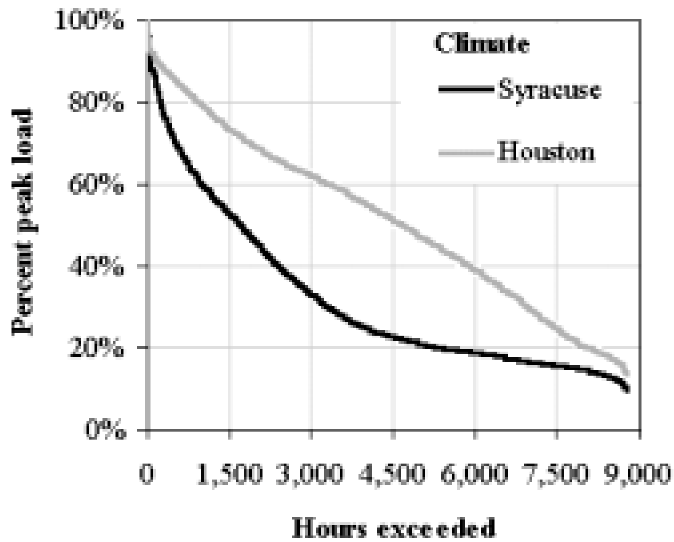


Figure 3-21: Cooling load duration curves for the district chilled water plant

3.2.5 Chilled Water ÷T

Chilled water ÷T was assumed to be a linear function of cooling load. This approach was validated using measured data from buildings located in Ithaca, NY. Figures 3-22 and 3-23 show the raw ÷T vs. load data, linear trend lines, and their equations for the office and district plant load types, respectively. The temperature differentials for the office building increased as the load decreased. This should be the trend for well-controlled loads with two-way valves, but for a variety of reasons, most chilled-water systems experience decreasing ÷T with decreasing load. In Figure 3-23 the opposite was true, as the temperature differential decreased with decreasing load.

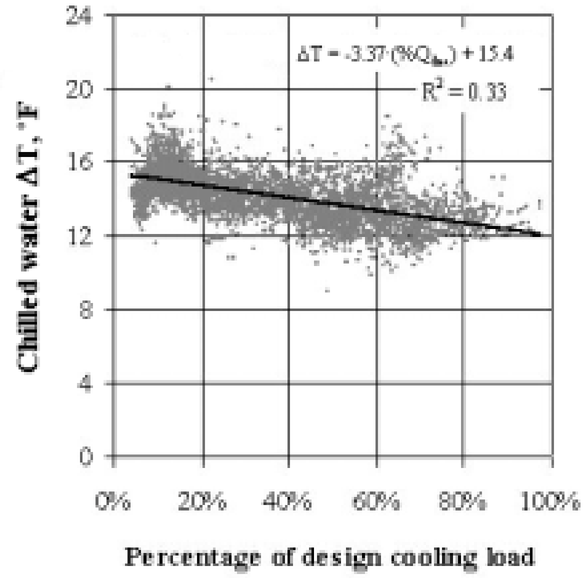


Figure 3-22: Measured chilled water temperature differentials for Ithaca office building load

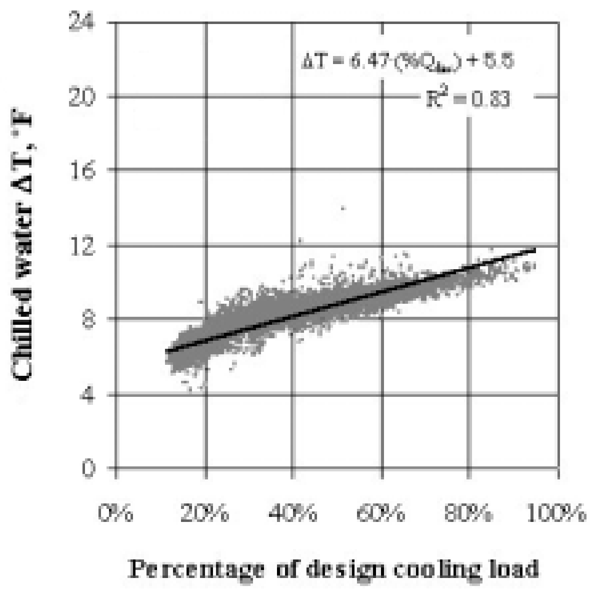


Figure 3-23: Measured chilled water temperature differentials for the Ithaca district plant

Figures 3-24 and 3-25 show the differences in plant performance as a result of the use of a linear model to represent actual measured data. Differences in total annual plant energy use, peak plant demand, average COP, and average number of chillers online were compared. The impact of the linear model was less than 2% for each of these performance characteristics.

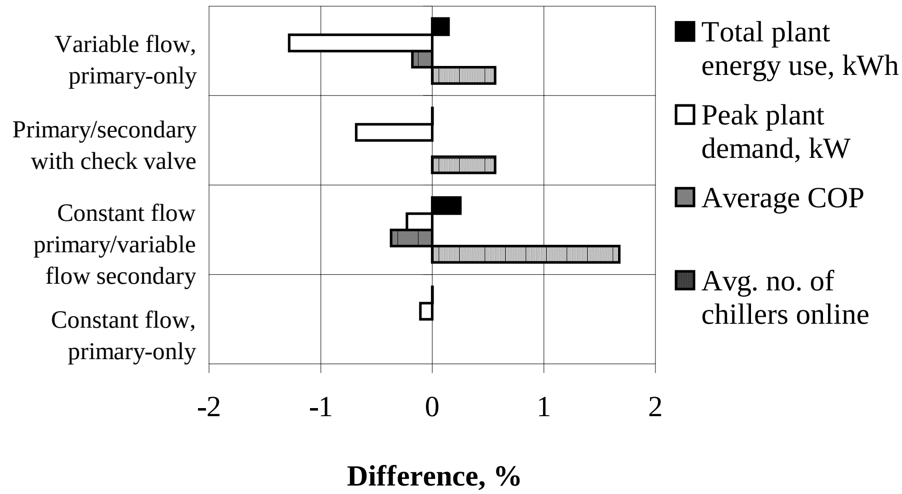


Figure 3-24: Differences in simulated plant performance due to use of linear ÷T vs. load model relative to measured data for Ithaca office building

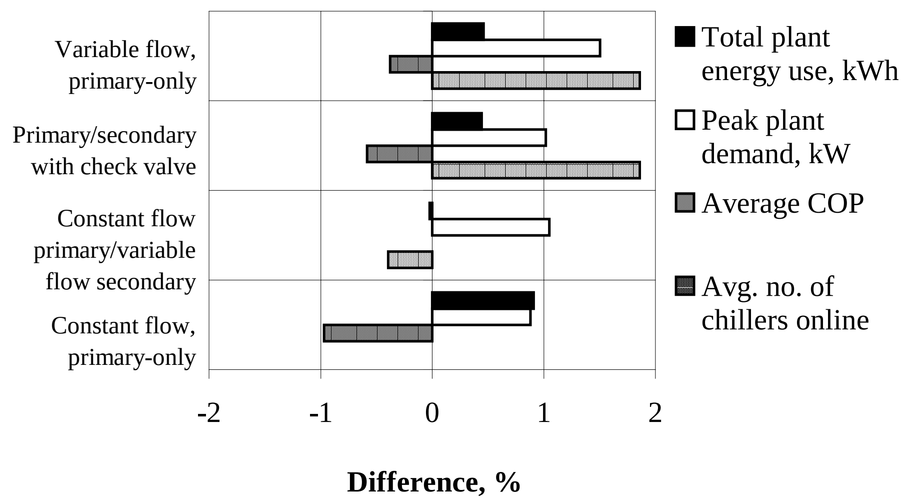


Figure 3-25: Differences in simulated plant performance due to use of linear ÷T vs. load model relative to measured data for Ithaca district cooling system

Having confirmed the validity of using linear ÷T vs. load approximations, this approach was used to investigate the effect of ÷T trends on variable primary flow energy savings using the constant, favorable and unfavorable characteristics shown in Figure 3-26. The constant case is a constant ÷T model. The favorable ÷T model increase by 4∀F as load decreases to zero and the unfavorable characteristic decreases by 4∀F as load falls to zero, both relative to a 12∀F base ÷T.

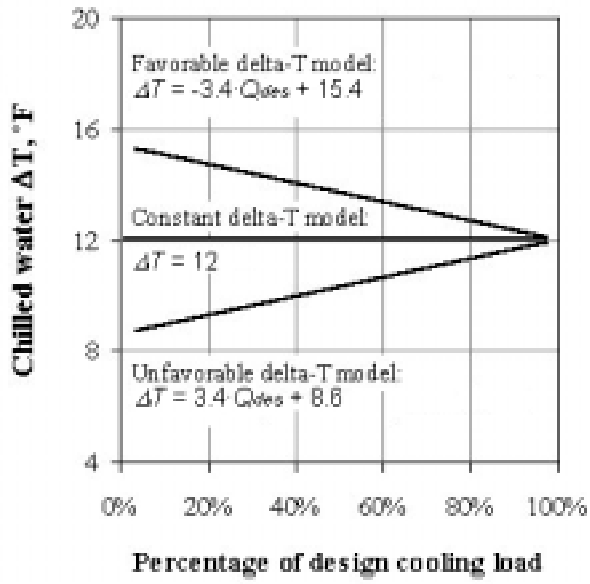


Figure 3-26: Chilled water ÷T models

3.3 Simulation Matrix

In total, 660 cases are generated by the possible combinations of the study parameters (Table 3-7). Preliminary studies to determine which parameters produced distinctive results permitted reduction of the final matrix to 348 cases.

Table 3-7: Summary of simulation cases

Parameter	*Maximum matrix size*		*Final study matrix*		
	Simulated data	*Measured data*	*Core simulations*	*Additional simulations*	
				Phoenix climate	*Measured data*
System type[1]	4	4	4	4	4
No. of chillers[1]	5	5	4	3	3
Load type[1]	3	3	3	1	3
Climate[1]	3	1	2	1	1
CHW ÷T[1]	3	2	3	2	1
Unique Combinations[2]	540	120	288	24	36
Total number of cases	660		348		

Note: [1]Number of parameter values [2]Product of all parameter value numbers in column

The core group of simulations in the reduced-size matrix included the four system types, three load types, two climates, three chilled water ÷T characteristics, and four chiller configurations with the exception that primary/secondary system with a check valve cases were limited to cases with multiple chiller configurations and an unfavorable chilled water ÷T characteristic. This was done because when chilled water ÷T is constant or favorable, a chiller will generally be capacity limited before it is flow limited. There is no advantage to the check valve configuration in such cases unless the ÷T is low even at full load, a real possibility that was not analyzed in this study.

The office building and constant ÷T model were selected as the basis of comparison for all load types and ÷T characteristics investigated.

The core group of simulations included two climates—Houston and Syracuse. Houston was selected as the basis of comparison because all load types simulated there had an 8760-hour cooling season. This permitted direct comparison of load types without the need to account for variation in cooling season length. Additional office building simulations were performed with Phoenix weather and compared to Houston results to determine whether there were significant differences between a dry climate and humid climate. Table 3-8 shows the average annual kW/ton of the variable primary flow and primary/secondary system types and the energy saved by the variable primary flow system relative the constant flow primary/variable flow secondary system.

Table 3-8: Variable primary flow (VPF) plant energy savings relative to constant flow primary/variable flow secondary system

Location	Houston	Phoenix
VPF, annual kW/ton [1]	0.78 to 0.65	0.73 to 0.61
Primary/secondary, annual kW/ton [1]	0.82 to 0.68	0.77 to 0.63
VPF savings relative to primary/secondary, kW/ton [1]	0.04 to 0.02	0.04 to 0.02
VPF annual energy savings relative to primary/secondary, % [1]	4.6 to 3.4	4.8 to 3.5

Note: [1]Ranges represent values for 1 through 3 chiller configurations

Variable primary flow energy savings for the two climates differed by no more than 0.2-percent. Although latent load contributes significantly to chilled water plant load, the plant itself is indifferent to whether the source is latent or sensible as the comparison of Houston and Phoenix results shows. Consequently, Phoenix was not included in the core set of simulations.

3.4 Simulation Results

Figure 3-27 shows the system head vs. flow characteristic for the Houston office building three chiller variable flow, primary-only system. Discontinuities in the system curve are flows at which chillers and pumps are staged on and off. It is a characteristic of single pump variable flow systems with parallel chillers that the system head drops significantly when a new chiller is activated.

Figure 3-28 shows the number of chillers running as a function of plant chilled water flow rate for constant and variable flow three chiller configurations. This figure illustrates two sources of variable primary flow system energy savings. The first is that most of the operation of the constant flow plant is at flow rates greater than the system flow rate. In the case of a variable primary flow system the plant flow rate and the system flow rate are identical. The second potential energy savings can occur when the flow through chillers in a variable primary system exceeds the design flow rate of an equivalent constant primary flow system. In this case, chillers are fully loaded and auxiliary energy consumption is reduced because additional pumps and cooling towers need not be started.

Table 3-9 summarizes the distribution of total plant energy consumption among plant components. Ranges representing all study cases are given. Location has little effect on the distribution of energy use. Load type and other factors such as the number of chillers in the plant are more significant. Chiller auxiliary energy is the total of the cooling tower and condenser water pump energy. Chilled water pump energy is the total consumed by primary and, where applicable, secondary pumps. For all load types, chillers are by far the greatest consumers of energy. Auxiliaries are generally the next most significant energy user, followed by chilled water pumps.

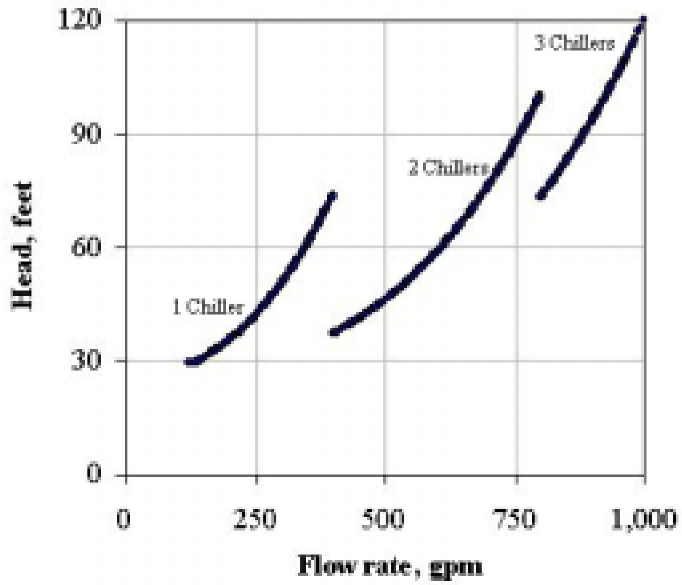


Figure 3-27: Chilled water system curves for Houston office building three chiller variable flow, primary-only system

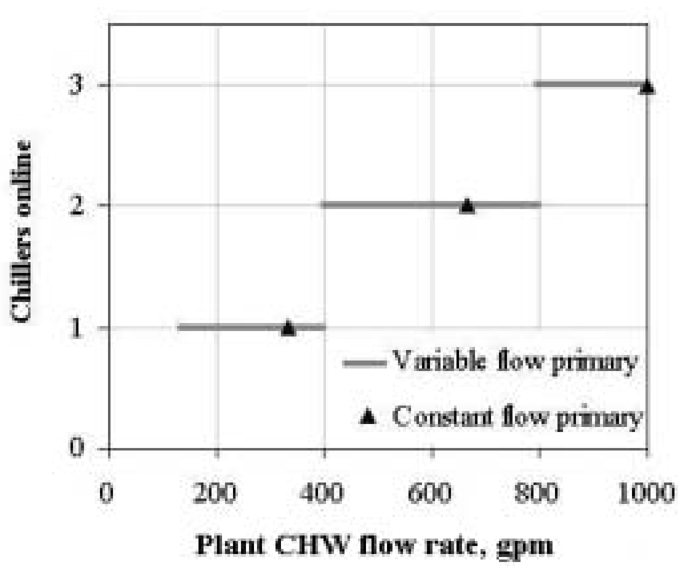


Figure 3-28: Chillers online as a function of plant chilled water flow rate for a three-chiller variable flow, primary-only and constant flow primary/variable flow secondary plant (Houston office building case)

Table 3-9: Percentage of total plant energy use by component.

Load type		Office building	Medical facility	District plant
Syracuse	*Chiller energy, %* [1]	60 to 78	49 to 72	49 to 74
	CHW pump energy, % [1]	6 to 18	6 to 22	6 to 25
	Chiller aux. energy, % [1]	16 to 25	22 to 34	20 to 27
Houston	*Chiller energy, %* [1]	62 to 77	55 to 73	57 to 74
	CHW pump energy, % [1]	5 to 15	5 to 18	5 to 19
	Chiller aux. energy, % [1]	17 to 25	21 to 31	20 to 26

Note: [1]Ranges represent values for all four study chiller configurations

3.4.1 Effect of Number of Chillers

Table 3-10 summarizes the plant and component energy consumption for the Houston office building with constant ÷T. The effect of number of chillers on savings generated by variable primary flow systems in this case is typical. For a given variable flow system type, an increase in the number of chillers in the plant generally has the effect of reducing the energy consumption. The constant flow, primary-only system has the same energy consumption regardless of the number of chillers operating because all equipment was operated continuously. An example of the impact of chiller staging on constant flow primary-only systems has been published previously by the authors (Bahnfleth and Peyer 2001, 2003).

Table 3-10: Total annual plant energy consumption, kWh/design ton, for the Houston office building with constant ÷T

Number of chillers		1	2	3	4
Constant flow, primary-only	*Chiller energy, kWh/ton*	2,073	2,073	2,073	2,073
	CHW pump energy, kWh/ton	515	515	515	515
	Chiller aux. energy, kWh/ton	742	742	742	742
	Total plant energy, kWh/ton	3,330	3,330	3,330	3,330
Constant flow primary/ variable flow secondary	*Chiller energy, kWh/ton*	2,073	1,931	1,909	1,898
	CHW pump energy, kWh/ton	343	258	236	223
	Chiller aux. energy, kWh/ton	741	541	491	466
	Total plant energy, kWh/ton	3,157	2,730	2,636	2,587
Primary/secondary with a check valve	*Chiller energy, kWh/ton*	2,073	1,930	1,910	1,901
	CHW pump energy, kWh/ton	345	255	232	219
	Chiller aux. energy, kWh/ton	741	524	471	448
	Total plant energy, kWh/ton	3,160	2,708	2,613	2,568
Variable flow, primary-only	*Chiller energy, kWh/ton*	2,073	1,930	1,910	1,901
	CHW pump energy, kWh/ton	199	165	170	170
	Chiller aux. energy, kWh/ton	741	524	469	446
	Total plant energy, kWh/ton	3,013	2,618	2,550	2,517

Figure 3-29 shows the total plant energy saved as the number of chillers increases from one to four. Solid lines identify the energy savings relative to the single chiller configuration and dotted lines represent the incremental savings generated by increasing the number of chillers by one unit (e.g., the three chiller configuration of the variable primary flow system type resulted in an energy savings of approximately 3-percent relative to the two chiller configuration). It can be seen that, although the energy savings continue to increase when design plant load is distributed between more than two chillers, the incremental savings diminish rapidly.

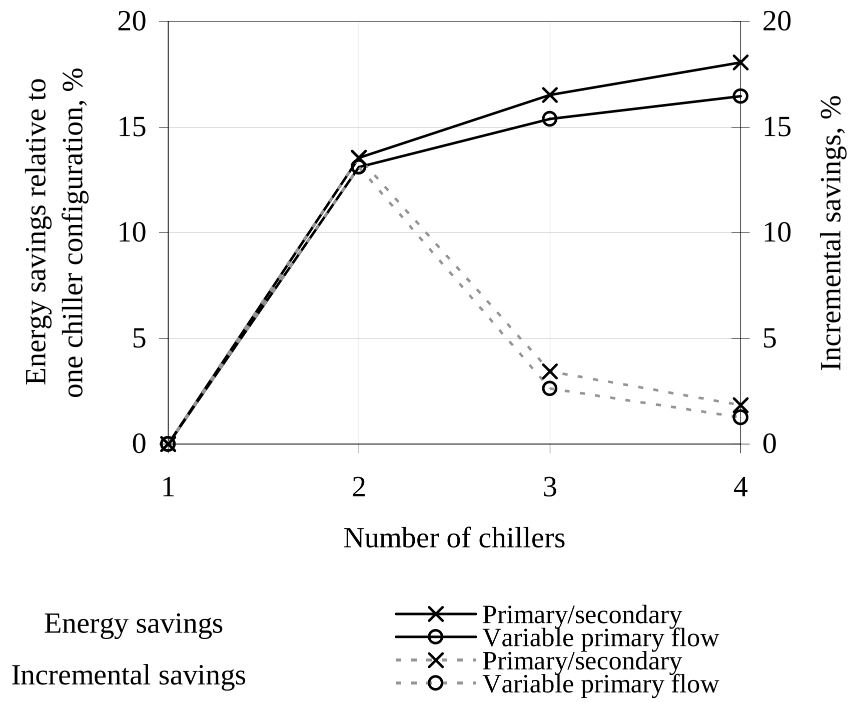


Figure 3-29: Total plant energy savings generated by increasing the number of chillers for Houston office building with constant ÷T

Other cases produced similar patterns for energy savings as a function of number of chillers. Table 3-11 summarizes the energy savings resulting from an incremental increase in number of chillers. Values represent the range of results from all cases simulated in the study. As was the case in the example presented above, the jump in energy savings was greatest when going from the one chiller configuration to the two chiller configuration. Energy savings diminished to between 1 and 3-percent when going from four to five chillers. These results were similar to those reported in similar study (Beyene and Lowrey 1994). For the cases Beyene and Lowrey documented, the incremental energy savings of increasing the number of chillers from one to two was approximately 13-percent. Incremental energy savings for increasing from two to three chillers was not more than 3-percent.

Table 3-11: Range of incremental plant energy savings (%) due to change in number of chillers for all study cases.

Comparison	From 1 to 2	From 2 to 3	From 3 to 4	From 4 to 5
+ ÷ *plant energy, Constant flow primary/ variable flow secondary*[1]	11 to 26	3 to 9	1 to 5	1 to 3
+ ÷ *plant energy, Primary/secondary with a check valve*[1]	14 to 26	5 to 9	2 to 5	1 to 2
+ ÷ *plant energy, Variable flow, primary-only*[1]	13 to 24	4 to 9	2 to 4	1 to 2

Note: [1]Ranges represent values for all study cases

Figure 3-30 shows plant energy savings for the base case constant flow primary/variable flow secondary system with a check valve and variable flow, primary-only system relative to the constant flow primary/variable flow secondary system. The variable flow, primary-only system realizes a maximum energy savings of 5.2-percent in the one chiller case. This savings diminishes with the addition of each unit of chiller capacity down to 2.4-percent for the four chiller case.

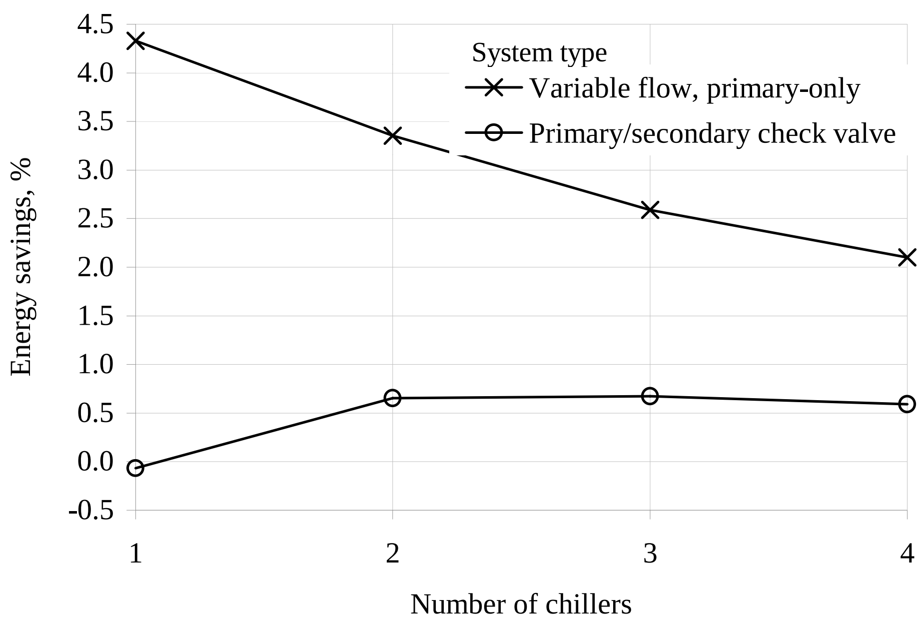


Figure 3-30: Annual variable primary flow plant energy savings (%)relative to constant flow primary/variable flow secondary system for Houston office building

Figure 3-31 shows the base case component energy savings for the variable flow, primary-only and primary/secondary check valve systems. Savings are relative to the constant flow primary/variable flow secondary system. In the case of the variable flow, primary-only system, chilled water pump energy account for most of the savings. Chiller auxiliaries make up a small portion of the savings in the multiple chiller configurations. Figure 3-31 clearly shows that diminishing return is due primarily to declining incremental savings of chilled water pump energy. In the case of the primary/secondary check valve system, the pump energy savings contribute only a small amount in the multiple chiller configurations, most of which are generated by the chiller auxiliaries. In both cases, the chiller does not contribute to the energy savings.

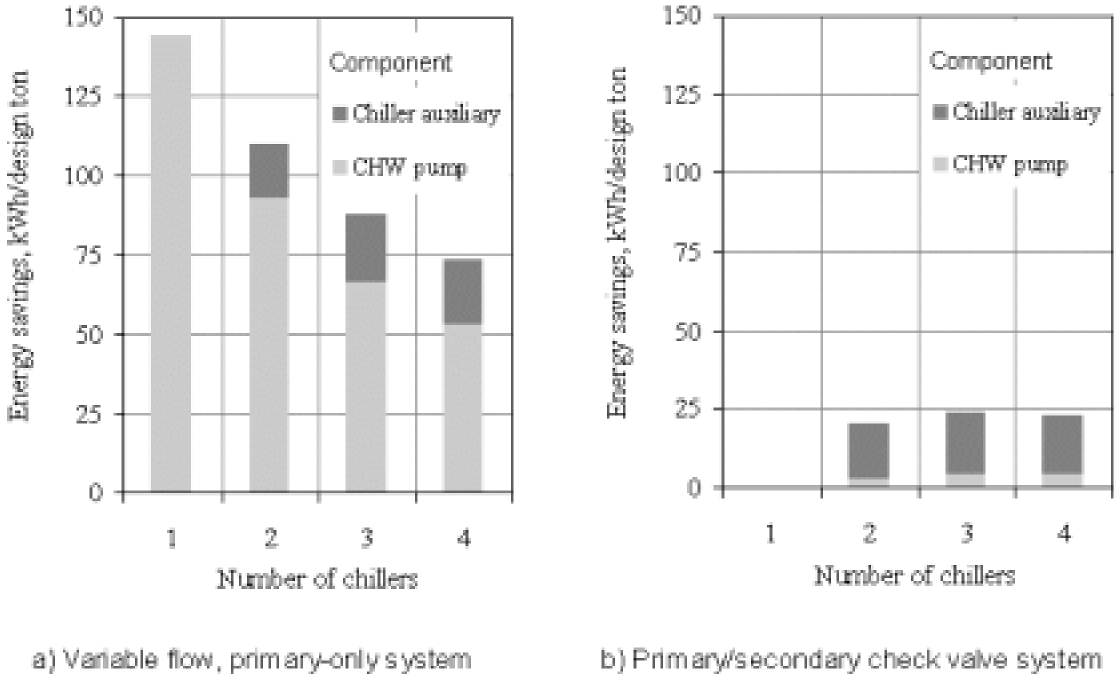


Figure 3-31: Annual plant energy savings (kWh/design ton) for variable primary flow systems relative to constant flow primary/variable flow secondary system for Houston office building

In summary:

- Total plant energy decreased with increased number of chillers.
- Increasing the number of chillers from one to two units provided the greatest incremental savings for all variable flow system types relative to other changes to chiller configuration.
- Variable primary flow savings relative to constant flow primary/ variable flow secondary systems diminished with increased number of chillers, although both benefited from having more than one chiller.

3.4.2 Effect of Chilled Water ÷T

Table 3-12 summarizes the impact of the three study chilled water ÷T models on plant energy consumption for each of the system types. Constant ÷T cases were used as the basis for comparison. Ranges represent the change in energy consumption for all four study chiller configurations. The Houston office building load type was used to represent the trends found in all study cases. Because all equipment was on continuously in constant flow, primary-only systems, the change in ÷T characteristics could have no effect on the energy consumption in these cases. All variable flow system alternatives experienced changes in energy consumption due to the ÷T behavior of the load.

Table 3-12: Plant energy use relative to constant flow primary/variable flow secondary system for study chilled water ÷T models of the Houston office building.

Chilled water ÷T model	Favorable ÷T	Constant ÷T	Unfavorable ÷T
Constant flow, primary-only plant energy, kWh/ton [1]	3,330	3,330	3,330
÷ plant energy, % [1]	0	Base	0
Constant flow primary/variable flow secondary plant energy, kWh/ton [1]	3,150 to 2,553	3,157 to 2,587	3,194 to 2,836
÷ plant energy, % [1]	0 to -1	Base	1 to 10
Primary/secondary check valve plant energy, kWh/ton [1]	3,152 to 2,554	3,160 to 2,568	3,196 to 2,725
÷ plant energy, % [1]	0 to -1	Base	1 to 7
Variable flow, primary-only plant energy, kWh/ton [1]	3,005 to 2,481	3,013 to 2,517	3,062 to 2,666
÷ plant energy, % [1]	0 to -1	Base	2 to 6

Note: [1]Ranges represent values for cases with 1 to 4 chillers

The annual plant energy consumption of each of the system types increased as the ÷T went from favorable to unfavorable. The variable flow, primary-only system remained the smallest energy consumer of the system types modeled for each of the ÷T cases.

Table 3-13 shows the impact of favorable and unfavorable ÷T models on annual energy consumption for the Houston office case. When compared to systems simulated with constant ÷T, the favorable ÷T model had little to no impact on the plant energy consumption. In all cases, the chiller and chiller auxiliary operation were relatively unaffected when ÷T was at or above design.

Table 3-13: Impact of favorable and unfavorable ÷T models on annual energy consumption for the Houston office case. Differences calculated relative to constant ÷T case.

CHW system type	Constant flow primary/ variable flow secondary		Constant flow primary/ variable flow secondary with check valve		Variable flow primary-only	
÷T model	Favorable	Unfavorable	Favorable	Unfavorable	Favorable	Unfavorable
÷ chiller energy, % [1]	0	0 to 2	0	0 to 1	0	0 to 1
÷ CHW pump energy, % [1]	-2 to -8	10 to 40	-2 to -5	11 to 36	-1 to -13	24 to 40
÷ chiller aux. energy, % [1]	0 to -4	0 to 27	0	0 to 16	0	0 to 16
÷ total plant energy, % [1]	0 to -1	1 to 10	0 to -1	1 to 7	0 to -1	2 to 6

Note: [1]Ranges represent range for 1 to 4 chillers.

Relative to constant ÷T, variable flow system types consumed less chilled water pump energy with a favorable ÷T characteristic. The chilled water pump energy of the variable flow, primary-only system had the greatest change from favorable to unfavorable ÷T. This is due to comprehensive variable speed pumping, which provides greater pump energy saving potential than is possible with systems having constant speed primary pumps.

All systems simulated with an unfavorable ÷T saw an increase in component energy relative to those using a constant ÷T model (Table 3-13). Chiller and chiller auxiliary energy consumption increased because chillers were flow limited at times when ÷T's were sufficiently low. The result is that at times there are more chillers and chiller auxiliaries operating than necessary to meet the cooling load. The average number of chillers required by each system type of the Houston office building case is shown in Table 3-14. As the ÷T characteristic degrades the average number of chillers increases. This is particularly true for the conventional flow primary/variable flow secondary system. In the four chiller configuration, for example, the average number of chillers increased from 2.3 for the constant ÷T case to 2.9 for the unfavorable ÷T case. The result was an increase in chiller auxiliary energy of 27-percent (Table 3-13).

Chilled water flow rates increased to account for less-than-design ÷T's, thereby increasing pump energy consumption. For the same reason that the variable flow, primary-only system produced the greatest chilled water pump energy savings when comparing the favorable ÷T case to that of the constant ÷T case, the variable flow, primary-only system also had the greatest increase in pump energy when comparing the unfavorable ÷T case to the base case (Table 3-13).

Table 3-14: Annual average number of chillers online for the Houston office case.

CHW system type	Constant flow primary/variable flow secondary				Const. flow primary/ var. flow secondary with check valve				Variable flow, primary-only			
Number of chillers in plant	1	2	3	4	1	2	3	4	1	2	3	4
Avg. no. of chillers online, favorable ÷T	1.0	1.3	1.8	2.2	1.0	1.3	1.8	2.2	1.0	1.3	1.8	2.2
Avg. no. of chillers online, constant ÷T	1.0	1.4	1.9	2.3	1.0	1.3	1.8	2.2	1.0	1.3	1.8	2.2
Avg. no. of chillers online, unfavorable ÷T	1.0	1.7	2.3	2.9	1.0	1.6	2.0	2.5	1.0	1.6	2.0	2.5

Unfavorable chilled water ÷T had a lesser effect on the average number of chillers online for the primary/secondary system with check valve and the variable flow, primary-only system. With an unfavorable ÷T, flow could exceed design value under some conditions and eliminate the need to add more chillers simply to increase plant flow rate. As a result, there was no significant difference between the average number of chillers online for the constant and favorable ÷T cases. Compared to the constant flow primary/variable flow secondary system the average number of chillers online for the unfavorable ÷T cases was significantly lower. In the four chiller configuration the average number of chillers increased from 1.9 to 2.3. The result was an increase of up to 16-percent in energy consumption for chiller auxiliaries (Table 3-13). Relative to constant ÷T, the favorable ÷T scenario resulted in no change in staging because chillers were capacity-limited rather than flow-limited.

The change in energy consumption due to ÷T effects as a percentage of the total plant energy consumption was relatively small. This was mainly due to the fact that chiller energy represents between 60 and 78-percent of the total annual plant energy use (Table 3-9) and ÷T did not significantly affect chiller energy.

Table 3-15 provides component and plant energy savings for the Houston office building system alternatives relative to the constant flow primary/variable flow secondary system. As mentioned previously, the constant flow, primary-only system's energy use was not impacted by changes to ÷T characteristics. Consequently, when compared to the constant flow primary/variable flow secondary system, the difference in energy use between the two system types decreased as the ÷T degraded from the favorable to the unfavorable ÷T case.

Table 3-15: Effect of ÷T on component and plant energy relative to constant flow primary/variable flow secondary system for the Houston office case

CHW system type		Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with a check valve	Variable flow, primary-only
Favorable ÷T	*Total plant energy, kWh/ton[1]#*	3,330	3,150 to 2,553	3,152 to 2,554	3,005 to 2,481
	÷ total plant energy, % [1]	6 to 30	Base	0	-5 to -3
	÷ chiller energy, % [1]	0 to 9	Base	0	0
	÷ CHW pump energy, % [1]	54 to 150	Base	0 to 1	-43 to -35
	÷ chiller aux. energy, % [1]	0 to 66	Base	0	0
Constant ÷T	*Total plant energy, kWh/ton[1]#*	3,330	3,157 to 2,587	3,160 to 2,568	3,013 to 2,517
	÷ total plant energy, % [1]	6 to 29	Base	0 to -1	-5 to -3
	÷ chiller energy, % [1]	0 to 9	Base	0	0
	÷ CHW pump energy, % [1]	50 to 131	Base	1 to -2	-42 to -24
	÷ chiller aux. energy, % [1]	0 to 60	Base	0 to -4	0 to -4
Unfavorable ÷T	*Total plant energy, kWh/ton[1]#*	3,330	3,194 to 2,836	3,196 to 2,725	3,062 to 2,666
	÷ total plant energy, % [1]	4 to 17	Base	0 to -4	-4 to -7
	÷ chiller energy, % [1]	0 to 7	Base	0 to -2	0 to -2
	÷ CHW pump energy, % [1]	36 to 65	Base	1 to -5	-35 to -24
	÷ chiller aux. energy, % [1]	0 to 26	Base	0 to -13	0 to -13

Note: [1]Ranges represent values for all four study chiller configurations

The addition of a check valve provided little to no energy savings over the base system when the ÷T was either favorable or constant. Savings upwards of 4-percent were realized in when the ÷T was unfavorable. A majority of the energy savings was due to reductions in chiller and chiller auxiliary energy.

In summary:

- Total plant energy use decreased as ÷T improved due to reduction in chilled water pump energy. Variable flow systems with unfavorable ÷T's consumed more chilled water pump energy and were likely to have greater chiller and chiller auxiliary energy use.
- Favorable ÷T reduced the total plant energy consumption by less than 1-percent. The savings realized were due to a decrease in variable speed pumping energy. Because chilled water pump energy is a relatively small portion of the plant, fractional energy savings were small.
- Overall, the variable flow, primary-only system was the least affected of all system types by the unfavorable ÷T and remained the lowest energy consumer as was shown in the Houston office case (Table 3-12).

3.4.3 Effect of Cooling Load Type

Table 3-16 shows the annual plant energy consumption in Houston for all system types. Houston was used for comparison, because all loads had 8760-hour cooling seasons in this location. The absolute differences in energy consumption between the study system types were greatest for the district plant case and least for the office building case, obviously, because of the differing sizes of the three load types. However, differences as a percentage of the total constant flow primary/variable flow secondary plant energy were not significantly affected by load type.

Table 3-16: Annual energy use comparisons, given in kWh, of study system types for Houston cases with constant ÷T versus load

CHW system type		Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with a check valve	Variable flow, primary-only
Office building	*Total plant energy, kWh[1]#*	1,665,116	1,578,723 to 1,293,616	1,579,862 to 1,283,799	1,506,606 to 1,258,620
	###- plant energy, kWh[1]	86,393 to 371,500	Base	1,139 to -9,817	-72,117 to -34,996
	###- plant energy, %[1]	5 to 29	Base	0 to -1	-5 to -3
Medical facility	*Total plant energy, kWh[1]#*	5,444,096	5,010,616 to 3,948,452	5,013,802 to 3,915,834	4,738,197 to 3,868,736
	###- plant energy, kWh[1]	433,480 to 1,495,643	Base	3,187 to -32,618	-227,419 to -79,717
	###- plant energy, %[1]	9 to 38	Base	0 to -1	-5 to -2
District plant	*Total plant energy, kWh[1]#*	18,446,810	15,391,166 to 14,349,189	15,179,786 to 14,167,258	14,777,038 to 14,038,615
	###- plant energy, kWh[1]	3,055,644 to 4,097,621	Base	-211,379 to -181,981	-614,128 to -310,574
	###- plant energy, %[1]	20 to 29	Base	-1	-4 to -2

Table 3-17 gives energy consumption per unit of design cooling load. This provides the common basis needed to compare systems of greatly differing sizes. Because of its higher load factor, the district plant consumed the most energy per ton, followed by the medical facility and office building. Energy savings due to variable chilled water flow were not appreciably different. Variable primary flow energy savings are greatest for a single chiller. The appearance of smaller maximum savings for the district plant is due only to the fact that a single chiller plant was not modeled.

Table 3-17: Annual energy comparisons, given in kWh per design ton, of study system types for Houston cases with constant ÷T versus load

CHW system type		Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with a check valve	Variable flow, primary-only
Office building	*Total plant energy, kWh/ton[1]#*	3,330	3,157 to 2,587	3,160 to 2,568	3,013 to 2,517
	###- plant energy, kWh/ton[1]	173 to 743	Base	2 to -20	-144 to -70
	Chiller energy, kWh/ton[1]	2,073	2,073 to 1,898	2,073 to 1,901	2,073 to 1,901
	###- Chiller energy, kWh/ton[1]	0 to 175	Base	0 to 3	0 to 3
	CHW pump energy, kWh/ton[1]	515	343 to 223	345 to 219	199 to 170
	###- CHW pump energy, kWh/ton[1]	172 to 292	Base	2 to -4	-144 to -53
	Chiller aux. energy, kWh/ton[1]	742	741 to 466	741 to 448	741 to 446
	###- chiller aux. energy, kWh/ton[1]	1 to 276	Base	0 to -19	0 to –20
Medical facility	*Total plant energy, kWh/ton[1]#*	3,629	3,340 to 2,632	3,343 to 2,611	3,189 to 2,579
	###- plant energy, kWh/ton[1]	289 to 997	Base	2 to -22	-152 to -53
	Chiller energy, kWh/ton[1]	2,011	2,011 to 1,844	2,011 to 1,847	2,011 to 1,847
	###- Chiller energy, kWh/ton[1]	0 to 167	Base	0 to 3	0 to 3
	CHW pump energy, kWh/ton[1]	638	349 to 220	351 to 217	197 to 187
	###- CHW pump energy, kWh/ton[1]	289 to 418	Base	2 to -3	-152 to -33
	Chiller aux. energy, kWh/ton[1]	980	980 to 568	980 to 546	980 to 545
	###- chiller aux. energy, kWh/ton[1]	0 to 412	Base	0 to -22	0 to -23
District plant	*Total plant energy, kWh/ton[1]#*	4,099	3,420 to 3,189	3,373 to 3,148	3,284 to 3,120
	###- plant energy, kWh/ton[1]	679 to 911	Base	-47 to -40	-136 to -69
	Chiller energy, kWh/ton[1]	2,377	2,268 to 2,243	2,267 to 2,250	2,267 to 2,250
	###- Chiller energy, kWh/ton[1]	108 to 134	Base	-1 to 7	-1 to 7
	CHW pump energy, kWh/ton[1]	685	334 to 287	322 to 276	233 to 248
	###- CHW pump energy, kWh/ton[1]	350 to 397	Base	-13 to -12	-101 to -40
	Chiller aux. energy, kWh/ton[1]	1,038	818 to 658	784 to 623	784 to 622
	###- chiller aux. energy, kWh/ton[1]	220 to 380	Base	-33 to -35	-34 to -36

Note: [1]Ranges represent kWh per design ton for all four study chiller configurations

In summary: When comparing differences in system energy use per unit of design load, load type had a relatively minor effect on variable primary flow savings. On a per design ton basis there was little difference between energy savings for the load types considered. Large differences in the ratio of primary pump head to total for various load types can affect savings. As shown in previous research (Bahnfleth and Peyer 2001), a reduction in the ratio of primary pump head to total plant head can further decrease variable primary flow chilled water pump energy savings.

3.4.4 Effect of Climate

Table 3-18 shows plant energy consumption per unit of design plant cooling load for the three study load types with a constant ÷T characteristic. Chilled water plants simulated with Houston load data consumed more energy than those using Syracuse load data because of the longer and hotter Houston cooling season. Table 3-18 also presents the increase in total plant energy consumption realized when comparing the Houston cases relative to those simulated using the Syracuse location. Plant energy use for the Houston office building was greater by between 96 and 114-percent relative to the same building in the Syracuse climate and depending on the system type and number of chillers in the plant. Plant energy use for the medical facility was 141 to 197 percent greater in Houston than in Syracuse. The increase in plant energy can be attributed to differences in cooling season length and ambient air temperatures. As mentioned previously, the Houston office building has a year-round cooling season and generally experienced higher outside air temperatures throughout the year. Conversely, the Syracuse season was approximately 40-percent shorter and had more occurrences where free cooling was used.

In the case of the district chilled water plant, both Syracuse and Houston systems had a yearlong cooling season. The result is that the difference in energy between the Syracuse and Houston district chilled water plants was smaller relative to that of the other study load types.

Table 3-18: Annual plant energy consumption per unit of design plant cooling load (kWh/ton) for the Syracuse and Houston cases.

	Constant flow primary-only		Constant flow primary/variable flow secondary		Variable flow primary-only	
	Syracuse	Houston	Syracuse	Houston	Syracuse	Houston
Office building annual energy use, kWh/design ton[1]	1,696	3,330	1,586 to 1,197	3,157 to 2,587	1,500 to 1,160	3,013 to 2,517
÷, kWh/design ton[1]	Base	1,634	Base	1,571 to 1,390	Base	1,513 to 1,357
÷, % [1]	Base	96	Base	99 to 116	Base	101 to 117
Medical facility annual energy use, kWh/design ton[1]	1,509	3,629	1,351 to 876	3,330 to 2,592	1,271 to 850	3,176 to 2,523
÷, kWh/design ton[1]	Base	2,120	Base	1,979 to 1,716	Base	1,905 to 1,673
÷, % [1]	Base	141	Base	147 to 196	Base	150 to 197
District plant annual energy use, kWh/design ton[1]	3,154	4,194	2,073 to 1,784	3,359 to 3,129	1,966 to 1,728	3,218 to 3,035
÷, kWh/design ton[1]	Base	1,040	Base	1,286 to 1,345	Base	1,252 to 1,307
÷, % [1]	Base	33	Base	62 to 75	Base	64 to 76

Note: [1]Ranges represent values for all four study chiller configurations

Table 3-19 compares annual plant and component energy use for Syracuse and Houston office buildings. Values were given as kWh/design ton and the constant flow primary/variable flow secondary system type is the basis of comparison. Not unexpectedly, variable primary flow energy savings were much larger in Houston due to its longer cooling season. Chiller auxiliary savings for the Houston case were at least twice that of the Syracuse case and chilled water pump energy savings were 1.5-times greater.

Table 3-19: Annual plant energy consumption per unit of design plant cooling load (kWh/ton) for Syracuse and Houston office buildings.

CHW system type		Constant flow primary-only	Constant flow primary/ variable flow secondary	Const. Flow Primary/ var. flow secondary with check valve	Variable flow, primary-only
Houston	*Total plant energy, kWh/ton[1]#*	3,330	3,157 to 2,587	3,160 to 2,568	3,013 to 2,517
	###÷ plant energy, kWh/ton[1]	173 to 743	Base	2 to -20	-144 to -70
	Chiller energy, kWh/ton[1]	2,073	2,073 to 1,898	2,073 to 1,901	2,073 to 1,901
	###÷ Chiller energy, kWh/ton[1]	0 to 175	Base	0 to 3	0 to 3
	CHW pump energy, kWh/ton[1]	515	343 to 223	345 to 219	199 to 170
	###÷ CHW pump energy, kWh/ton[1]	172 to 292	Base	2 to -4	-144 to -53
	Chiller aux. energy, kWh/ton[1]	742	741 to 466	741 to 448	741 to 446
	###÷ chiller aux. energy, kWh/ton[1]	1 to 276	Base	0 to -19	0 to -20
Syracuse	*Total plant energy, kWh/ton[1]#*	1,696	1,589 to 1,211	1,591 to 1,204	1,501 to 1,107
	###÷ plant energy, kWh/ton[1]	107 to 486	Base	1 to -7	-88 to -41
	Chiller energy, kWh/ton[1]	1,018	1,018 to 899	1,018 to 903	1,018 to 903
	###÷ Chiller energy, kWh/ton[1]	0 to 119	Base	0 to 4	0 to 4
	CHW pump energy, kWh/ton[1]	302	196 to 111	197 to 109	107 to 76
	###÷ CHW pump energy, kWh/ton[1]	106 to 191	Base	1 to -2	-88 to -36
	Chiller aux. energy, kWh/ton[1]	376	376 to 200	376 to 192	376 to 191
	###÷ chiller aux. energy, kWh/ton[1]	0 to 176	Base	0 to -9	0 to -9

Note: [1]Ranges represent values for all four study chiller configurations

In summary:

- ∉# Annual plant energy consumption in Houston was greater than in Syracuse, because of increased hours of plant operation and higher average cooling loads for equal peak loads.
- ∉# The variable flow, primary-only system was the lowest consumer in all cases.
- ∉# Variable flow, primary-only and primary/secondary check valve system savings relative to constant flow primary/variable flow secondary systems were greatly affected by climate. Savings in the warmer Houston climate were nearly twice the savings in Syracuse.

3.5 Economic Analysis

The economic performance of variable primary flow systems was compared with other system types on the basis of both life cycle cost and simple payback using representative capital cost estimates and electric energy rates.

3.5.1 Capital Cost

Capital cost models were based on manufacturer's quotations, standard estimating data (RS Means 2002) and estimates supplied by mechanical contractors. These sources were used to determine material, labor, and installation equipment costs for the major chilled water equipment. Base RS Means (2002) labor estimates were used in all cases. Labor estimates were not adjusted for regional labor cost differences. Overhead and profit was included at 18-percent of the total labor and material costs. Materials common to all system alternatives and chiller configurations, i.e., distribution and common plant piping, cooling coils, and control valves, were omitted from the cost estimate. Cost data used in model development is provided in Appendix F (Tables F-1 through F-9).

Capital Cost Models

Capital cost models were developed for chillers, cooling towers, pumps, and, when applicable, piping and fittings associated with the decoupler or low-flow bypass line. Polynomial models were used to capture the trend of installed equipment costs without including the discontinuous variations that occur due to changes in size.

Chiller

Chiller cost estimates were obtained for water-cooled centrifugal chillers ranging in size from 167 to 2,250-tons of cooling capacity (see Appendix F). Estimates include the major costs of purchasing and installing each chiller circuit. These include the cost of piping the chiller to the plant's supply and return headers. Table 3-20 gives sample estimate data for a 500-ton chiller.

Table 3-20: Sample installed centrifugal chiller cost estimate

Capacity, tons	*Design flow rate, gpm*	*Chiller cost, $*	*Labor cost, $*	*Installation equipment cost, $*	*Overhead & profit, $*	*Piping circuit cost, $*	*Total cost, $*
500	1,000	130,151	24,720	1,324	28,115	8,109	192,419

Chiller material cost was modeled using a function provided by a major chiller manufacturer (Figure 3-32). The unit cost, (US$ per nominal ton), decreases with increasing chiller size because labor costs for manufacturing a larger chiller are comparable to those for a smaller one while material cost per ton is relatively constant. The labor cost per ton, therefore, diminishes rapidly as size increases until only the unit material cost is significant. This is evident in the leveling off of cost per ton at nominal sizes above 1,000 tons in Figure 3-32.

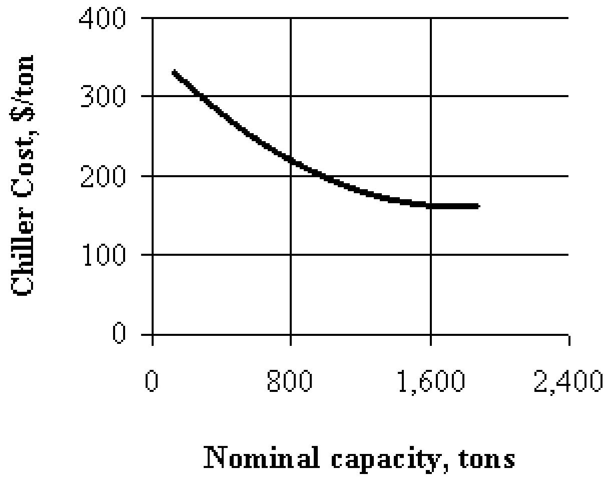


Figure 3-32: Chiller capital cost model

Labor and installation equipment (i.e., installation equipment necessary to deliver and set the equipment into place) costs for chiller installation were taken from standard estimating data (RS Means 2002). Continuous functions were fit to the data to model cost per ton (Figures 3-33 and 3-34). Unit costs tended to decrease with increasing chiller size toward a minimum at a capacity somewhat in excess of 1,000 tons. These models were compared to actual cost estimate prepared by a mechanical contractor (Vascellaro 2002) and found to be within 15-percent of the contractor's quotations.

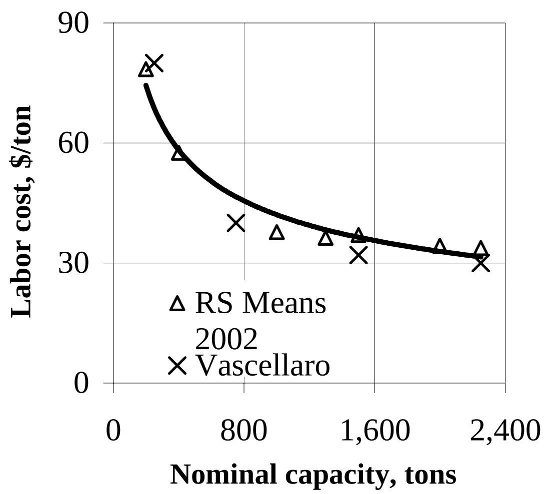


Figure 3-33: Labor cost model for chiller installation

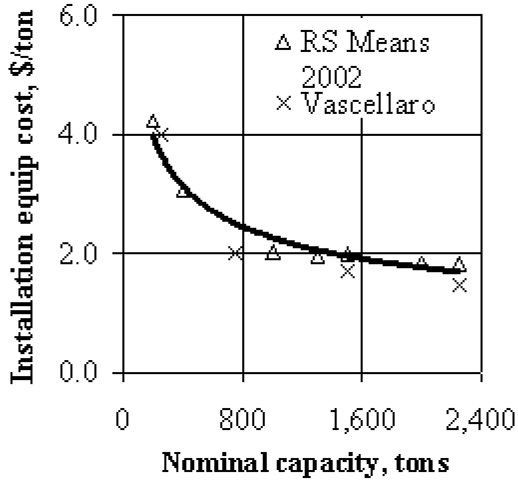


Figure 3-34: Installation equipment cost model for chiller installation

The chiller piping circuit cost includes material, labor, and overhead and profit costs for piping and fittings necessary to connect the chiller to supply and return headers. The model was developed using standard cost data (RS Means 2002). An itemized list of all materials appears in Appendix F (Table F-2). The first order polynomial model is shown in Figure 3-35.

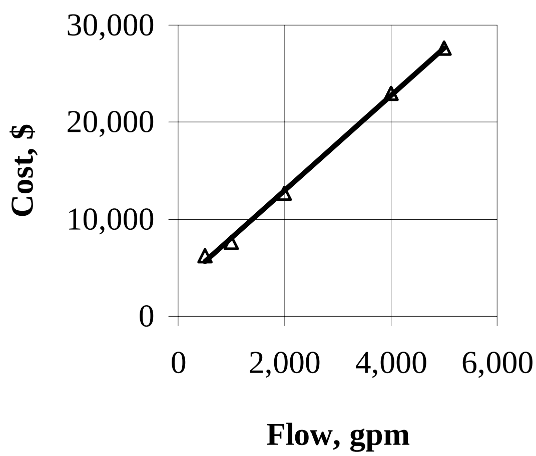


Figure 3-35: Piping, fittings, and accessories cost model for cooling tower and chiller circuits

Cooling tower

Cooling tower capital costs were estimated for induced draft towers ranging in size from 167 to 2,250 tons. Estimates include material, labor, overhead and profit, and piping costs for purchasing and installing the cooling tower. Table 3-21 illustrates an estimate for a 500-ton cooling tower.

Table 3-21: Sample cooling tower cost estimate

Design flow rate, gpm	*Capacity, tons*	*Tower cost, $*	*Labor cost,*	*Overhead & profit cost, $*	*Piping circuit cost, $*	*Total cost, $*
1,500	500	26,452	2,600	5,229	10,554	44,835

Cooling tower material and labor costs were modeled by curve fitting a combination of standard estimating data (RS Means 2002) and estimates provided by a manufacturer representative (Bullock 2002). These models are shown in Figures 3-36 and 3-37. The cost per cooling ton was greater for the Houston case because it takes a larger cooling tower to provide the same performance at selection conditions in Houston than in Syracuse. The piping circuit cost model discussed previously (Figure 3-35) was also used to approximate the piping costs for the cooling tower cost estimates.

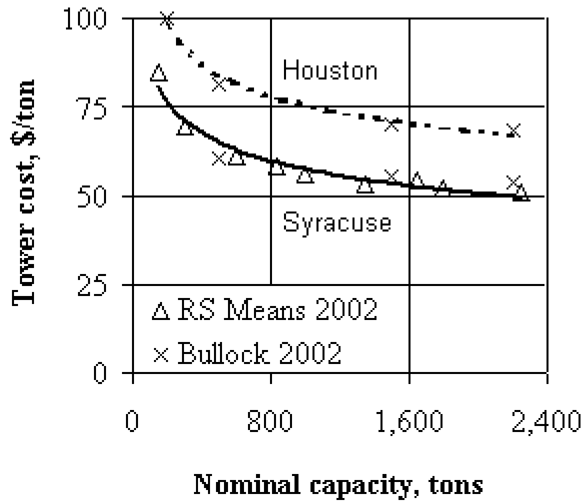


Figure 3-36: Cooling tower cost models for the Houston and Syracuse climates

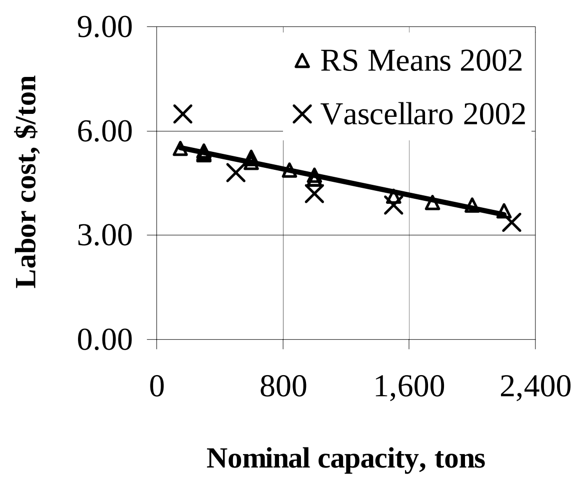


Figure 3-37: Labor cost model for cooling tower installation

Pump

Pump capital cost models were developed for centrifugal end-suction and double-suction pumps. Inspection of material cost estimates revealed that pump costs could be closely approximated using flow rate and motor size. End-suction and double-suction pumps were modeled to handle a range of operating conditions. The end-suction pump model covers selections ranged from 333 to 1,125 gpm and from 50 and 120 feet of head. The double-suction pump model covers selections ranged from 1,500 to 6,750 gpm and from 50 and 150 feet of head. Table 3-22 illustrates estimates for several pumps selected for a design flow rate of 1,000 gpm and between 50 and 120 feet of head. Estimates include material, labor, overhead and profit, variable frequency drive or starter, and electrical service costs.

Table 3-22: Sample pump cost estimate, ref. Table F-12(a)

Design capacity, gpm	*Design head, feet*	*Motor size, hp*	*Pump material cost, $*	*Labor cost, $*	*O&P cost, $*	*Piping cost, $*	*VFD/ starter cost, $*	*Electrical service cost, $*	*Total cost, $*
1,000	120	50	5,803	574	1,148	13,790	10,317[1]	2,233	33,865
1,000	120	50	5,803	574	1,148	13,790	2,299[2]	2,233	25,847
1,000	70	25	4,446	465	884	13,790	6,328[1]	1,605	27,518
1,000	50	20	4,174	434	830	13,790	1,026[2]	1,480	21,733

Note: [1]Cost of VFD for variable speed pump [2]Cost of starter for constant speed pump

Pump material cost models were developed by regression of equipment cost quotations supplied by a manufacturer representative (Anzelone 2002). Pump cost is a function of both flow rate and motor power, which indirectly represents the effect of head. Figure 3-38 is a plot of the cost model for end-suction pumps, and Figure 3-39 shows the model for double-suction pumps.

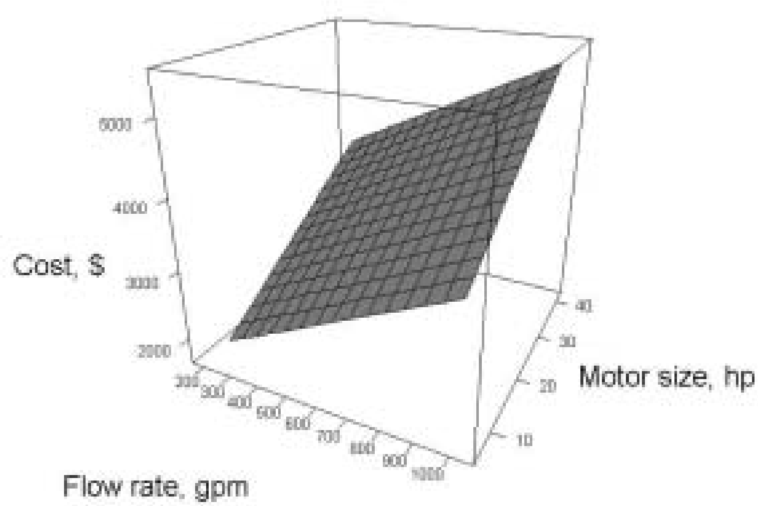


Figure 3-38: Cost model for end suction, flexible-coupled centrifugal pumps

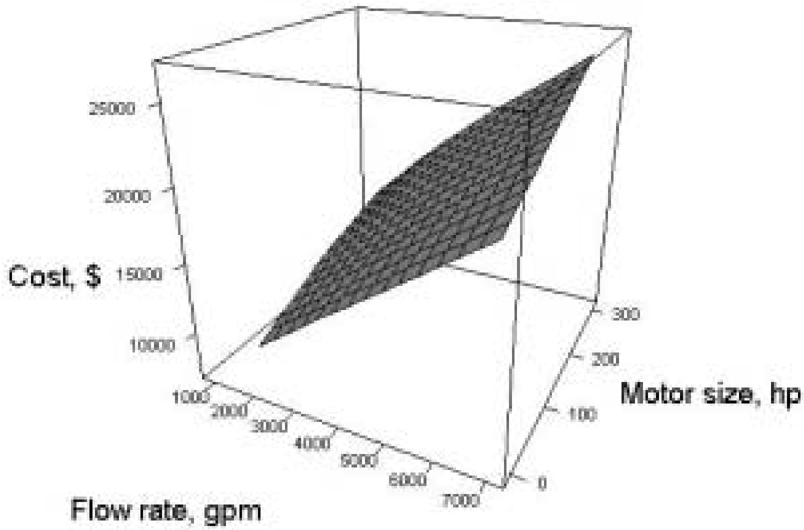


Figure 3-39: Cost model for double suction, flexible-coupled centrifugal pumps

Labor costs for pump installation were taken from standard estimating data (RS Means 2002). The model fit to the data (Figure 3-40) closely tracks estimates provided for validation by a mechanical contractor (Vascellaro 2002).

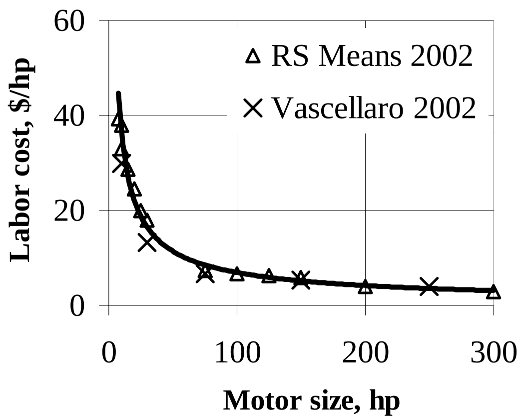


Figure 3-40: Labor cost model for pump installation

The pump piping circuit cost includes material, labor, and overhead and profit costs for the piping and fittings necessary to connect the pump to the supply and return headers. An itemized list is provided in Appendix F. Cost figures were taken from Means cost data (RS Means 2002). A first order polynomial was fit to the data (Figure 3-41).

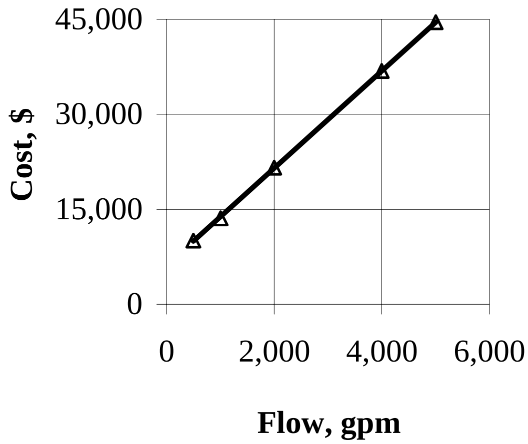


Figure 3-41: Piping, fittings, and accessories cost model for pump circuits

Motor starters and variable frequency drive costs were modeled using 480-volt cost data from Means (2002). Second and first order polynomials were fit to the variable frequency drive and starter data, respectively (Figure 3-42).

Electrical service costs were taken from Means (2002). An itemized list of materials is provided in Appendix F. Figure 3-43 illustrates the model used to represent the pump electrical service costs.

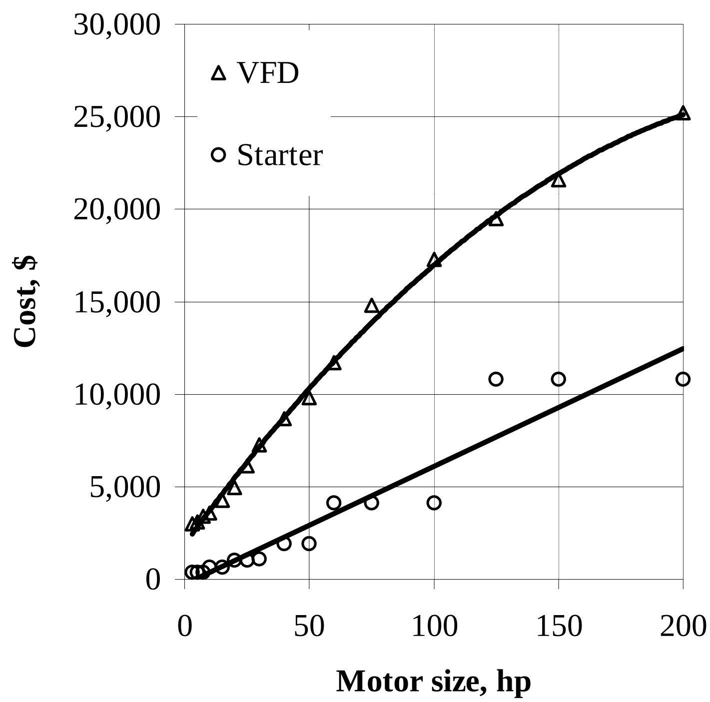


Figure 3-42: Variable frequency drive and starter installed cost models

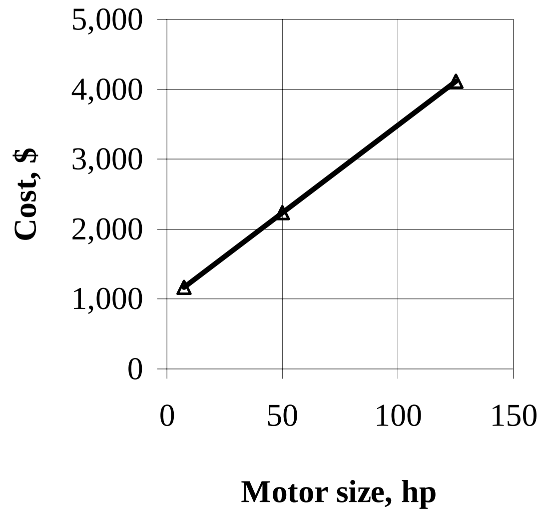


Figure 3-43: Electrical service cost model for pumps

Decoupler and bypass line

Decoupler and bypass line costs include pipe, fittings, and, if necessary, check valve or control valve costs. The line size was determined by the number of chillers, design flow rate, and system type. For example, a two chiller primary/secondary system would require a decoupler line sized to accommodate the design flow rate of the larger of the two chillers. Conversely, a variable primary flow system would require a smaller bypass line than the primary/secondary system because the bypass would be sized for the minimum flow rate of the larger chiller.

A sample decoupler/bypass cost estimate is shown in Table 3-23. Pipe, tee, reducer, check valve, and control valve costs were estimated using Means (2002). Check valve costs is included only in decoupler cost estimates for the primary/secondary check valve system and control valve cost is included in bypass line estimates for variable primary flow system. Decoupler or bypass pipe and fitting cost summaries for all study systems are provided in Appendix F.

Table 3-23:   Sample decoupler/bypass piping and fittings estimate, ref. Table F-14(a)

Design capacity, gpm	*Nominal pipe size, inches*	*Pipe cost, $*	*Tee cost, $*	*Reducer cost, $*	*Decoupler subtotal cost, $*	*Check valve cost, $*	*Control valve cost, $*
1,000	8	587	1,097	NA	1,685	3,869	2,995

Capital Cost Comparisons

Capital cost comparisons considered system components that varied with system type and number of chillers. Chilled water plant costs common to all cases examined were neglected in order to simplify the analysis. Table 3-24 summarizes percentage of total equipment and installation cost of each system component. The chiller is by far the most costly component, accounting for between 54 and 71-percent of the total. The chiller, cooling tower, and condenser water pump represent between 77 and 91-percent of the total plant cost.

Table 3-24: Percentage of total capital cost for all study cases

Load type	Office building	Medical facility	District plant
Chiller, % of total plant cost [1]	54 to 66	61 to 71	56 to 63
Cooling tower, % of total plant cost [1]	14 to 16	7 to 8	14 to 16
Condenser water pump, % of total plant cost [1]	9 to 11	9 to 12	10 to 12
Decoupler pipe and accessories, % of total plant cost [1]	0 to 1	0 to 3	1 to 2
Chilled water pump, % of total plant cost [1]	9 to 21	10 to 19	10 to 19

Note: [1]Ranges represent values for all four study chiller configurations and system types

Table 3-25 shows the total plant capital costs for all study cases. Ranges represent the cost per design ton for all chiller configurations. The office building, because it is relatively small has the highest cost per ton. The larger medical center and district system plants have smaller and roughly comparable costs.

Table 3-25: Total plant capital costs ($/design ton) for the study cases

CHW system type	Constant flow primary-only	Constant flow primary/variable flow secondary	Primary/ secondary with check valve	Variable flow, primary-only
$/ton, office building	585 to 917	634 to 1,030	642 to 1,032	609 to 953
%÷, office building	-8 to -11	Base	1 to 0	-4 to -8
$/ton, medical facility	372 to 595	413 to 651	421 to 653	391 to 617
%÷, medical facility	-10 to -9	Base	2 to 0	-5
$/ton, district plant	375 to 487	413 to 535	417 to 537	387 to 503
%÷, district plant	-9	Base	1 to 0	-6

Constant flow, primary-only system types have 8 to 11-percent lower capital costs than comparable primary/secondary systems. Variable flow, primary-only systems are 4 to 8-percent lower in capital cost. The addition of the check valve increases the cost of the primary/secondary system by no more than 2-percent.

Table 3-26 shows the added cost of increasing the number of chillers relative to the single chiller configuration. The district chilled water plant cases were not included in the comparison because there it had no single chiller configuration. The cost of increasing the number of constant flow, primary-only chillers from one to two was considerably greater than the cost of converting a single chiller constant flow, primary-only plant to any of the variable flow alternatives.

Table 3-26:   Added cost (given in percentage of total) of increasing the number of chillers

CHW system type	Constant flow, primary-only		Constant flow primary/variable flow secondary		Primary/ secondary with check valve		Variable flow, primary-only	
Load type	Office building	Medical facility	Office building	Medical facility	Office building	Medical facility	Office building	Medical facility
1 chiller config.	Base	Base	Base	Base	Base	Base	Base	Base
2 chiller config.	23	28	24	27	24	26	23	27
3 chiller config.	41	47	45	45	43	43	41	44
4 chiller config.	57	60	63	58	61	55	56	58

Figure 3-44 shows the capital cost difference and incremental cost for the constant flow primary/variable flow secondary and the variable flow, primary-only system types of the office building plant equipment.   The incremental cost of adding additional units of capacity beyond two units decreases for both system types.

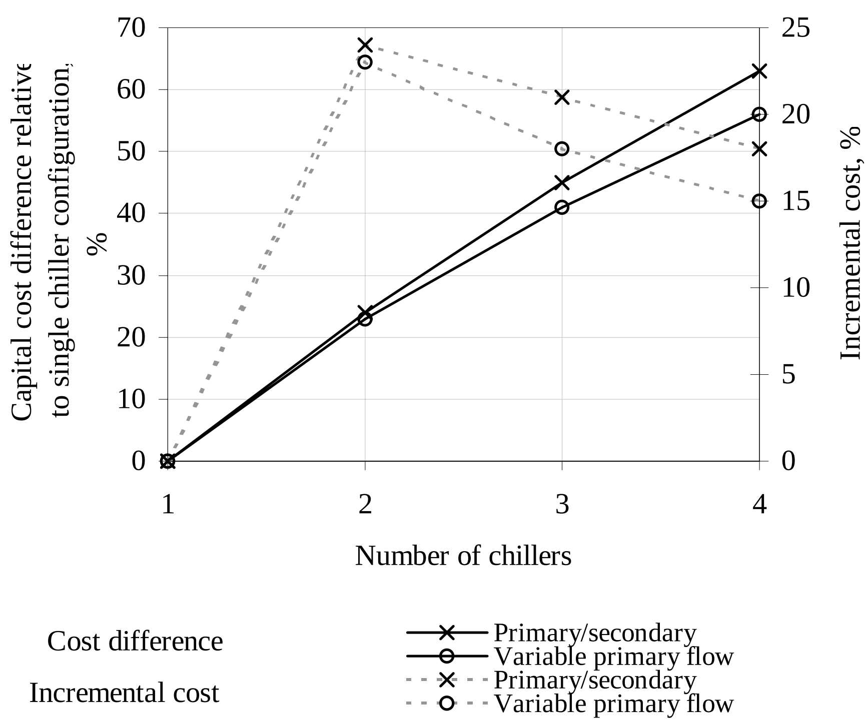


Figure 3-44: Capital cost difference and incremental cost relative to a single chiller configuration

3.5.2 Annual Energy Cost

Table 3-27 provides an annual energy cost summary for all constant ÷T cases. Operating costs were calculated using a simple electric rate with an energy charge of $0.035 per kWh and a demand charge of $12 per monthly peak kW Annual energy charges vary with system type, load type, number of chillers, and other study parameters, while distribution, or demand, charges were not greatly affected.
Tables 3-28 and 3-29 summarize the contribution of components to annual energy and demand charges, respectively. The chiller energy charges represent between 58 and 74-percent of the total, while chiller auxiliary energy charges represent between 14 and 35-percent and chilled water pump energy charges represent between 7 and 15-percent. These values roughly correspond to the percentages of total plant energy use for each component presented in Table 3-9.

Table 3-27: Annual energy cost comparisons of Houston study system types with a constant ÷T.

CHW system type		Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with a check valve	Variable flow, primary-only
Office building	*Total plant energy cost, $/ton[1]#*	205	198 to 178	198 to 178	192 to 175
	Energy charge, $/ton[1]	117	111 to 91	111 to 90	105 to 88
	Energy charge, % of total cost[1]	57	56 to 51	56 to 51	55 to 50
	Demand charge, $/ton[1]	89	88	88	87
	Demand charge, % of total cost[1]	43	44 to 49	44 to 49	45 to 50
Medical facility	*Total plant energy cost, $/ton[1]#*	199	186 to 160	186 to 159	180 to 157
	Energy charge, $/ton[1]	114	105 to 83	105 to 82	100 to 81
	Energy charge, % of total cost[1]	57	56 to 52	56 to 52	56 to 52
	Demand charge, $/ton[1]	85	81 to 77	81 to 77	79 to 76
	Demand charge, % of total cost[1]	43	44 to 48	44 to 48	44 to 48
District plant	*Total plant energy cost, $/ton[1]#*	227	203 to 193	201 to 191	196 to 189
	Energy charge, $/ton[1]	136	113 to 105	112 to 104	109 to 103
	Energy charge, % of total cost[1]	60	56 to 54	56 to 54	56 to 54
	Demand charge, $/ton[1]	91	90 to 88	90 to 87	88 to 86
	Demand charge, % of total cost[1]	40	44 to 46	44 to 46	44 to 46

Note: [1]Ranges represent annual energy cost per design ton for all four study chiller configurations

The chiller represents between 75 and 77-percent of the total demand charges, while chiller auxiliaries represent between 15 and 16-percent and chilled water pumps between 8 and 10-percent of the total demand charges.

Table 3-28: Contribution of various system components to total plant annual energy charge for the Houston office building case with a constant ÷T.

CHW system type	Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with check valve	Variable flow, primary-only
Chiller energy charge, %[1]	62	58 to 65	65 to 74	63 to 74
CHW pump energy charge, %[1]	15	7 to 11	9 to 11	6 to 7
Chiller aux. energy charge, %[1]	23	14 to 24	17 to 24	17 to 25

Note: [1]Ranges represent values for all four study chiller configurations

Table 3-29: Contribution of various system components to total plant annual demand charge for the Houston office building case with a constant ÷T.

CHW system type	Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with check valve	Variable flow, primary-only
Chiller demand charge, %[1]	75	75	75 to 76	76 to 77
CHW pump demand charge, %[1]	10	9	9	8
Chiller aux. demand charge, %[1]	15	15 to 16	15	15 to 16

Note: [1]Ranges represent values for all four study chiller configurations

Effect of Number of Chillers

Table 3-30 compares annual energy costs for the Houston office building. Annual energy costs decreased with increasing number of chillers, mainly as a result of reduced auxiliary energy consumption. Varying the number of chillers had greater impact on energy (use) charges than it did on demand charges. For example, in the case of the variable flow, primary-only system, the annual energy charges was $105/ton for a on- chiller plant but only $88/ton for the four chiller configuration, while the demand charge was $87/ton in both cases. Consequently, the operating savings associated with variable primary flow were smaller than might have been expected, since only one component of energy cost was affected.

The variable flow, primary-only system had the lowest annual energy cost in all chiller configurations. The single chiller variable flow, primary-only system saved $6/ton in annual energy cost over the equivalent constant flow primary/variable flow secondary system. These savings diminished slightly with increasing number of chillers. The four chiller variable flow, primary-only system provided a smaller savings of $4/ton.

Table 3-31 shows component energy charges for the Houston office building cases. Most of the differences in energy charges found between the variable flow study system types are generated by the chilled water pumps and, in the multiple chiller cases, chiller auxiliaries. Despite the fact that the variable flow, primary-only chilled water pumps represent only 6 to 7-percent of the total plant energy charges (Table 3-28), they account for most savings relative to other system alternatives. As was the case for pumping energy, the pump energy charges saved by the variable flow, primary-only system, relative to the primary/secondary system alternatives diminished with increasing units of capacity. For example, in the single chiller configuration, the chilled water pumps of the variable flow, primary-only system saved $5 per design ton relative to the primary/secondary systems. This savings amounts to only $2 per ton in the four chiller configuration.

In summary:

- Annual energy costs decreased with increasing number of chillers.
- As was the case for energy savings, there is a diminishing return of annual energy cost savings as additional chillers are added. The two chiller configuration produced the greatest incremental cost savings.
- Variable primary flow energy cost savings relative to constant flow primary/ variable flow secondary systems diminished with increased number of chillers, although both benefited from having more than one chiller.

Table 3-30: Comparison of annual energy cost figures for the study chiller configurations of the Houston office building with constant ΔT.

CHW system type		Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with a check valve	Variable flow, primary-only
1 chiller	*Annual plant energy cost, $/ton*	205	198	198	192
	Δ plant energy cost, $/ton	7	Base	0	-6
	Annual energy charge, $/ton	117	111	111	105
	Annual demand charge, $/ton	89	88	88	87
2 chillers	*Annual plant energy cost, $/ton*	205	183	183	178
	Δ plant energy cost, $/ton	22	Base	-1	-5
	Annual energy charge, $/ton	117	96	95	92
	Annual demand charge, $/ton	89	88	88	87
3 chillers	*Annual plant energy cost, $/ton*	205	181	179	176
	Δ plant energy cost, $/ton	25	Base	-1	-5
	Annual energy charge, $/ton	117	92	91	89
	Annual demand charge, $/ton	89	88	88	87
4 chillers	*Annual plant energy cost, $/ton*	205	178	178	175
	Δ plant energy cost, $/ton	27	Base	-1	-4
	Annual energy charge, $/ton	117	91	90	88
	Annual demand charge, $/ton	89	88	88	87

Table 3-31: Breakdown of component energy charges for the study chiller configurations of the Houston office building with constant ÷T.

CHW system type		Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with a check valve	Variable flow, primary-only
1 chiller	*Chiller energy charges, $/ton*	73	73	73	73
	CHW pump energy charges, $/ton	18	12	12	7
	÷ CHW pump energy charges, $/ton	6	Base	0	-5
	Chiller aux. energy charges, $/ton	26	26	26	26
	Annual plant energy cost, $/ton	117	111	111	105
2 chillers	*Chiller energy charges, $/ton*	73	68	68	68
	CHW pump energy charges, $/ton	18	9	9	6
	÷ CHW pump energy charges, $/ton	9	Base	0	-3
	Chiller aux. energy charges, $/ton	26	19	18	18
	÷ chiller aux. energy charges, $/ton	7	Base	-1	-1
	Annual plant energy cost, $/ton	117	96	95	92
3 chillers	*Chiller energy charges, $/ton*	73	67	67	67
	CHW pump energy charges, $/ton	18	8	8	6
	÷ CHW pump energy charges, $/ton	10	Base	0	-2
	Chiller aux. energy charges, $/ton	26	17	16	16
	÷ chiller aux. energy charges, $/ton	9	Base	-1	-1
	Annual plant energy cost, $/ton	117	92	91	89
4 chillers	*Chiller energy charges, $/ton*	73	66	67	67
	CHW pump energy charges, $/ton	18	8	8	6
	÷ CHW pump energy charges, $/ton	10	Base	0	-2
	Chiller aux. energy charges, $/ton	26	17	15	15
	÷ chiller aux. energy charges, $/ton	9	Base	-2	-2
	Annual plant energy cost, $/ton	117	91	90	88

Effect of Chilled Water ÷T

Table 3-32 compares annual energy costs as a function of chilled water ÷T for the Houston office building. As would be expected, annual energy costs decreased with improving chilled water ÷T because of decreased pumping volumes. Unlike a change in the number of chillers, a change in chilled water ÷Π affects both energy and demand charges. For example, annual energy charges of the variable flow, primary-only system with favorable ÷T were between $105 and $87/ton, while the same system with an unfavorable ÷T had energy charges of $107 and $93/ton. Demand charges increased from $86 with favorable ÷Π to $88/ton with an unfavorable ÷Π. Still, the effect on energy ($6/ton) was three times larger than the effect on demand ($2/ton).

The variable flow, primary-only system had the lowest annual energy cost in all scenarios. The variable flow, primary-only system with favorable ÷T saved $3 to $6/ton in annual energy cost over the constant flow primary/variable flow secondary system. With the unfavorable ÷T model, variable primary flow savings were between $5 and $6/ton.

The check valve system generated non-negligible energy cost savings over the constant flow primary/variable flow secondary system only when ÷T's were less than design. In the unfavorable ÷T scenario, the check valve system saved more than $4/ton over the constant flow primary/variable flow secondary system type.

Table 3-33 shows the impact of the favorable and unfavorable ÷T characteristics on demand charge relative to equivalent constant ÷T cases. The unfavorable ÷T scenario provided an increase in total plant demand charge of between 1 and 2-percent. Chilled water pumps experienced an increase of between 19 and 23-percent and the chiller auxiliaries between 0 and 8-percent. Neither the favorable or unfavorable ÷T scenario impacted chiller demand charges. However, if chilled water ÷T could not meet the design value at full load, chiller demand charges would be adversely affected. This scenario was not included in the study.

Table 3-32: Comparison of annual energy cost figures for the study chilled water ÷T's of the Houston office building.

CHW system type		Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with a check valve	Variable flow, primary-only
Favorable ÷T	*Annual plant energy cost, $/ton*[1]	205	198 to 177	198 to 177	192 to 174
	#÷ plant energy cost, $/ton[1]	7 to 28	Base	0	-6 to -3
	Annual energy charge, $/ton[1]	117	110 to 89	110 to 89	105 to 87
	Annual demand charge, $/ton[1]	89	88 to 87	88 to 87	86
Constant ÷T	*Annual plant energy cost, $/ton*[1]	205	198 to 178	198 to 178	192 to 175
	#÷ plant energy cost, $/ton[1]	7 to 27	Base	0	-6 to -3
	Annual energy charge, $/ton[1]	117	111 to 91	111 to 90	105 to 88
	Annual demand charge, $/ton[1]	89	88	88	87
Unfavorable ÷T	*Annual plant energy cost, $/ton*[1]	205	200 to 188	200 to 184	195 to 182
	#÷ plant energy cost, $/ton[1]	5 to 17	Base	0 to -4	-5 to -6
	Annual energy charge, $/ton[1]	117	112 to 99	112 to 95	107 to 93
	Annual demand charge, $/ton[1]	89	88	88	88

Note: [1]Ranges represent values for all four study chiller configurations

Table 3-33: Impact of favorable and unfavorable ÷T models on demand charge relative to constant ÷T.

CHW system type	Constant flow primary/ variable flow secondary		Primary/ secondary with a check valve		Variable flow primary-only	
÷T model	Favorable	Unfavorable	Favorable	Unfavorable	Favorable	Unfavorable
÷ chiller demand charge, %[1]	0	0	0	0	0	0
÷ CHW pump demand charge, %[1]	-5 to -8	19 to 23	-5	19 to 23	-6 to -13	23 to 25
÷ chiller aux. demand charge, %[1]	0 to -6	0 to 8	0 to -1	0 to 6	0	0 to 6
÷ total plant demand charge, %[1]	0 to -1	1 to 2	0	1	0 to -1	2

Note: [1]Ranges represent values for all four study chiller configurations

Table 3-34 shows the impact of the favorable and unfavorable ÷T characteristics on energy charge relative to equivalent constant ÷T cases. As was the case for energy use, relative to the base cases, energy charges decreased by not more than 1-percent in the favorable ÷T cases and increased by between 1 and 10-percent in the unfavorable ÷T cases. The unfavorable ÷T increased the chilled water pump energy charges by 10 to 40-percent and chiller auxiliary energy charges by 0 to 27-percent.

Table 3-34: Impact of favorable and unfavorable ÷T models on energy charge relative to constant ÷T

CHW system type	Constant flow primary/ variable flow secondary		Primary/ secondary with a check valve		Variable flow primary-only	
÷T model	Favorable	Unfavorable	Favorable	Unfavorable	Favorable	Unfavorable
÷ chiller energy charge, %[1]	0	0 to 2	0	0 to 1	0	0 to 1
÷ CHW pump energy charge, %[1]	-2 to -8	10 to 40	-2 to -5	11 to 36	-1 to -13	24 to 40
÷ chiller aux. energy charge, %[1]	0 to -4	0 to 27	0	0 to 16	0	0 to 16
÷ total plant energy charge, %[1]	0 to -1	1 to 10	0 to -1	1 to 7	0 to -1	2 to 6

Note: [1]Ranges represent values for all four study chiller configurations

In summary:

- Total plant annual energy cost decreased as ÷T improved due to reduction in chilled water pump energy charges. Systems experiencing less-than-design ÷T's had higher chilled water pump, chiller, and chiller auxiliary energy charges.
- Overall, the variable flow, primary-only system annual energy costs were the least affected of all system types by the unfavorable ÷T and had the lowest annual energy cost.

Effect of Cooling Load Type

Table 3-35 provides a comparison of annual energy cost for the Houston load types with a constant ÷T. As was the case for energy consumption, annual energy costs were greatest for the district plant and lowest for the office building load type.

Table 3-35: Comparison of annual energy costs for the Houston load types with a constant ÷T.

CHW system type		Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with a check valve	Variable flow, primary-only
Office building	*Plant energy cost, $[1]#*	102,570	99,180 to 89,214	99,238 to 88,778	96,089 to 87,402
	###- plant energy cost, $[1]	3,390 to 13,356	Base	58 to -436	-3,091 to -1,812
	###- plant energy cost, %[1]	3 to 15	Base	0	-3 to -2
Medical facility	*Plant energy cost, $[1]#*	298,449	279,463 to 240,419	279,617 to 238,386	269,274 to 235,713
	###- plant energy cost, $[1]	18,986 to 58,030	Base	154 to -2,033	-10,189 to -4,706
	###- plant energy cost, %[1]	7 to 24	Base	0 to -1	-4 to -2
District plant	*Plant energy cost, $[1]#*	1,021,194	912,863 to 869,108	904,708 to 860,801	884,250 to 851,730
	###- plant energy cost, $[1]	108,331 to 152,086	Base	-8,155 to -8,307	-28,613 to -17,378
	###- plant energy cost, %[1]	12 to 17	Base	-1	-3 to -2

Note: [1]Ranges represent values for all four study chiller configurations

Table 3-36 gives comparisons of annual energy cost figures per design ton for the Houston load types with a constant ÷T. The load type impacted both the energy and demand charges. For example, the annual energy charges for the office building variable flow, primary-only system cases were between $105 and $87/ton. The energy charges for the medical facility and district plant variable primary flow systems were between $100 and $81/ton for the medical facility and between $109 and $103/ton for the district plant. Demand charges were $87/ton for the office building, between $79 to $76/ton for the medical facility, and between to $88 to $86/ton for the district plant.

The variable flow, primary-only system had the lowest annual energy cost of the system alternatives for all load types. The variable flow, primary-only system simulated using the office building data saved between $3 and $6/ton in annual energy cost over the equivalent constant flow primary/variable flow secondary system. Variable primary flow savings were the same for the medical facility. In the district plant case, the variable primary flow savings were between $4 and $7/ton.

In summary:

- Load type had a moderately significant impact on the total annual energy cost of each plant. The district plant load had the highest cost per ton, roughly 10% greater than the medical facility, which had the least.
- Variable primary flow annual energy cost savings, given in $/year, generated over primary/secondary systems are greatly affected by load type. The savings in the district plant case were approximately 10-times larger than that of the office building case.
- However, when comparing differences in system energy costs on a $/ton basis, the savings for all three load types were approximately the same.

Table 3-36: Comparison of annual energy cost figures for the Houston load types with a constant ÷T.

CHW system type		Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with a check valve	Variable flow, primary-only
Office building	*Total plant energy cost, $/ton[1]#*	205	198 to 178	198 to 178	192 to 175
	÷ energy cost, $/ton[1]	7 to 27	Base	0	-6 to -3
	Energy charge, $/ton[1]	117	111 to 91	111 to 90	105 to 87
	÷ energy charge, $/ton[1]	6 to 25	Base	0 to -1	-6 to -3
	Demand charge, $/ton[1]	89	88	88	87
	÷ demand charge, $/ton[1]	1	Base	0	-1
Medical facility	*Total plant energy cost, $/ton[1]#*	199	186 to 160	186 to 159	180 to 157
	÷ energy cost, $/ton[1]	13 to 39	Base	0 to -1	-6 to -3
	Energy charge, $/ton[1]	114	105 to 83	105 to 82	100 to 81
	÷ energy charge, $/ton[1]	9 to 31	Base	0 to -1	-5 to -2
	Demand charge, $/ton[1]	85	81 to 77	81 to 77	79 to 76
	÷ demand charge, $/ton[1]	4 to 8	Base	0	-2 to -1
District plant	*Total plant energy cost, $/ton[1]#*	227	203 to 193	201 to 191	196 to 189
	÷ energy cost, $/ton[1]	24 to 34	Base	-2	-7 to -4
	Energy charge, $/ton[1]	136	113 to 105	112 to 104	109 to 103
	÷ energy charge, $/ton[1]	23 to 31	Base	-1	-4 to -2
	Demand charge, $/ton[1]	91	90 to 88	90 to 87	88 to 86
	÷ demand charge, $/ton[1]	1 to 3	Base	0 to -1	-2

Note: [1]Ranges represent annual energy cost per design ton for all four study chiller configurations

Effect of Climate

Table 3-37 gives annual energy costs for the office building in Houston and Syracuse. Energy costs in Houston were nearly twice as large as in Syracuse.

Variable primary flow energy cost savings relative to primary/secondary systems were approximately 1.5-times greater in Houston than in Syracuse. Most of the cost savings are contributed to energy charges.

Table 3-37: Comparison of annual energy cost figures for the study climates of the office building with a constant ÷T.

CHW system type		Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with a check valve	Variable flow, primary-only
Houston	*Total plant energy cost, $/ton[1]#*	205	198 to 178	198 to 178	192 to 175
	÷ energy cost, $/ton[1]	7 to 27	Base	0	-6 to -3
	Energy charge, $/ton[1]	117	111 to 91	111 to 90	105 to 87
	÷ energy charge, $/ton[1]	6 to 25	Base	0 to -1	-6 to -3
	Demand charge, $/ton[1]	89	88	88	87
	÷ demand charge, $/ton[1]	1	Base	0	-1
Syracuse	*Total plant energy cost, $/ton[1]#*	111	107 to 93	107 to 92	103 to 91
	÷ energy cost, $/ton[1]	4 to 18	Base	0 to -1	-4 to -2
	Energy charge, $/ton[1]	59	56 to 42	56 to 42	53 to 41
	÷ energy charge, $/ton[1]	3 to 17	Base	0	-3 to -1
	Demand charge, $/ton[1]	51	51 to 50	51 to 50	50
	÷ demand charge, $/ton[1]	0 to 1	Base	0	-1 to 0

Note: [1]Ranges represent annual energy cost per design ton for all four study chiller configurations

In summary, variable flow, primary-only and primary/secondary check valve system annual energy cost savings generated over constant flow primary/variable flow secondary systems were greatly affected by climate with larger savings occurring in climates with larger cooling loads.

3.5.3 Life Cycle Cost and Payback Period

Economic comparisons of alternative systems were made on a simple payback and life cycle cost basis. Life cycle costs were calculated for a 20-year economic life. Costs were calculated on a constant-dollar basis using a 3.1% real discount rate and commercial electric price escalation indices taken from a standard US government source (Fuller 2002). Capital costs were assumed to occur at the time of construction. Energy consumption was assumed to be identical for each year of the analysis.

In each case the constant flow, primary-only system type with corresponding chiller configuration was used as the reference for determining the simple payback period of other system alternatives. Life cycle cost and simple payback period optima were based solely on energy and capital cost considerations. Redundancy and firm capacity concepts could change the optima in some scenarios. However, they were not considered in this study.

Effect of Number of Chillers

Table 3-38 provides the life cycle cost per design ton as a function of number of chillers for the Houston office building. Because annual energy costs decrease and capital costs increase with increasing number of chillers, the chiller configuration producing the lowest life-cycle cost varies. The same is true for simple payback period.

The variable flow, primary-only system was the lowest life cycle cost alternative in all chiller configurations. Variable flow, primary-only life-cycle cost savings relative to the constant flow primary/variable flow secondary system increased from $109/ton for the single chiller configuration to $128/ton for the four chiller configuration. The check valve system had savings relative to the constant flow primary/variable flow secondary system of between $6 and $17/ton depending on the chiller configuration.

Figure 3-45 shows the simple payback period as a function of the number of chillers in the plant for the Houston office building. Payback periods for this and other office building system types are shortest when the two chiller configuration is considered. Payback periods for all of the cases with multiple chiller configurations were at or below 4.2 years. The variable flow, primary-only system payback period was not more than two years in any chiller configuration.

Table 3-38: Comparison of life-cycle costs and payback periods for the study chiller configurations of the Houston office building with a constant ÷T.

CHW system type		Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with check valve	Variable flow, primary-only
1 chiller	*Life cycle cost, $/ton*	3,395	3,351	3,361	3,242
	÷ life cycle cost, $/ton	44	Base	10	-109
	÷ life cycle cost, %	1.3	Base	0.3	-3.3
	Simple payback period, yrs.	Base	7.2	8.5	1.9
2 chillers	*Life cycle cost, $/ton*	3,531	3,306	3,300	3,189
	÷ life cycle cost, $/ton	225	Base	-6	-117
	÷ life cycle cost, %	6.8	Base	-0.2	-3.5
	Simple payback period, yrs.	Base	3.3	3.4	1.0
3 chillers	*Life cycle cost, $/ton*	3,636	3,390	3,373	3,270
	÷ life cycle cost, $/ton	246	Base	-17	-120
	÷ life cycle cost, %	7.3	Base	-0.5	-3.6
	Simple payback period, yrs.	Base	3.7	3.6	1.1
4 chillers	*Life cycle cost, $/ton*	3,728	3,475	3,465	3,347
	Life cycle cost, $/ton	253	Base	-10	-128
	÷ life cycle cost, %	7.3	Base	-0.3	-3.7
	Simple payback period, yrs.	Base	4.2	4.2	1.2

In summary:

- The variable flow, primary-only had the lowest life-cycle cost and shortest payback period of the study system types for all chiller configurations.
- The number of chillers significantly impacts the payback period and life cycle cost.
- The number of chillers with the lowest life-cycle cost for the variable flow system types was either the 2 or 3 in all cases.

∉# Variable primary flow energy life cycle cost savings relative to constant flow primary/variable flow secondary systems increased with increasing number of chillers.

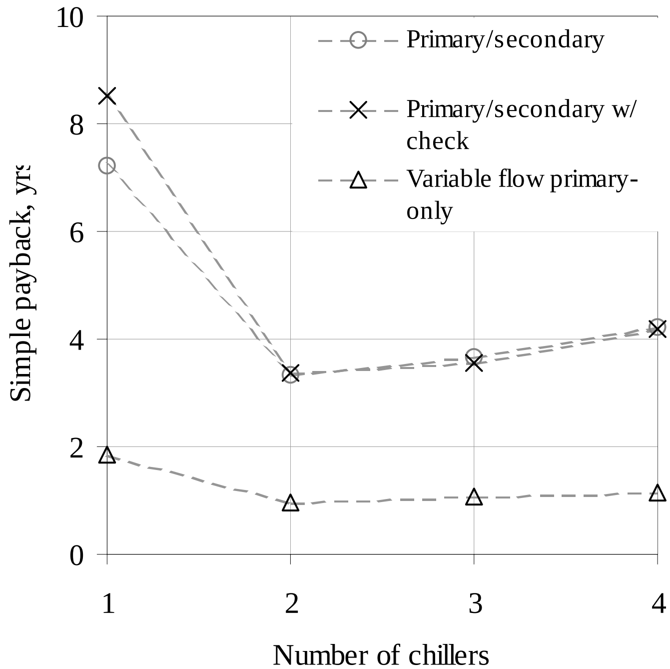


Figure 3-45: Simple payback period as a function of number of chillers for base cases

Chilled Water ÷T

Table 3-39 provides the comparison of life-cycle costs and payback periods for the study chilled water ÷T's of the Houston office building. Because annual energy costs decrease with improving chilled water ÷T's, the systems simulated with favorable ÷T's produced the lowest life-cycle cost and those simulated with unfavorable ÷T's the highest life-cycle costs. Payback periods were based on constant flow, primary-only system types. Because the constant flow, primary-only systems consumed the same energy for all chilled water ÷T models; the variable flow systems generated the most favorable payback periods in the favorable ÷T scenario.

In all cases the variable flow, primary-only system had the lowest life cycle cost and shortest simple payback period. The variable primary flow system experienced the greatest life cycle cost savings relative to the constant flow primary/variable flow secondary system when simulated using the favorable ÷T model.

Figure 3-46 shows the impact of favorable ÷T on simple payback period for the Houston office building. Systems with favorable ÷T characteristics tended to have shorter payback periods. Reductions of payback period were no more than 6 months for any system type and chiller configuration.

Table 3-39: Comparison of life-cycle costs and payback periods for the study chilled water ÷T's of the Houston office building.

CHW system type		Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with check valve	Variable flow, primary-only
Favorable	*Life cycle cost, $/ton*[1]	3,396 to 3,728	3,287 to 3,480	3,292 to 3,478	3,168 to 3,336
	÷ *life cycle cost, $/ton*[1]	50 to 271	Base	9 to -2	-112 to -145
	÷ *life cycle cost, %*[1]	1.5 to 8.0	Base	0.3 to -0.1	-3.4 to -4.2
	Payback period, yrs.[1]	Base	3.1 to 6.8	3.3 to 8.0	0.8 to 1.8
Constant	*Life cycle cost, $/ton*[1]	3,395 to 3,728	3,306 to 3,475	3,300 to 3,465	3,189 to 3,347
	÷ *life cycle cost, $/ton*[1]	44 to 252	Base	10 to -17	-109 to -128
	÷ *life cycle cost, %*[1]	1.3 to 7.3	Base	0.3 to -0.3	-3.3 to -3.7
	Payback period, yrs.[1]	Base	3.3 to 7.2	3.4 to 8.5	1.0 to 1.9
Unfavorable	*Life cycle cost, $/ton*[1]	3,395 to 3,728	3,377 to 3,607	3,387 to 3,549	3,280 to 3,432
	÷ *life cycle cost, $/ton*[1]	18 to 127	Base	9 to -57	-97 to -117
	÷ *life cycle cost, %*[1]	0.5 to 3.6	Base	0.3 to -1.6	-2.9 to -4.9
	Payback period, yrs.[1]	Base	5.7 to 10.0	4.7 to 11.8	1.3 to 2.4

Note: [1]Ranges represent values for all four study chiller configurations

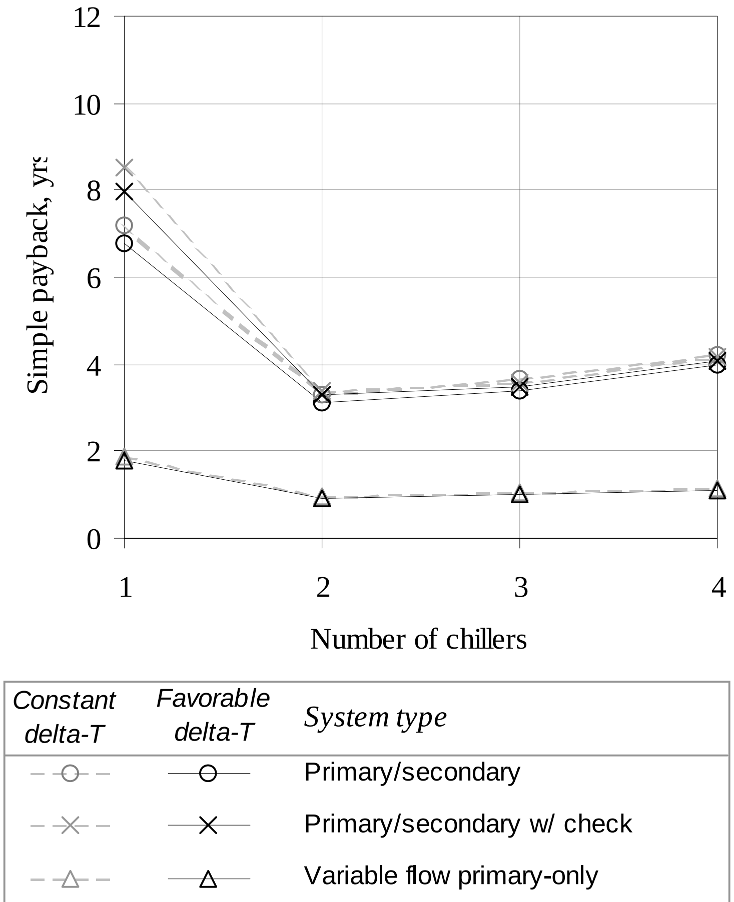


Figure 3-46: Impact of the favorable ÷T model on simple payback period for the Houston office building case.

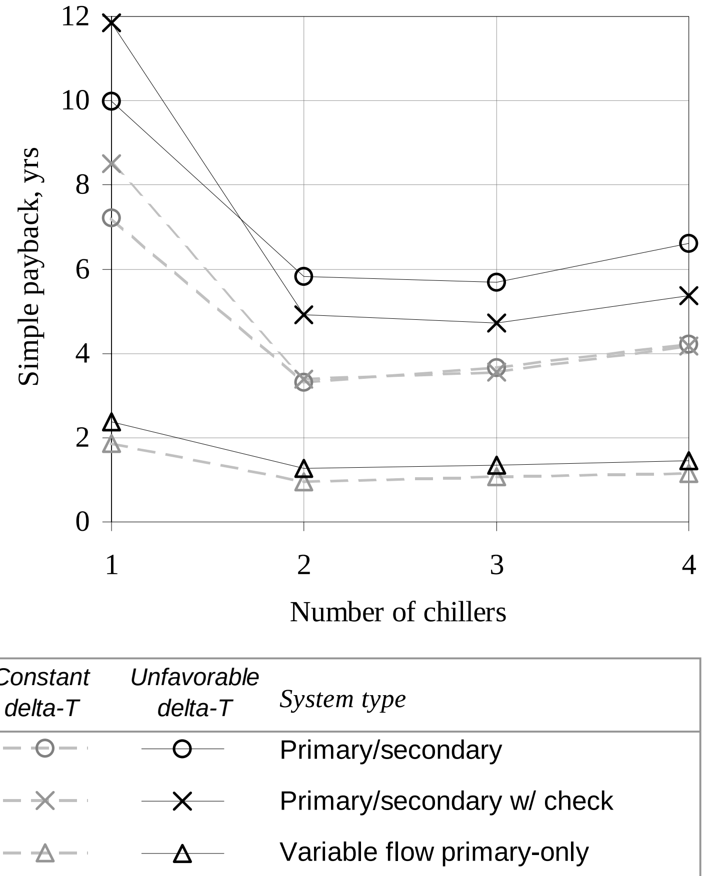


Figure 3-47: Impact of unfavorable ÷T model on simple payback period of the Houston office building.

Figure 3-48 shows the impact of favorable ÷T on life cycle cost for the base case. Favorable ÷T decreased the life cycle cost, but not significantly. As was the case with the constant ÷T scenario, the two chiller configuration returned the minimum life cycle cost.

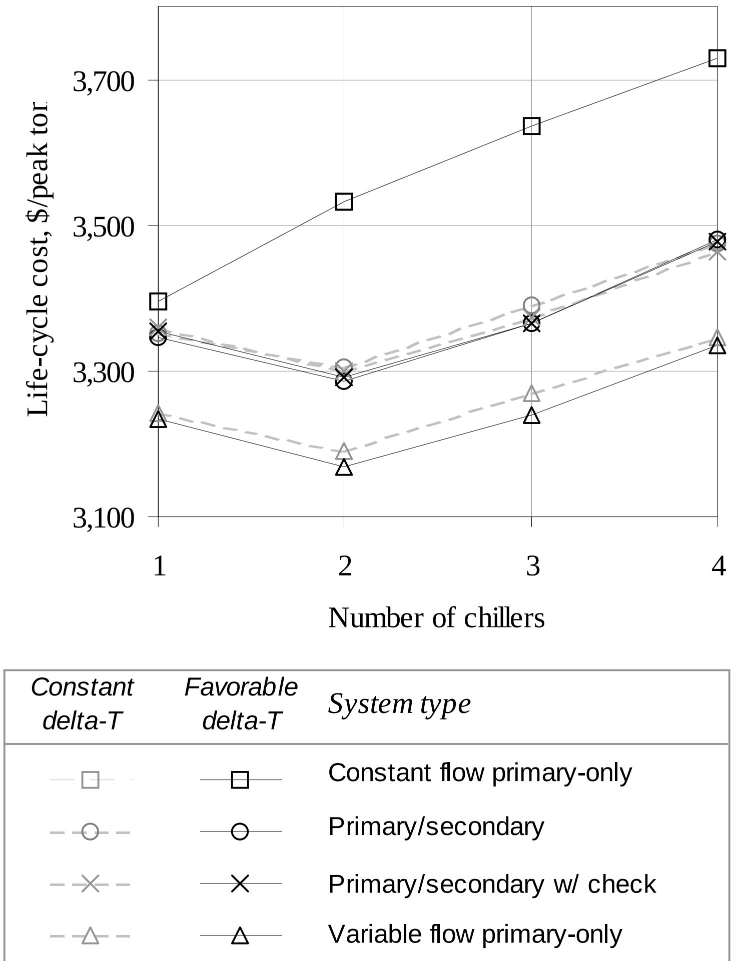


Figure 3-48: Impact of the favorable ÷T on life cycle cost of Houston office building

Figure 3-49 shows the impact of unfavorable ÷T on life cycle cost for the base case. The unfavorable ÷T increased the life cycle cost of each of the variable flow system alternatives. It also had the affect of changing the minimum life cycle cost point from the two to the one chiller configuration.

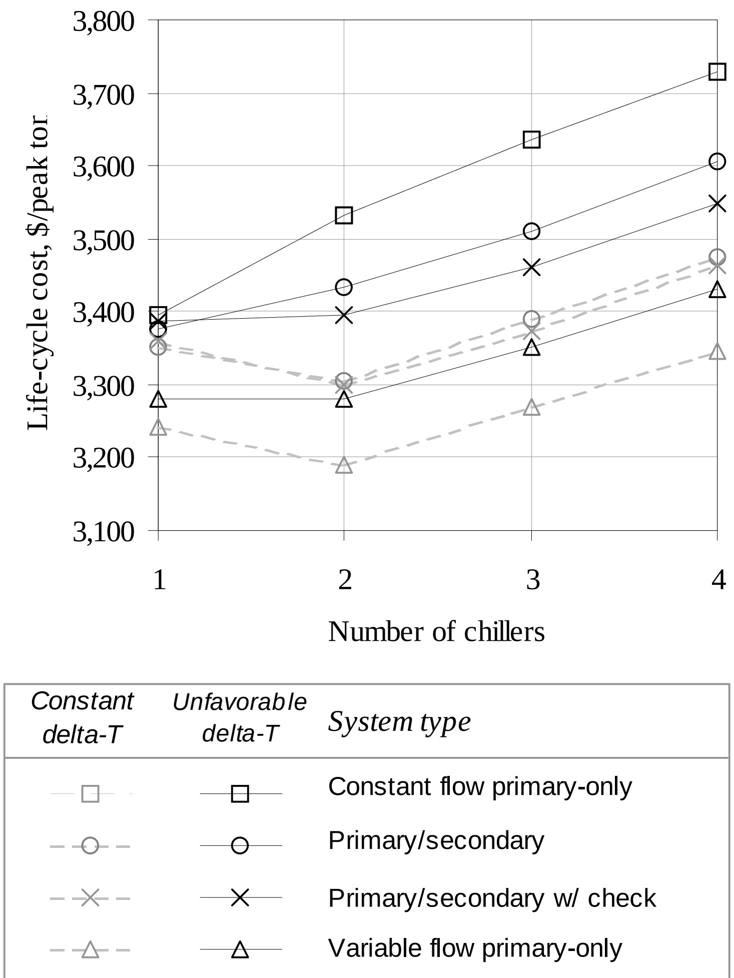


Figure 3-49: Impact of the unfavorable ÷T model on life cycle cost of Houston office building

Relative to the simulations performed using the constant ÷T scenario:

- Favorable ÷T decreased the life cycle cost and shortened the payback period of the variable flow cases.
- Unfavorable ÷T had the opposite effect on life cycle cost and payback period. In addition, unfavorable ÷T tended to reduce the number of chillers that minimized life cycle cost from two chillers to one.
- Favorable ÷T increased the life cycle cost savings of the variable primary flow systems relative to primary/secondary and shortened the simple payback period.

Cooling Load Type

Table 3-40 compares life-cycle costs as a function of load type for Houston climate and constant chilled water ÷T. Table 3-41 shows life-cycle costs in $/design ton and payback periods. Payback periods for the medical facility and district plant cases were generally 50-percent of the office building cases.

Primary-only variable flow payback period was short and the life cycle savings low relative to primary/secondary systems.

Table 3-40: Comparison of life-cycle costs ($) for the system types serving the study Houston load types with constant ÷T.

CHW system type		Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with a check valve	Variable flow, primary-only
Office building	*Life-cycle cost, $[1]#*	1,697,694 to 1,863,869	1,652,770 to 1,737,315	1,649,834 to 1,732,440	1,594,490 to 1,673,637
	# life-cycle energy cost, $[1]	21,988 to 126,553	Base	4,671 to -8,657	-54,732 to -63,678
	# life-cycle energy cost, %[1]	1.3 to 7.3	Base	0 to -1	-3 to -4
Medical facility	*Life-cycle cost, $[1]#*	5,162,125 to 5,534,133	4,729,136 to 4,942,147	4,690,525 to 4,957,114	4,572,378 to 4,750,974
	# life-cycle energy cost, $[1]	219,978 to 789,128	Base	14,967 to -38,611	-128,460 to -191,173
	# life-cycle energy cost, %[1]	4.5 to 16.6	Base	0 to -1	-3 to -4
District plant	*Life-cycle cost, $[1]#*	16,806,979 to 17,081,134	14,928,205 to 15,097,767	14,814,346 to 15,311,302	14,513,195 to 14,876,311
	# life-cycle energy cost, $[1]	1,394,987 to 1,983,367	Base	-100,690 to -137,131	-401,943 to -535,681
	# life-cycle energy cost, %[1]	9 to 13	Base	-1	-3 to -4

Table 3-41: Comparison of life-cycle costs ($/ton) and payback periods for the system types serving the study Houston load types with constant ÷T.

CHW system type		Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with check valve	Variable flow, primary-only
Office building	*Life cycle cost, $/ton*[1]	3,395 to 3,728	3,306 to 3,475	3,300 to 3,465	3,189 to 3,347
	÷ life cycle cost, $/ton[1]	44 to 252	Base	10 to -17	-109 to -128
	÷ life cycle cost, %[1]	1.3 to 7.3	Base	0.3 to -0.3	-3.3 to -3.7
	Payback period, yrs.[1]	Base	3.3 to 7.2	3.4 to 8.5	1.0 to 1.9
Medical facility	*Life cycle cost, $/ton*[1]	3,441 to 3,689	3,153 to 3,295	3,127 to 3,305	3,048 to 3,167
	÷ life cycle cost, $/ton[1]	147 to 526	Base	10 to -26	-86 to -127
	÷ life cycle cost, %[1]	4.5 to 16.6	Base	0.3 to -0.8	-2.7 to -3.9
	Payback period, yrs.[1]	Base	1.5 to 3.3	1.5 to 3.9	0.5 to 1.0
District plant	*Life cycle cost, $/ton*[1]	3,727 to 3,808	3,328 to 3,436	3,303 to 3,414	3,235 to 3,317
	÷ life cycle cost, $/ton[1]	309 to 440	Base	-22 to -30	-90 to -120
	÷ life cycle cost, %[1]	9.0 to 13.1	Base	-0.6 to -0.9	-2.7 to -3.5
	Payback period, yrs.[1]	Base	1.2 to 1.6	1.2 to 1.6	0.4

Note: [1]Ranges represent values for all four study chiller configurations

In summary:

- Life cycle savings per design ton were on the order of $100 for variable flow, primary-only systems relative to primary/secondary systems. System type had only a small effect on the magnitude of savings.
- Simple payback period for variable flow, primary-only systems relative to constant flow primary-only systems was less than two years in all cases. Payback period was somewhat sensitive to load type, with the high load factor district cooling system having the shortest payback.

Climate

Table 3-42 compares office building life cycle costs for the Houston and Syracuse climate. Due to the long cooling season, life cycle costs for Houston were nearly 1.5-times that of the Syracuse climate.

Table 3-42: Life cycle cost comparisons of Houston study system types with a constant ÷T.

CHW system type		Constant flow primary-only	Constant flow primary/ variable flow secondary	Primary/ secondary with a check valve	Variable flow, primary-only
Houston	*Life-cycle cost, $/ton[1]#*	3,395 to 3,728	3,306 to 3,475	3,300 to 3,465	3,189 to 3,347
	÷ life-cycle cost, $/ton[1]	44 to 252	Base	10 to -17	-109 to -128
	÷ life-cycle cost, %[1]	1.3 to 7.3	Base	0.3 to -0.3	-3.3 to -3.7
	Simple payback period, yrs.	Base	3.3 to 7.2	3.4 to 8.5	1.0 to 1.9
Syracuse	*Life-cycle cost, $/ton[1]#*	2,103 to 2,435	2,095 to 2,301	2,104 to 2,299	2,016 to 2,194
	÷ life-cycle cost, $/ton[1]	8 to 135	Base	9 to -4	-79 to -107
	÷ life-cycle cost, %[1]	0.4 to 6.1	Base	0.4 to -0.2	-3.8 to -4.7
	Simple payback period, yrs.	Base	5.3 to 11.8	5.4 to 13.9	1.5 to 3.0

Note: [1]Ranges represent annual energy cost per design ton for all four study chiller configurations

Figure 3-50 shows the simple payback period for Houston and Syracuse office building cases. The simple payback period in Syracuse was nearly doubled and as much as 5 years longer.

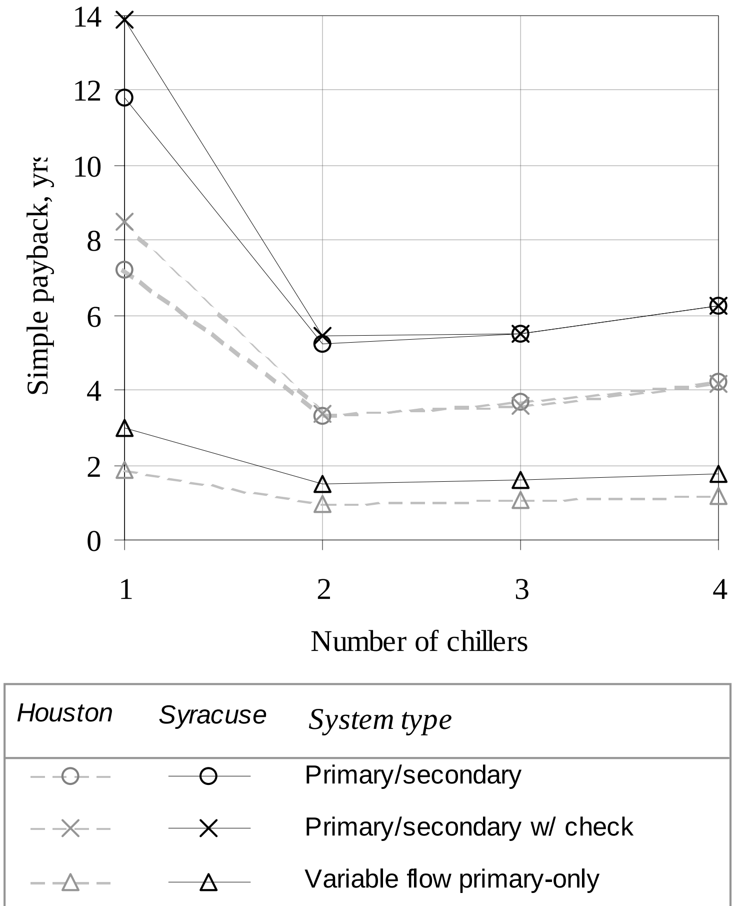


Figure 3-50: Impact of the Syracuse climate on simple payback period for the base case

Figure 3-51 compares life cycle costs for Syracuse and Houston office buildings. Syracuse life cycle costs decreased by nearly 50-percent for all system types relative to those for the Houston cases.

Table 3-42 shows the life cycle cost savings relative to the constant flow primary/variable flow secondary system type. Relative to Houston cases, the life cycle cost savings of the Syracuse variable flow, primary-only system decreased by between 28 and 16-percent depending on the chiller configuration.

Syracuse cases experienced longer payback periods and lower life cycle costs relative to those in the Houston climate. Variable-primary-flow life cycle cost savings relative to the constant flow primary/variable flow secondary system were greater for the Houston cases than for Syracuse.

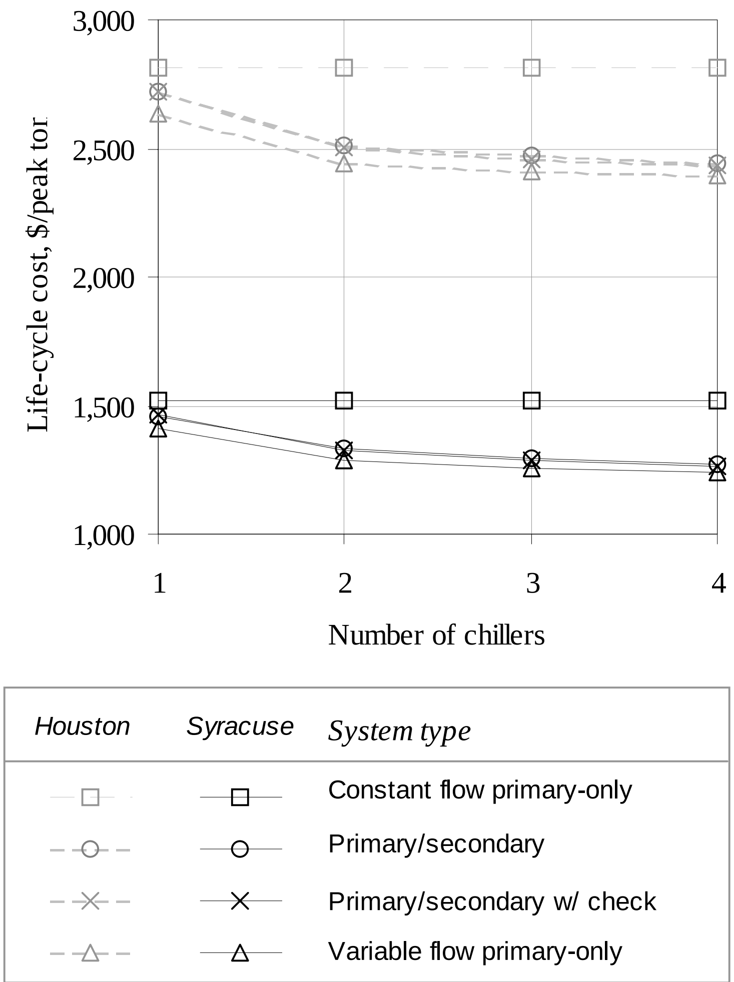


Figure 3-51: Impact of the Syracuse climate on the life cycle cost for the base case

In summary:

- Life-cycle cost was greater for Houston cases than for Syracuse cases.
- The variable flow, primary-only system maintained the lowest life-cycle cost in all cases.
- Variable flow, primary-only and primary/secondary check valve system annual energy cost savings relative to constant flow primary/variable flow secondary systems were greatly affected by climate. The savings in the Houston case were larger by 1.2-times or more than savings in Syracuse.

3.5.4 Sensitivity Analysis

Sensitivity analysis was performed to determine whether plausible changes in economic parameters change the conclusions of the base analysis. Energy and demand charges and fuel price indices were varied. Table 3-43 provides the energy and demand charge scenarios considered in the sensitivity analysis.

Table 3-43: Energy and demand charge scenarios used in sensitivity analysis

Scenario	*Energy charge, $/kWh*	*Demand charge, $/kW*
Base analysis	0.035	12.00
Decreased demand charge	0.035	8.00
Increased demand charge	0.035	16.00
Decreased energy charge	0.02	12.00
Increased energy charge	0.05	12.00

Figures 3-52 and 3-53 show the effect of demand charge changes on simple payback period. Increasing the demand charge had the effect of slightly decreasing the simple payback period of the primary/secondary systems with a single chiller configuration and had relatively no effect on multiple chiller configurations. The opposite was true of the decreased demand charge—payback periods were longer for single chiller configurations. Changes to demand charges had little impact on simple payback period because the demand charge did not affect the energy cost savings drastically.

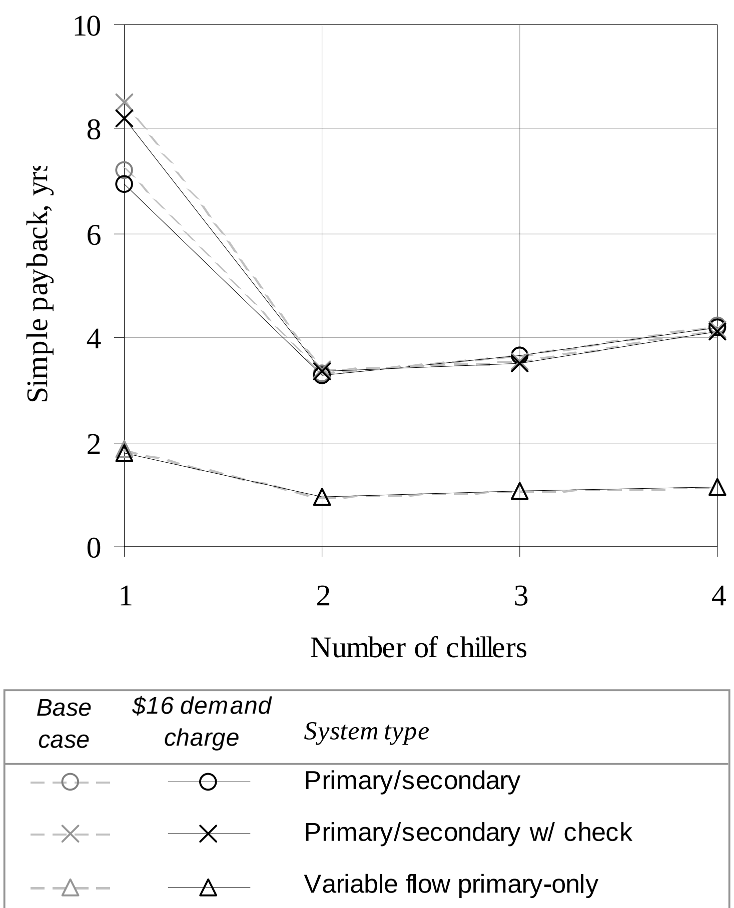


Figure 3-52: Impact of increased demand charge on simple payback period

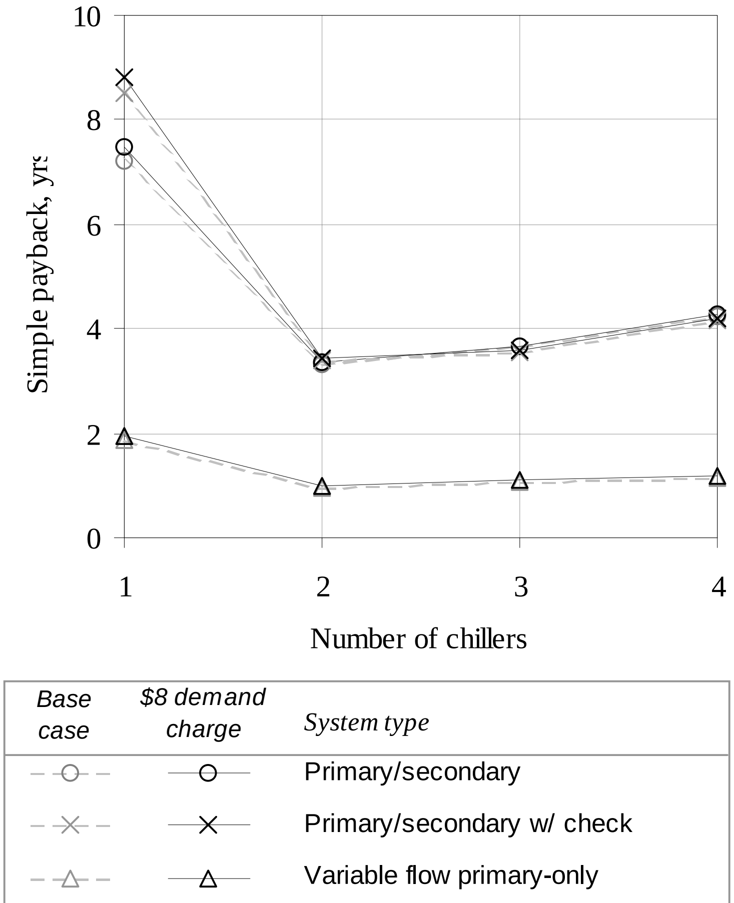


Figure 3-53: Impact of decreased demand charge on simple payback period

Figures 3-54 and 3-55 show the effect of changes to energy charge on simple payback period. An increase in energy charge resulted in a decrease in payback period for all cases with the single chiller configurations experiencing the greatest drop in payback period. A decrease in energy charge had a greater effect on simple payback periods, as primary/secondary alternatives went from having an acceptable payback period to one greater than 5 years in all multiple chiller cases and greater than 11 years in both single chiller cases.

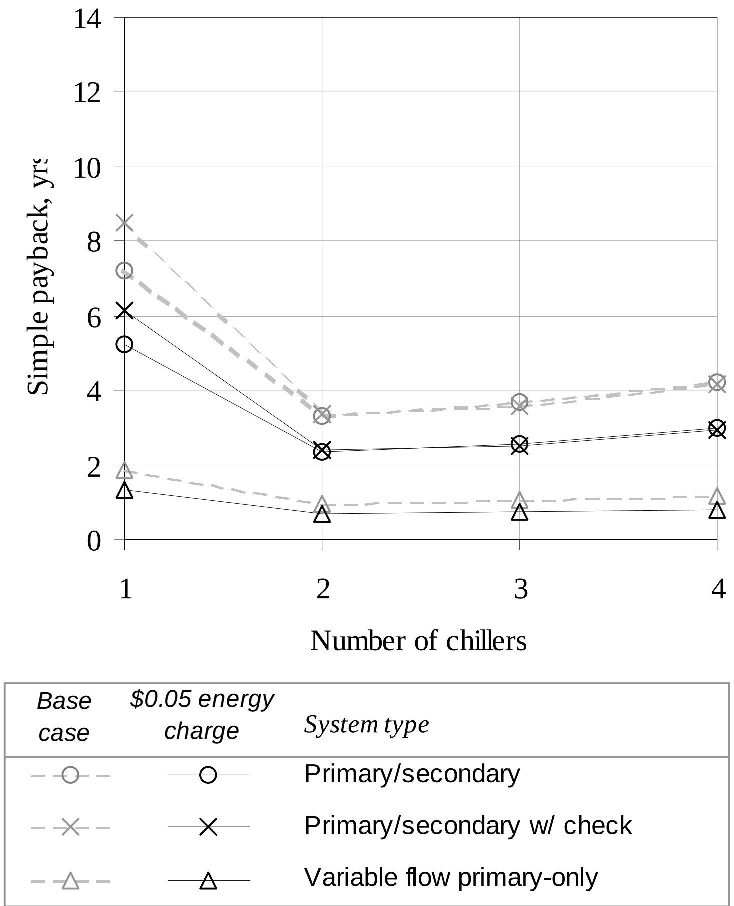


Figure 3-54: Impact of increased energy charge on simple payback period

In summary, the simple payback period and life cycle cost results were particularly sensitive to changes in energy charges. Changes in demand charge resulted in insignificant changes to base case results.

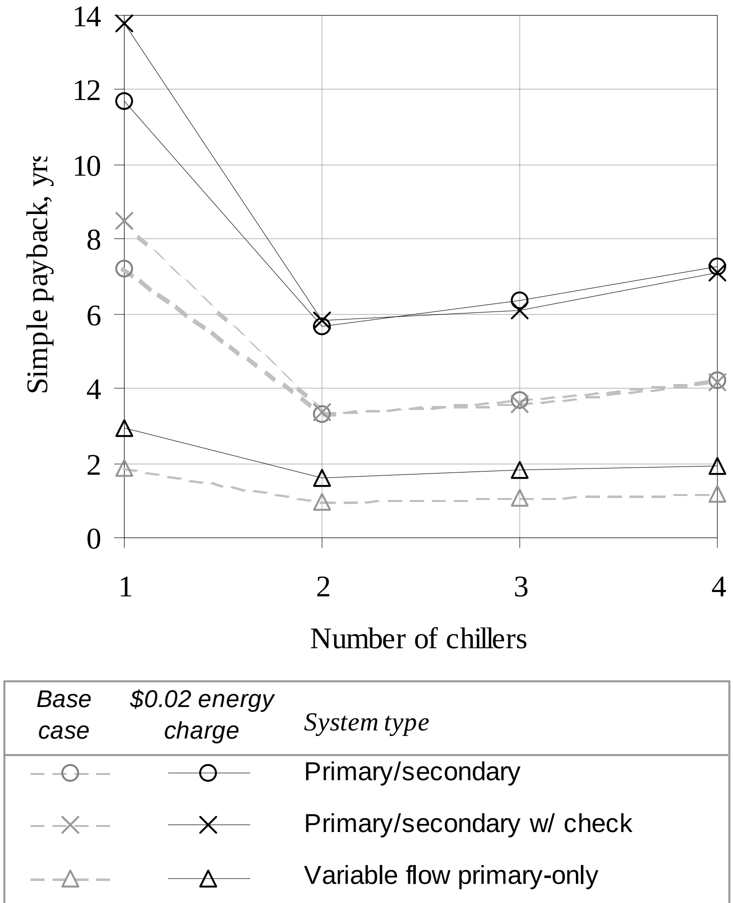


Figure 3-55: Impact of decreased energy charge on simple payback period

Variations in projected cost indices of +/-10-percent were used to determine the sensitivity of life cycle cost to changes in projected cost indices. Figure 3-56 shows a plot of the projected cost indices used in the sensitivity analysis.

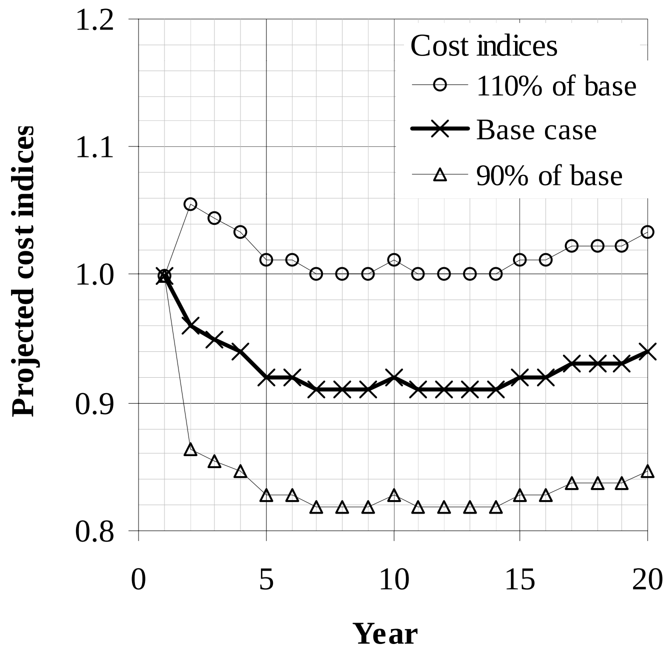


Figure 3-56: Cost indices used in sensitivity analysis

Figure 3-57 shows a comparison of the life-cycle costs for the base case cost indices and a case where cost indices were 10-percent lower than the base case. When the cost indices were 10-percent less than that of the base case the life cycle costs for all system alternatives decreased by between 6 and 8-percent. The two chiller configuration remained the lowest life cycle cost alternative for the variable flow system types.

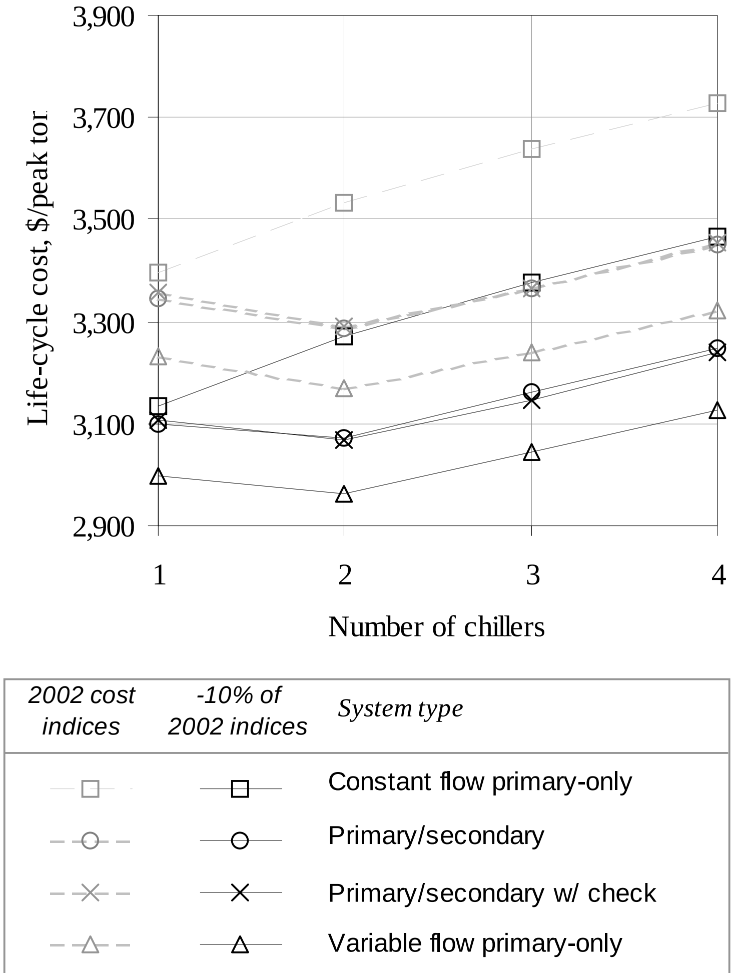


Figure 3-57: Sensitivity of life cycle costs to a 10-percent decrease in cost indices

Figure 3-58 shows a comparison of the life-cycle costs for the base case cost indices and a case where cost indices were 10-percent greater than the base case. When the cost indices were 10-percent higher than that of the base case the life cycle costs for all system alternatives increased by between 6 and 8-percent. The two chiller configuration remained the lowest life cycle cost alternative for the variable flow system types.

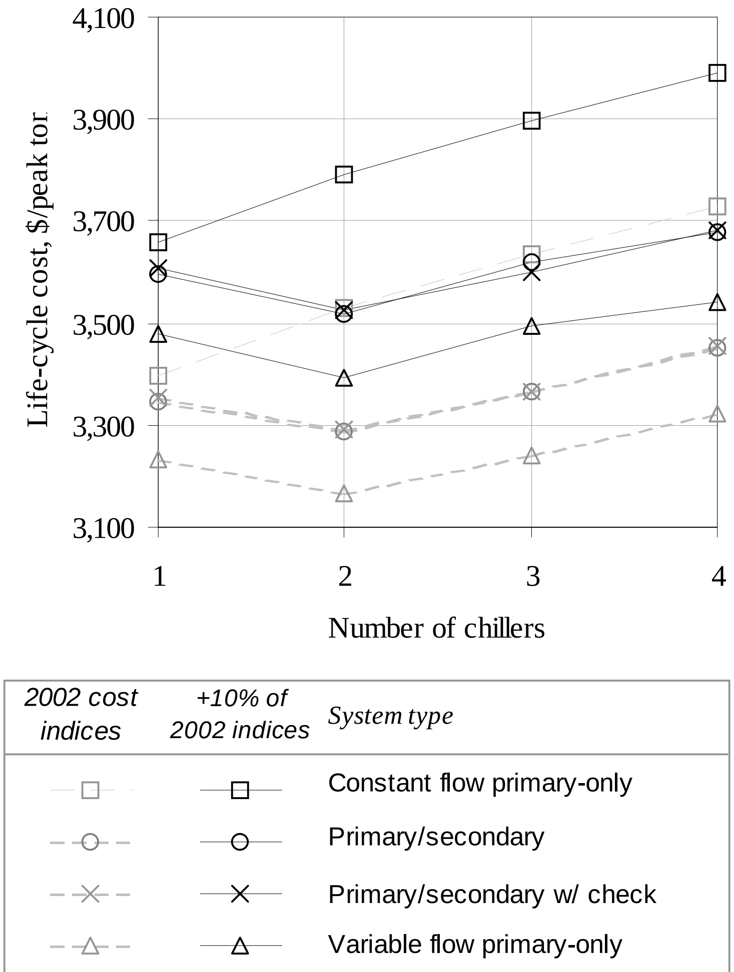


Figure 3-58: Sensitivity of life cycle costs to a 10-percent increase in cost indices

Figure 3-59 provides the life cycle cost savings for the primary/secondary check valve and variable flow, primary-only systems relative to the constant flow primary/variable flow secondary cases. In both cases the life cycle cost savings did not change by more than 1-percent when other life cycle cost indices were considered.

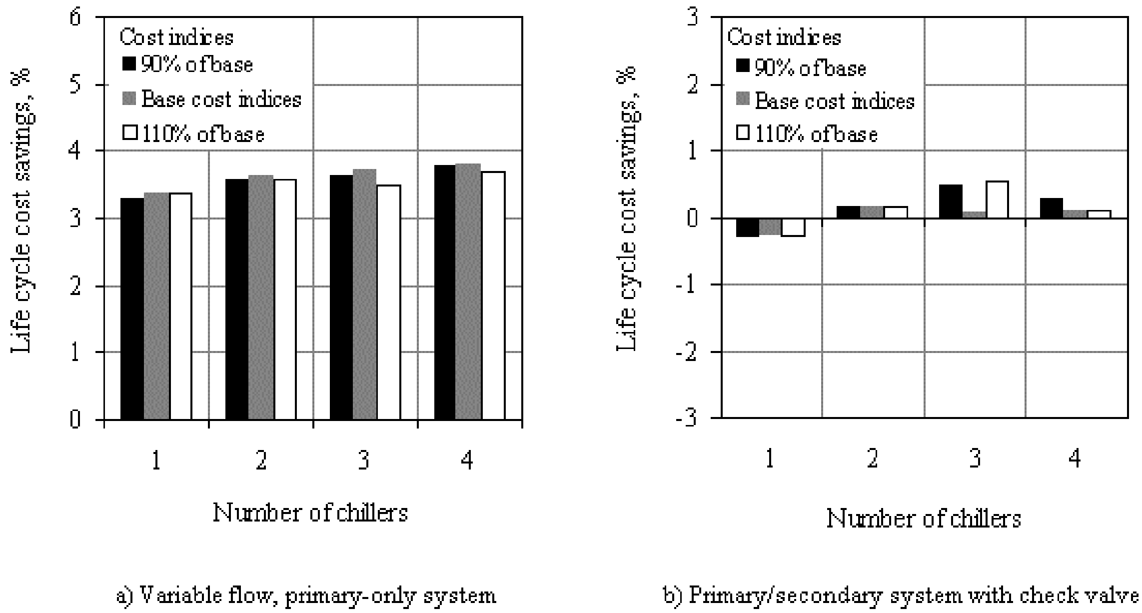


Figure 3-59: Life cycle cost savings for the variable flow, primary-only (a) and primary/secondary check valve system (b) relative to constant flow primary/variable flow secondary system

In summary, projected cost indices had a significant impact on life cycle costs of all cases considered. However, the change in variable flow, primary-only and primary/secondary check valve life cycle cost savings relative to constant flow primary/variable flow secondary systems was generally less than 0.5-percent.

4. CONCLUSIONS

The two major thrusts of this study have been to summarize the state of the art of variable primary flow and compare the energy performance and feasibility of variable primary flow relative to other system architectures. The state of the art was arrived at through review of information from published sources, a survey of chiller manufacturers, designers, and system owners/operators, and follow-up correspondence with a number of the survey respondents. The performance of variable primary flow was compared to three other chilled water system types using a parametric modeling study. The parameters used were the following: load type, climate, number of chillers, and chilled water temperature differential versus cooling load characteristic.

4.1 State of the Art of Variable Primary Flow

As documented in the literature and through discussion with chiller manufacturers, chillers outfitted with modern controls are capable of practical variable primary flow operation. Advances in capacity controls, freeze protection, and flow detection have increased chiller stability—a particular concern in variable primary flow applications because evaporator flow rates can change abruptly during chiller staging.

Manufacturers are providing more detailed variable flow application guidance than in the recent past, including chilled water velocity limitations and rates of flow variation, for most chiller models. Recommended evaporator water velocity limits are roughly 3 to 11 ft/s for flooded evaporators and chiller-specific for direct-expansion-type evaporators. This range of velocity provides sufficient opportunity for flow rate turndown and evaporator overflow. However, an evaporator design velocity must be selected to accommodate the anticipated needs of the system. Rates of flow variation range from less than 2 percent to as much as 30 percent of design flow per minute depending on the make and model of the chiller and the turnover time of the chilled water system. It is important that the system's anticipated turnover time be considered when determining the rate of flow variation permitted.

Variable primary flow systems are perceived to be more complicated than comparable primary/secondary systems. This is partly because chiller staging requires more care in order to achieve stable operation and anticipated energy savings. Chiller isolation valves should open and close at a rate that corresponds with the response time of the chiller's capacity control. The low flow bypass control required in most variable primary flow systems adds further complexity to the system. The bypass and valve should be sized for the minimum required flow rate of the largest chiller and should be located close to the plant. A flow measurement device that has sufficient turndown to measure flow accurately throughout the range of flow rates anticipated.

The literature contains considerable discussion of constant and variable speed chiller staging methods. Most are based on the concept of optimizing plant efficiency by either minimizing the number of chillers and auxiliary equipment on line for constant speed chillers or maximizing the number of chillers on line for variable speed chillers. There are several possible indicators available for determining the proper time to stage chiller capacity (i.e., chilled water flow rate, calculated cooling load, compressor current, leaving-chilled-water-temperature, etc.).

Several trends can be identified in the literature and survey responses:

- Primary-only pumping arrangements rather than multi-level pumping systems dominate variable primary flow system designs due to reductions in first cost.
- A majority of variable primary flow systems use bypass with control valve for low flow control rather than continuous bypass using three way valves or control of pump speed.
- Most designers prefer the use of pressure differential measurement across the chiller evaporator to self contained flow meters for flow measurement in variable primary flow systems.

When compared to constant flow primary/variable flow secondary chilled water systems designers and system owners cited the following benefits of variable primary flow:

- Energy savings that result in reduced operating costs
- Lower first cost and less space due to fewer plant components, namely chilled water pumps and associated piping and accessories.
- Ability to improve chiller loading in systems experiencing less-than-design chilled water ÷T.

These same survey respondents also commented on the following concerns with regards to variable primary flow systems:

- Lack of support from equipment manufacturers and lack of guidance in the literature.
- Increased control, commissioning, and maintenance costs associated with variable primary flow due to the increased complexity of the system and lack of familiarity of the people involved with performing various tasks during construction, start-up, and operation.
- Chilled water flow stability during plant operation, particularly when staging chiller capacity.

Nearly half of the survey respondents have not designed variable primary flow systems. Those without variable primary flow expertise identify of lack of guidance as a reason why they have not designed variable primary flow systems. While most claims of variable primary flow superiority over other system alternatives revolve around energy and first cost savings, there is little in the way of quantitative evidence; most arguments in favor of variable primary flow tend to be based anecdotes and generally lack rigor. Designers and system owners with variable primary flow experience generally are willing to consider the use of variable primary flow for future projects.

4.2 Parametric Study

The variable flow, primary-only alternative provided an energy efficient solution and low capital investment alternative to other variable flow system types investigated. This resulted in short payback periods and lower life cycle costs than comparable systems.

Overall, the variable flow, primary-only system reduced total annual plant energy by 3 to 8-percent, reduced the first cost by 4 to 8-percent, and reduced the life cycle cost by 3 to 5-percent for all cases relative the conventional constant primary flow/variable secondary flow system. Differences in annual energy costs between the various study system types closely tracked energy consumption because peak demand was not strongly affected by the type of pumping system.

Several parameters significantly impacted the energy and life cycle cost savings and simple payback period of the variable primary flow system relative to other system alternatives. These included the number of chillers, climate, and chilled water temperature differential. In particular, the following factors tended to maximize variable primary flow energy savings relative to other system alternatives:

- Chilled water plants with fewer chillers
- A longer, hotter cooling season
- Less than design chilled water temperature differential

The load type had little impact on variable primary flow energy savings relative to other system alternatives. Although the magnitude of the savings was much larger for load types with greater cooling loads, when savings were standardized on a per design ton basis the differences were relatively small.

Chilled water pump and chiller auxiliary energy savings accounted for essentially all savings, while differences in chiller energy use were not significant. Variable flow, primary-only systems had chilled water pump energy use 25 to 50 percent lower than that of primary/secondary chilled water systems. In systems with two or more chillers configured in parallel, chiller auxiliary energy savings were 13-percent or more.

The addition of a bypass check valve to the typical primary/secondary system architecture resulted in total plant energy savings of up to 4 percent and a life cycle cost savings of not more than 2 percent.

Savings occurred only when chilled water ÷T's were less than the design value. Chilled water pump savings were 5 percent or less and chiller auxiliary savings were 13 percent or less.

Conclusions based on these results must be qualified in several respects.

- Only constant-speed, electric-driven, water-cooled centrifugal chillers were considered.
- Multiple units chillers, pumps and cooling towers were equally sized and configured in parallel.
- Chilled water plants were simulated using quasi-static models with an hourly time increment.
- The constant flow, primary-only system type, used as a basis for comparison of all variable flow systems, was modeled with the assumption that all equipment was in operation all the time.
- A single, simple electric rate structure was used for the economic analysis.

4.3 Future research

Continued research and testing of the performance of variable primary flow system types is needed. The following are some of the areas that need to be addressed:

- Additional chiller configurations, i.e., chillers configured in series and chillers of unequal sizes.
- Applicability to absorption chillers. More documentation of tube velocity limits and acceptable rates of flow variation are needed.
- Use of variable frequency drive chillers and a study focused on alternatives to optimize variable speed chillers in variable primary flow systems.
- Collection and analysis of measured variable primary flow performance data.
- Investigation of the dynamic effects of variable primary flow.
- Feasibility of converting constant flow primary/variable flow secondary systems to variable primary flow without the use of a check valve.

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