New York State Energy Research and Development Authority

Use of Fisonic Devices in Con Edison Service Territory

Final Report

January 2009

No. 11-19

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Use of Fisonic Devices in Con Edison Service Territory

Final Report

Prepared for the

New York State
Energy Research and
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Executive Summary

Purpose of the Study

Most of the buildings in New York City, supplied by Con Edison district steam system, currently convert the district steam into hot water in tube and shell heat exchangers. The hot water is then distributed by electrically driven pumps throughout the building for space heating and domestic hot water service. The steam also is supplied to absorption or steam driven chillers for generation of chilled water. After steam utilization, the condensate is discharged to the city sewer system. In order to reduce the condensate temperature from 215-220°F to about 150°F, which is the city sewer requirement, the condensate is mixed with cold potable water. The described system requires expensive heat exchangers (with associated maintenance cost), electrically driven pumps for hot water transport (with associated maintenance and electric cost) and substantial amounts of cold potable water.

The main purpose of this study was to investigate the feasibility of Fisonic Devices (FD) in order to improve the end-use energy efficiency and the environment for steam customers in the Con Edison service territory and New York State. The project was sponsored by New York State Energy Research and Development Authority (NYSERDA), Con Edison, and by Hudson Fisonic Corporation (involved in the development of the FDs) and was performed in close cooperation with Con Edison district steam customers.

FDs are supersonic, condensing heat pumps with a patented internal geometry. The FD causes steam and water to mix and accelerate, converting a minute fraction of fluid's thermal energy to physical trust (pump head) with the outlet pressure higher than the pressure of the working medium at the inlet of the FD. The FD replaces both the tube and shell heat exchanger and the electrically driven pump. The FD heats the building hot water by direct contact with steam and transports the water throughout the building, thus eliminating the tube and shell heat exchanger, and the electrically driven pump. The FD eliminates the terminal temperature difference between steam and water, thus reducing the steam consumption and the amount of cold potable water as compared to the existing system with heat exchangers.

During the project the following tasks have been performed:

- Assessment of potential applications of the fisonic technology
- Site visits and surveys of Con Edison representative customers
- Assessment of customer steam loads and costs
- Capital and O&M cost estimates for selected FD alternatives
- Comparison of the FD and existing system costs and feasibility analysis
- Development of a commercialization plan for manufacturing the FDs in New York State

Evaluated Customers and Study Results

Assessment of the Con Edison Headquarters Building

This building is located at 4 Irving Place and occupies 27 stories, with a total area of 1,170,000 ft². The building is supplied with steam from the Con Edison district steam system with a pressure of 170–200 psig. The steam pressure is reduced by three pressure reducing valves to 3–5 psig for space heating and domestic hot water. The building is equipped with fifty-four large fan rooms with steam and cooling coils. The perimeter space hot water radiation system is supplied by two tube and shell heat exchangers and by four electric driven pumps. The space hot water temperature is modulated from 90°F to 180°F. Two instantaneous heat exchangers supply the domestic hot water load. The steam condensate is collected in a dilution tank located in the basement. The condensate temperature is reduced by the addition of cold potable water. The mixture is then pumped to the salvage tank located at the 22nd floor from which the water, by gravity, is used for toilets. The excess condensate/cold water mixture is discharged to the sewer.

The existing space heating system, which can be retrofitted using the FDs, is the hot water system supplying heat to the building periphery baseboards. A conceptual design of this system equipped with the FDs was developed. In this system when compared with the existing system, the pressure reducing valves, heat exchangers and the hot water circulating pumps are eliminated. This equipment is replaced with the FDs that provide direct contact water heating with steam and water pumping. No steam pressure reduction valves are required; the district steam pressure is reduced by the FD. In order to keep the FD discharge water flow constant, a single electric driven pump is installed. This pump will operate for a limited number of hours during the year. The hot water return temperature is reduced by heating the DHW in a plate and frame heat exchanger. Afterward the return water is discharged to the existing dilution tank.

The operating parameters, capital, and O&M cost estimates for the system retrofit with FDs have been developed. The annual steam consumption for the FD's system is reduced by 16.4% when compared with the existing system. This is the result of the direct contact steam condensation in the FD and condensate mixing with the hot water. The make-up water flow rate and the total water flow rate to be discharged to the sewer for the FD system are reduced by about 50%. The simple payback for the system retrofit is estimated at two years.

Assessment of the Woolworth Office Building

This building is located at 233 Broadway and occupies 56 floors, with a total area of 900,000 ft^2. Presently, only 28 floors are occupied. The building is supplied with steam from the Con Edison district steam system with a pressure of 170–180 psig. The steam pressure is reduced by two pressure reducing valves to 3 psig for space heating and domestic hot water. The cooling is provided by two 400 ton and two 300 ton electric driven chillers. The perimeter space hot water radiation is supplied from four tube and shell heat exchangers, and by four electric driven pumps. The space hot water temperature is modulating from 90°F to 135°F. Four instantaneous heat exchangers supply the domestic hot water load. The condensate from the heat exchangers is used to preheat the DHW and Afterward mixed with cold water and discharged into the sewer system.

The existing space heating system, which can be retrofitted by the FDs is the hot water system supplying heat to the building periphery baseboards and the air conditioning hot water coils. The DHW system cannot use the FDs because steam cannot be mixed with hot water for sanitary requirements.

In the retrofitted system using the FDs, when compared with the existing system, the pressure reducing valves, heat exchangers and the hot water circulating pumps are eliminated. This equipment is replaced with the FDs which provide direct contact water heating with steam and water pumping. In order to keep the FD discharge water flow constant a single electric driven pump is installed. The pump will operate for a limited number of hours during the year. The hot water return temperature is reduced by heating the DHW in a plate and frame heat exchanger. Afterward the return water is discharged to the sewer.

The operating parameters, capital and O&M cost estimates for the retrofit system with FDs have been developed. The annual steam consumption for the FD's system is reduced by 16.4% when compared with the existing system. The make-up water flow rate and the total water flow rate to be discharged to the sewer for the FD system are reduced by about 50%.The simple payback for the system retrofit is estimated at 1.2 years.

Assessment of the Louis Lefkowitz Building

This building is located at 80 Center Street and has nine floors, with a total area of 400,000 ft^2. The Con Edison district steam pressure at the entrance of the building is 170–180 psig. The district steam pressure is reduced by two pressure reducing stations to about 3 -5 psig and used to supply the steam radiators throughout the building. The radiators are not equipped with any thermostatic control valves. The temperature in the rooms is controlled by the window air conditioners.

The conversion of the building to hot water includes installation of thermostatic control valves and vents on each radiator, removal of the trap internals, installation of an expansion tank, temperature control valves, fisonic devices, and a water circulating pump. All the existing steam and condensate piping and radiators remain in place.

The capital and O&M cost estimates for conversion of the building space heating system to hot water with FDs have been developed. The simple payback for the building retrofit is estimated at 4.5 years.

Conclusions

- The study has demonstrated that the use of FDs for Con Edison customers will reduce their annual steam consumption by 16.4% when compared with the existing system.
- The make-up water flow rate and the total water flow rate to be discharged to the sewer for the FD system are reduced by about 50%.
- The simple payback period for retrofitting an existing hot water space heating system with FDs is under two years.
- The simple payback period for conversion of steam heated building to hot water with FDs is about 4.5 years.

The potential benefits of use of FDs for Con Edison customers are estimated as follows:

- **Energy Savings**: assuming the implementation of Fisonic Devices by 30% of the Con Edison customers (540 buildings with current steam consumption of about 8.4 million Mlbs per year), the potential reduction in steam consumption will result in annual cost savings to the customers of about $42.7 million. The potential annual savings of the customers in electric consumption are estimated at $6.7 million.
- **Water and Sewer Savings**: for the above assumptions, the annual savings associated with cold water consumption and sewer discharge are estimated at $4 million.
- **Job Creation**: Using 32 job years per $1 million of energy savings (EPRI, "Guidelines for Assessing the Feasibility of District Energy Projects," Ref.11) it is estimated that project implementation will result in creation of 1,580 job years.
- **Environmental Benefits**: The environmental benefits are estimated as the following pollution reductions in lbs/year: NO_x : 15,000; Particulates : 5,900; VOC : 2,100; and CO_2 : 37,600 ton.

The potential benefits of use of FDs in the State of New York are estimated as follows:

> The annual primary energy consumption in NYS for buildings and industry is about 1,300 Tbtu (NYSERDA Patterns and Trends, December 2002). Assuming that 5% of this energy is supplied by steam with conventional heat exchangers and electric driven pumps, the current steam consumption can be estimated at 65 million Mlbs/yr. Applying the above described methodology for estimating the energy savings with the use of FDs, the potential benefits for NYS are estimated as follows: **Energy Savings**: $380 million; **Water and Sewer discharge savings**: $31 million; **Environmental Pollution**

Reductions: NO_x :116,000 lbs/yr, Particulates : 45,600 lbs/yr, VOC : 16,200 lbs/yr and CO_2 : 290,600 ton/yr.

Future Work

In order to start the FD commercialization plan the following actions are planned:

- Demonstrate the operation of FDs at the Con Edison Headquarter and the Woolworth buildings in New York City.
- Hudson Fisonic Corporation will market the FDs to the Con Edison customers, city and state administration, consulting engineers and HVAC equipment vendors. All entities must be comfortable with the concept of using FDs, and all must be convinced of the cost effectiveness and benefits that it will provide.
- Many existing customers are in the process of replacing or upgrading an existing HVAC system, so local engineers and HVAC contractors are in an ideal position to market FDs by providing technical guidance. The city and state administration can also play an important role in marketing, especially when it comes to renovations of existing buildings and new construction, by passing an ordinance that would require that every building in the district steam service area consider use of FDs as a prerequisite for obtaining a construction permit.
- The Hudson Fisonic Corporation, in close cooperation with Con Edison, will conduct regular seminars aimed at providing to potential customers, consulting engineers, and HVAC equipment vendors complete information concerning the benefits of FDs and recommending retrofit procedures and equipment.
- The involvement of Con Edison in the project and marketing activities will motivate the steam customers into ordering the FDs for their buildings. This will allow planning the manufacturing of the FDs in NYS by the Hudson Fisonic Corporation. The manufacturing of the FDs in the NYS can be started within 9 to 12 months after the completion of the demonstration project. After receiving customer orders for the FDs the manufacturing facility will be financed by the HFC and a bank loan.
- The results of the demonstration project will be widely disseminated to the steam customers in New York City and NYS by publications in technical magazines and presentation to the following organizations: Building Owners and Management Association (BOMA), Manufacturing Association and Industrial Development Agencies.

Section 1: Application of Fisonic Devices

Introduction

Consolidated Edison Company of New York (Con Edison) currently serves with electricity, gas, and district steam, about 1,800 large customers in Manhattan and owns three steam generating plants: 59th station, 74th station and the new East River combined cycle plant. Con Edison total annual electric sales are 47 million MWHR (revenues about $6.5 billion), gas sales are 117 million dekatherms (revenues about $1.6 billion) and district steam sales are 28 million Mlbs (revenues about $610 million).

The steam pressure delivered to the customers ranges between 150 to 400 psig. The customers use the steam for space heating, domestic hot water and cooling (through steam driven or absorption chillers). Many customers currently convert the district steam into hot water in tube and shell heat exchangers. The hot water is then distributed by electrically driven pumps throughout the building for space heating and domestic hot water service. The steam condensate is discharged into the city sewer system. The discharge of the condensate (maximum discharge flow rate reaches about 20,000 gpm during the peaking loads) consumes a substantial capacity of the sewer system and the sewer treatment facilities. In order to reduce the temperature of the discharged stream to 150°F the condensate is mixed with cold potable water, thus further aggravating the sewer system problems. The described system requires expensive heat exchangers (with associated maintenance cost), electrically driven pumps for hot water transport (with associated maintenance and electric cost) and substantial amounts of cold potable water. This situation results in high energy, water, and sewer charges to the customers and high make-up water cost for Con Edison steam generating plants.

The purpose of this project was the evaluation of the feasibility of the Fisonic Devices (FDs), which can improve the end-use energy efficiency and the environmental impact of the steam based customers. The FDs are supersonic, condensing heat pumps with the outlet pressure higher than the pressure of the working medium at the inlet of the FD. The FD heats the re-circulated building water by direct contact with steam and transports the water throughout the building, thus eliminating the tube and shell heat exchanger and the electrically driven pump. The use of the FD allows reducing the terminal temperature difference between steam and water, the required steam consumption and the amount of cold potable water. The use of the FD also allows reducing the steam consumption for the chillers. The FD can be used as a pump, direct contact heat exchanger and a de-aerator.

The project was sponsored by New York State Energy Research and Development Authority (NYSERDA), Con Edison, and Hudson Fisonic Corporation (involved in the development of the FDs). The project was performed by Joseph Technology Corporation, Inc. (JTC) in close cooperation with Con Edison district steam customers.

The potential market for use of FDs includes all buildings in NYS that convert steam into hot water. The economic motivation for the customers for buying the FD is the reduction in capital, maintenance and operating cost, and reduction in emission discharge to the environment. The principal differences between FDs and conventional Jet Apparatus are described below.

Jet Apparatus

The Jet Apparatus (JA) are widely used in various industries and include venturi desuperheaters, steam ejectors, jet exhausters and compressors, jet eductors and jet vacuum pumps. The JA consists of three principal parts: a converging (working) nozzle surrounded by a suction chamber, mixing nozzle and a diffuser (Figure 1-1). The working (motive) and injected (entrained) streams enter into the mixing nozzle where the velocities are equalized and the pressure of the mixture is increased. From the mixing nozzle the combined stream enters the diffuser where the pressure is further increased. The diffuser is so shaped that it gradually reduces the velocity and converts the energy to the discharge pressure with as little loss as possible. The JA transforms the kinetic energy of the working stream to the injected stream by direct contact without consumption of mechanical energy. The JAs operate with high expansion and moderate or high compression ratios.

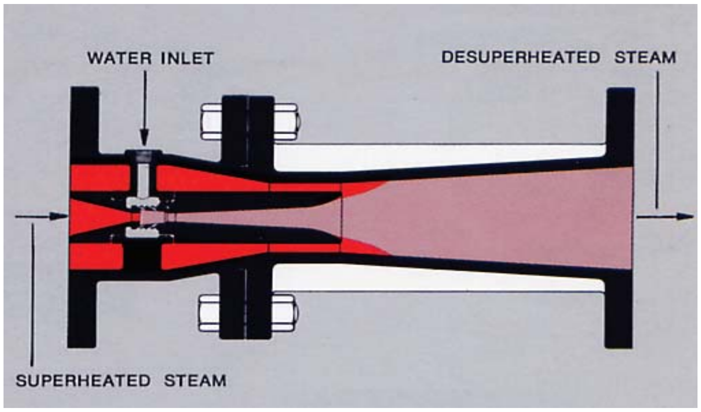


Figure 1-1. Typical Jet Steam Desuperheating Apparatus

In the JAs the interaction of two streams with various velocities provides an increase in entropy of the mixed stream takes place, as compared with an invertible mixing, resulting in the reduction of the pressure of the discharged stream. Therefore, typically the discharge pressure of the JA is higher than the pressure of the injected stream but lower than the pressure of the working stream.

Fisonic Devices

The Fisonic Devices (FDs) are pumps with patented optimized internal geometry. The injected water is typically supplied through a narrow circumferential channel surrounding the working nozzle (Figure 1-2). With this design, the injected water enters the mixing chamber with high velocity in parallel with the velocity of the working stream. The mixing chamber typically has a conical shape. The FDs operate with high expansion and small compression ratios. The discharge pressure in the FDs is typically higher than the pressure of the working and injected streams.

The optimized internal geometry of the FD causes the working and the injected streams to mix and accelerate, creating transonic conditions and converting the minute fractions of the streams thermal energy to physical trust (pump head) with the discharge pressure higher than the pressure of the mixing streams. The main reason behind this phenomenon is the high compressively of homogeneous two-phase flows. The sonic speed in such systems is much lower than the sonic speed in liquids and in gases. The important feature of the FD is also the independence of the discharge flow from the changing parameters of the customer system (like back pressure). Theoretical bases of FDs are described in the Appendix Section A-1.

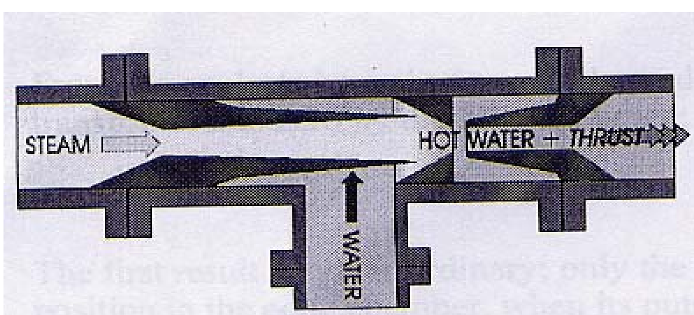


Figure 1-2. The Fisonic Device Diagram

Application of FDs for Space Heating

An example of FD use for a district steam heating application is presented in Figure 1-3. In this application, the FD replaces the surface type steam to hot water heat exchanger and the hot water electric driven circulating pump. The space heating requirements of the building are closely related to the outdoor temperature which varies, for example for the New York City, from -1°F to 60°F. To control the building space heat supply, the working steam flow rate of the FD should vary in accordance with changes of outdoor temperatures. At these conditions, because of the close relationship between the thermal and hydraulic modes of the FD, the discharge pressure and the water flow rate will vary. In order to keep the discharge water flow constant, an electric driven pump is installed, refer to Figure 1-3. The pump will operate for a limited number of hours during the year.

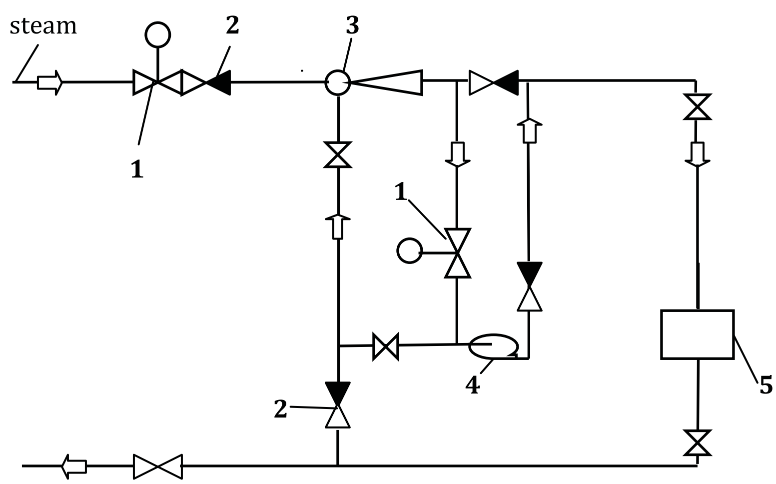


Figure 1-3. District Steam System with Fisonic Device

1 – Temperature Controller; 2 – Non-Return Valve; 3 – FD; 4 – Electric Driven Pump; 5 – District Heating Customer.

The FDs also can use hot water as a working medium. In this case, it is beneficial to increase the temperature difference between supply and return temperatures. A principal diagram of such a system is presented in Figure 1-4. In a central district system, the steam supplied from a power plant or by boilers heats the district return water. The heated water is supplied through a district heating system to the district customers. Each customer is equipped with a FD where the district supply water heats the building return water from 160°F to 190°F. About 20% of the total water flow rate circulates in the district heating system and 80% in the customer systems. The customer FDs not only heat the water but also operate as circulating pumps at the customer systems.

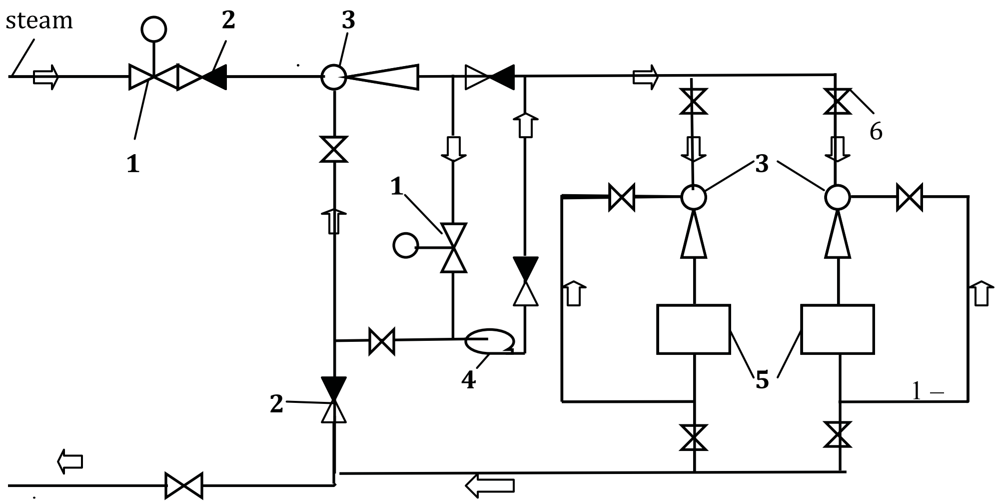


Figure 1-4. District Steam/Hot Water System with FDs

1 – Temperature Controller; 2 – Non-return Valve; 3 – FD; 4 – Pump; 5 – District Heating Customers; 6 – Valve.

Other applications of FDs are described in the Appendix Section A-5.

Control of Heat Supply from FDs

There are three following methods for control of the heat supply from the FD:

1. Constant water flow rate with variable temperature difference
2. Variable flow rate with constant temperature difference
3. Variable flow rate with variable temperature difference

The first control method is typically used. Figure 1-5 presents the relationship for this method between water supply and return temperatures and the working pressure of the FD. Figure 1-6 presents the relationship between the working steam flow rate and heat supply and the working pressure of the FD. From these figures one can see that the steam flow rate, heat load, and water temperatures have a linear dependence from the FD working steam pressure. With the reduction of the outdoor temperature the water supply and the return temperatures are increased. This is accompanied with the corresponding increase in the FD working steam flow rate. Typical dependence of the discharge water pressure on the outdoor temperature is presented in Figure 1-7.

Figure 1-5 Dependence of the Water Temperature on FD Working Steam Pressure

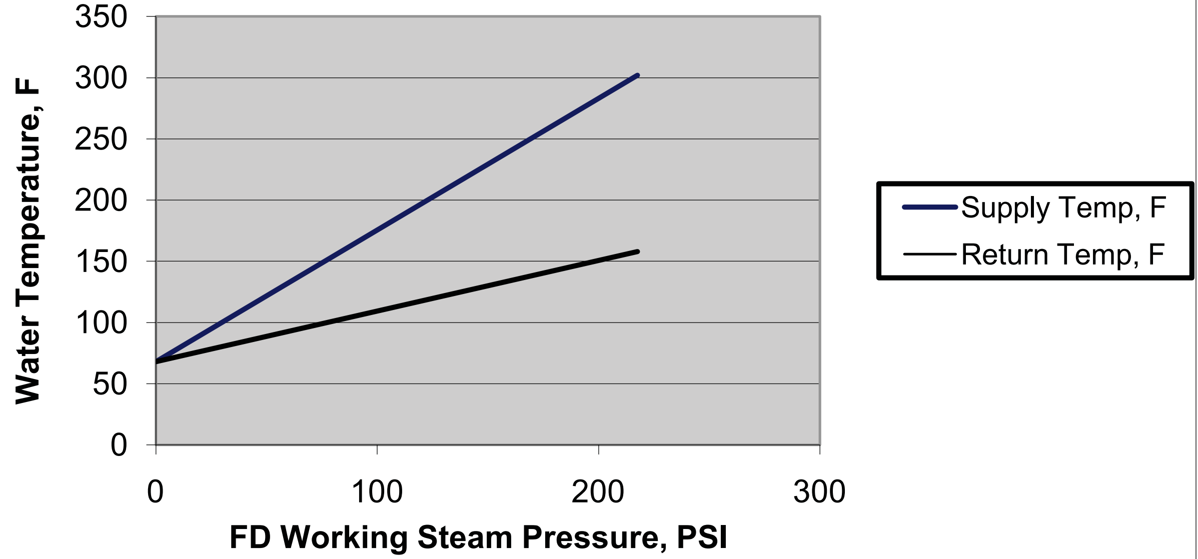
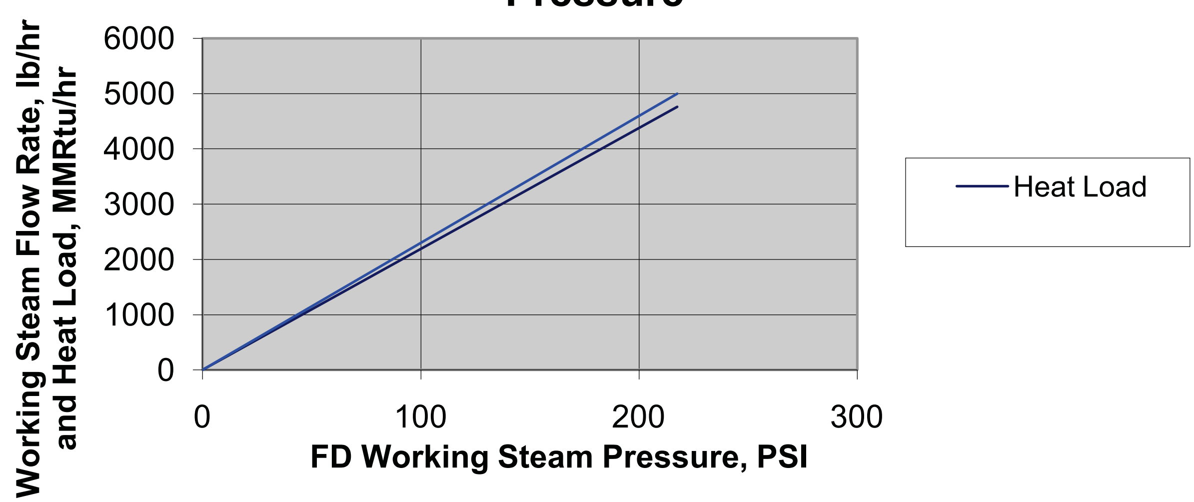
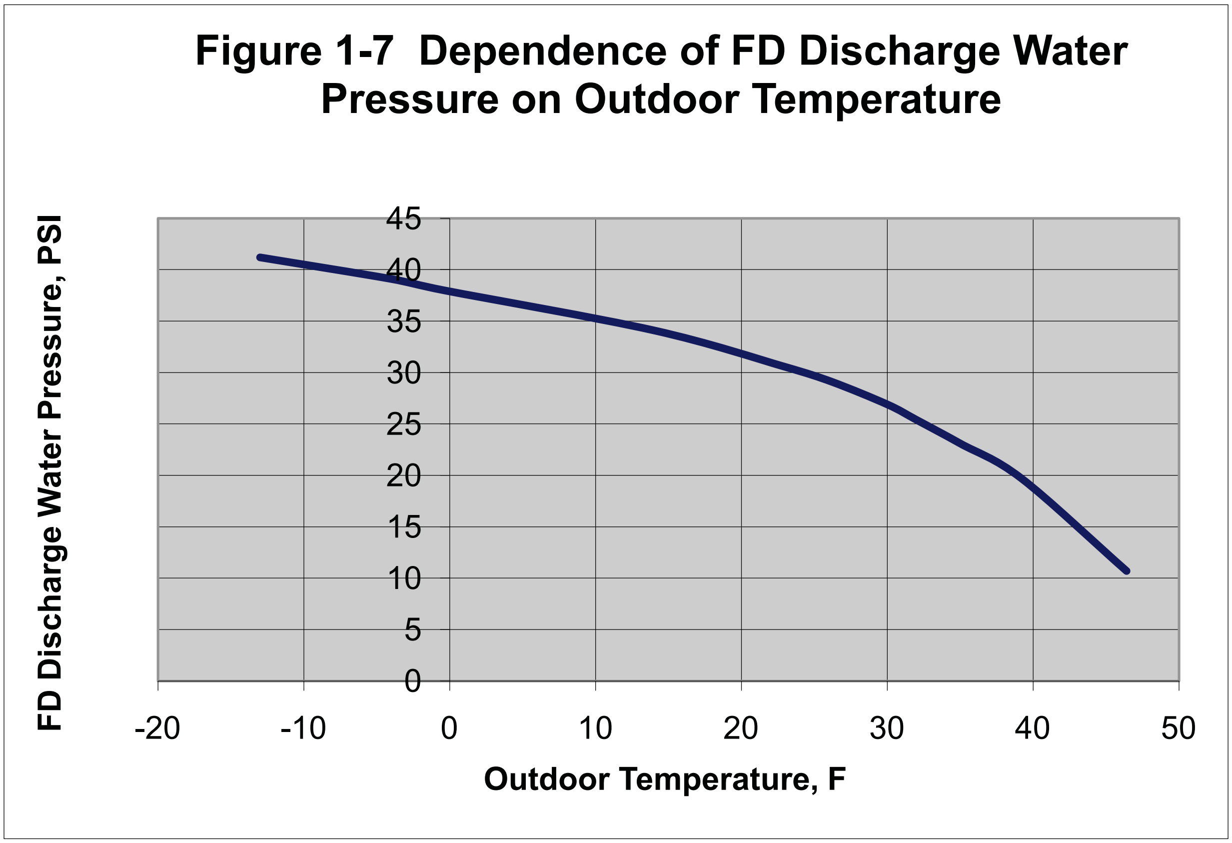


Figure 1-6 Dependence of Working Steam Flow Rate and Heat Load on FD Working Steam Pressure





The control diagram for a space heating system with a Fisonic Device is presented in Figure 1-8. This control system allows modulating the hot water supply temperature in accordance with the outdoor temperature as presented in Figure 1-9.

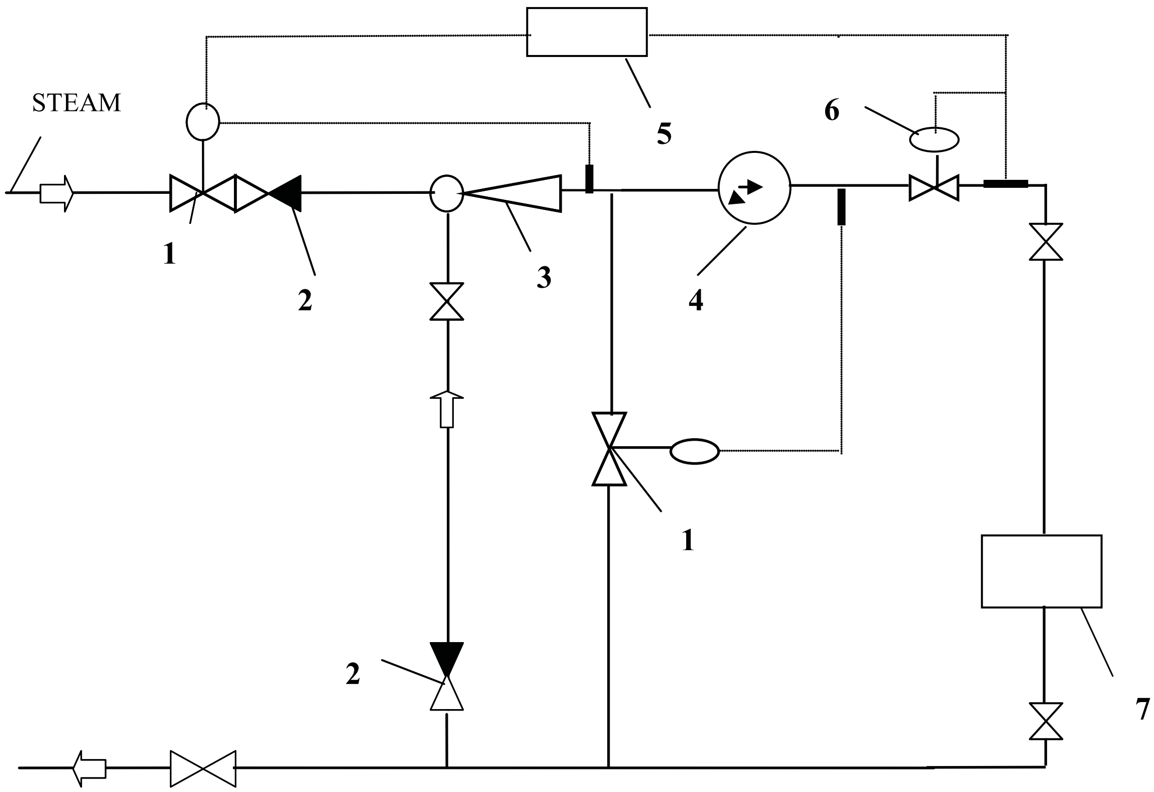
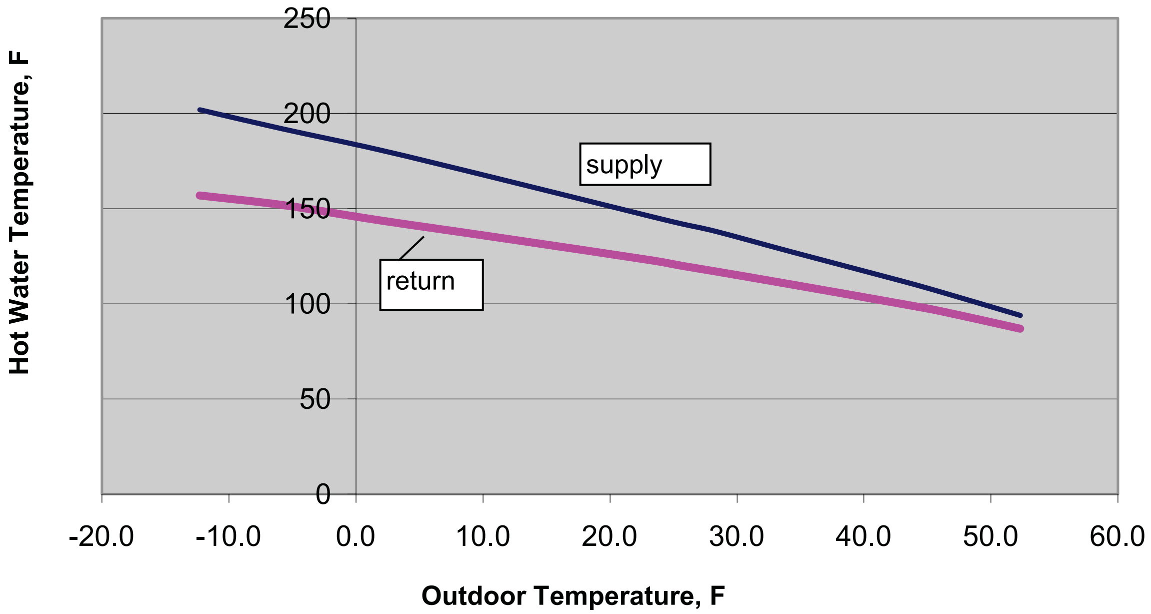


Figure 1-8 FD Space Heating Control Diagram

1 – Temperature Control Valve; 2 – Non Return Valve; 3 – Fisonic Device; 4 – Electric Driven Pump; 5 – Automation System; 6 – Pressure Control Valve; 7 – Space Heating Customer

Figure 1-9 Dependence of Water Supply and Return Temperatures from Outdoor Temperature



Section 2: Assessment of FDs for the Con Edison Headquarters Building

Description of the HVAC System and Heat Loads

The building is the Con Edison Headquarters, at 4 Irving Place, and occupies 27 stories with a total area of 1,170,000 ft². The building is supplied with steam from the Con Edison district steam system with a pressure of 170–200 psig. The steam pressure is reduced by pressure reducing valves to 3–5 psig for space heating and domestic hot water. The building is equipped with fifty-four large fan rooms with switchover steam and cooling coils. The perimeter space hot water radiation is supplied from two tube and shell heat exchangers by four electric driven pumps. The space hot water temperature is modulating from 90°F to 180°F. Two instantaneous heat exchangers supply the domestic hot water load.

The heating load of the building consists of a space heating hot water supply to the building perimeter radiators, space heating steam for ventilation, a direct steam supply to the air handler heating coils, domestic hot water and a direct steam supply to the cafeteria kitchen. An assessment of the building heating loads has been performed using the monthly steam consumption obtained from Con Edison. The methodology for this analysis is described in the Appendix Section A-2. The heat load analysis indicated that the peak heat load of the space heating hot water system as 3.76 MMBtu/hr and the ventilation peak heat load as 3.74 MMBtu/hr. The average DHW and cafeteria kitchen load is 3.93 MMBtu/hr. This information was used to size and select the heat transfer equipment.

The existing space heating system, which can be retrofitted by the FDs, is the hot water system supplying heat to the building periphery radiators. The DHW system cannot use the FDs because steam cannot be mixed with hot water for sanitary requirements. The ventilation and kitchen heat load requirements are satisfied with direct steam supply and also cannot use FDs.

Space Heating

Figure 2-1 presents the existing hot water system that currently supplies heat to the building periphery radiators. The system includes three steam pressure reducing valves (which reduce the district steam pressure from 170-200 psig to about 5 psig in stages), two tube and shell heat exchangers, and four hot water pumps. The condensate from the heat exchangers is collected in the dilution tank located close to the heat exchangers in the basement. Into the dilution tank, the condensate temperature is reduced by adding cold city water. From the dilution tank, the condensate/water mixture is pumped to the salvage water tank located at the 22nd floor. From the salvage tank, the water is supplied to the toilets. The balance of the water with a maximum temperature of 150°F is discharged into the sewer system. The condensate from the heat exchangers is not used to preheat the DHW. The DHW is heated in two separate tube and shell heat exchangers.

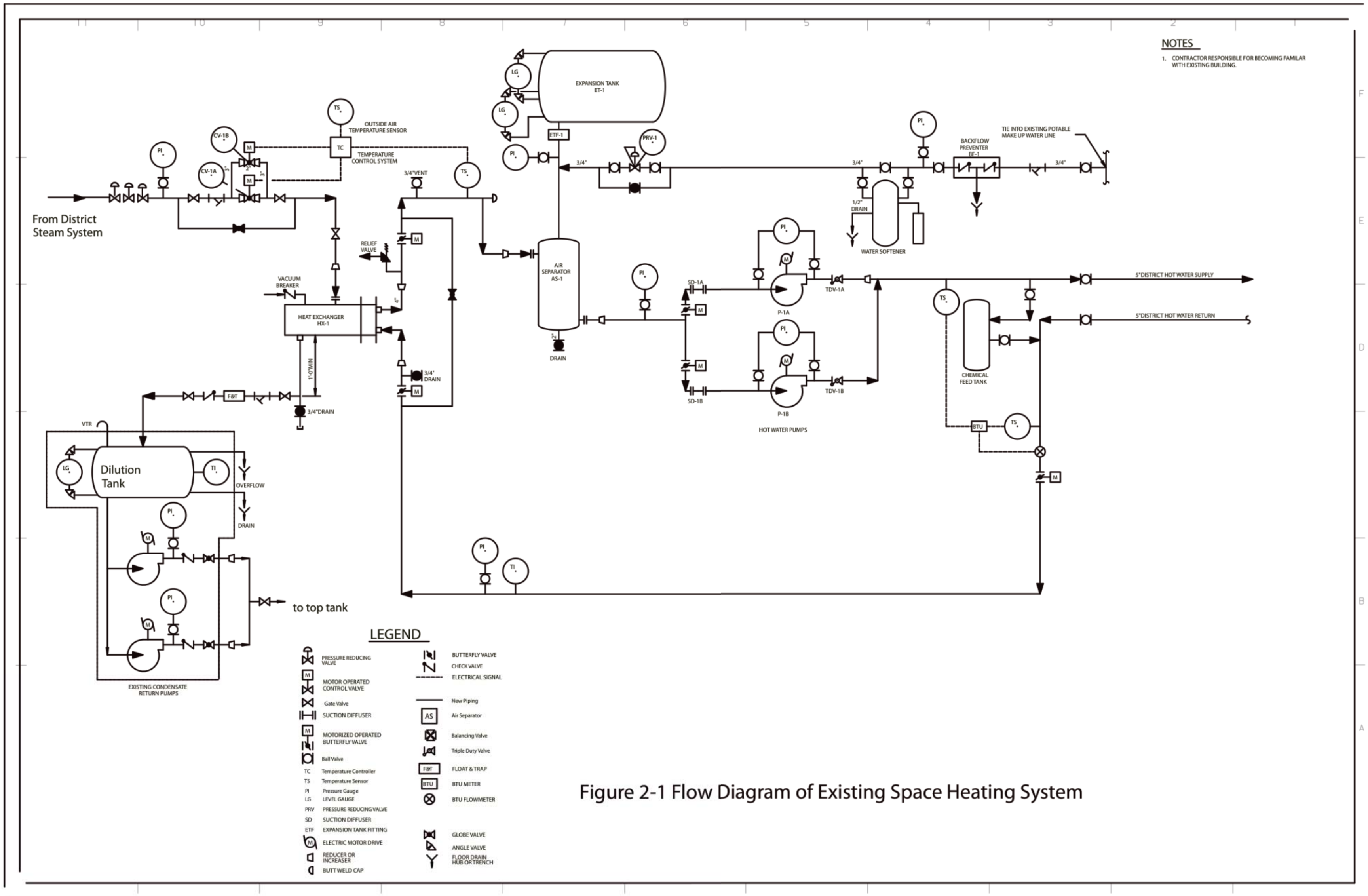


Figure 2-1 Flow Diagram of Existing Space Heating System

Figure 2-1 also presents standard hydraulic equipment (expansion tank and air separator) necessary for the reliable operation.

Figure 2-2 presents the retrofitted space hot water system equipped with the FDs. In this system, when compared with the existing system (Figure 2-1), the pressure reducing valves, heat exchangers, and the hot water circulating pumps are eliminated. This equipment is replaced with the FDs, which provide direct contact water heating with steam and water pumping. No steam pressure reduction valves are required. The district steam pressure is reduced by the FD. In order to keep the FD discharge water flow constant a single electric driven pump is installed. The pump will operate for a limited number of hours during the year. The hot water return temperature is reduced by heating the DHW in a plate and frame heat exchanger. Afterward, the return water is discharged to the existing dilution tank.

Capital Cost Estimates

Table 2-1 presents the capital cost estimates for the retrofit of existing system with FDs. The capital costs include the material and installation costs. The material costs were provided by HVAC equipment vendors and HFC, and the installation costs were supplied by the New York City mechanical contractors.

Table 2-1. Capital Cost of Existing System Retrofit with FDs

Items	Capital Cost, $
Fisonic Devices	21,000
Temperature Control Valve	9,000
Pressure Control Valve	5,330
Ball Valves	5,610
Nonreturn Valves	4,100
Piping,Small Valves,Supports, Insulation and Painting	13,910
Temperature and Pressure Sensors,Electric Power Supply,Connection to Graphic Screen	13,000
Subtotal	71,950
Engineering,Permitting,Drawings and Specification	28,780
Contingency	14,390
Total	115,120

O&M Cost Estimates

Table 2-2 presents the annual operational parameters of the existing and the FD space heating system. The consumption and peak load data in Table 2-2 were obtained from the heat load calculations. The steam parameters were recorded during the site survey. The make-up water flow rate was estimated from the heat balance calculation in order to provide the temperature of the total flow of condensate and make-up water to be discharged to the sewer at 150°F. The hot water peak flow rate was estimated from the water and steam heat balances.

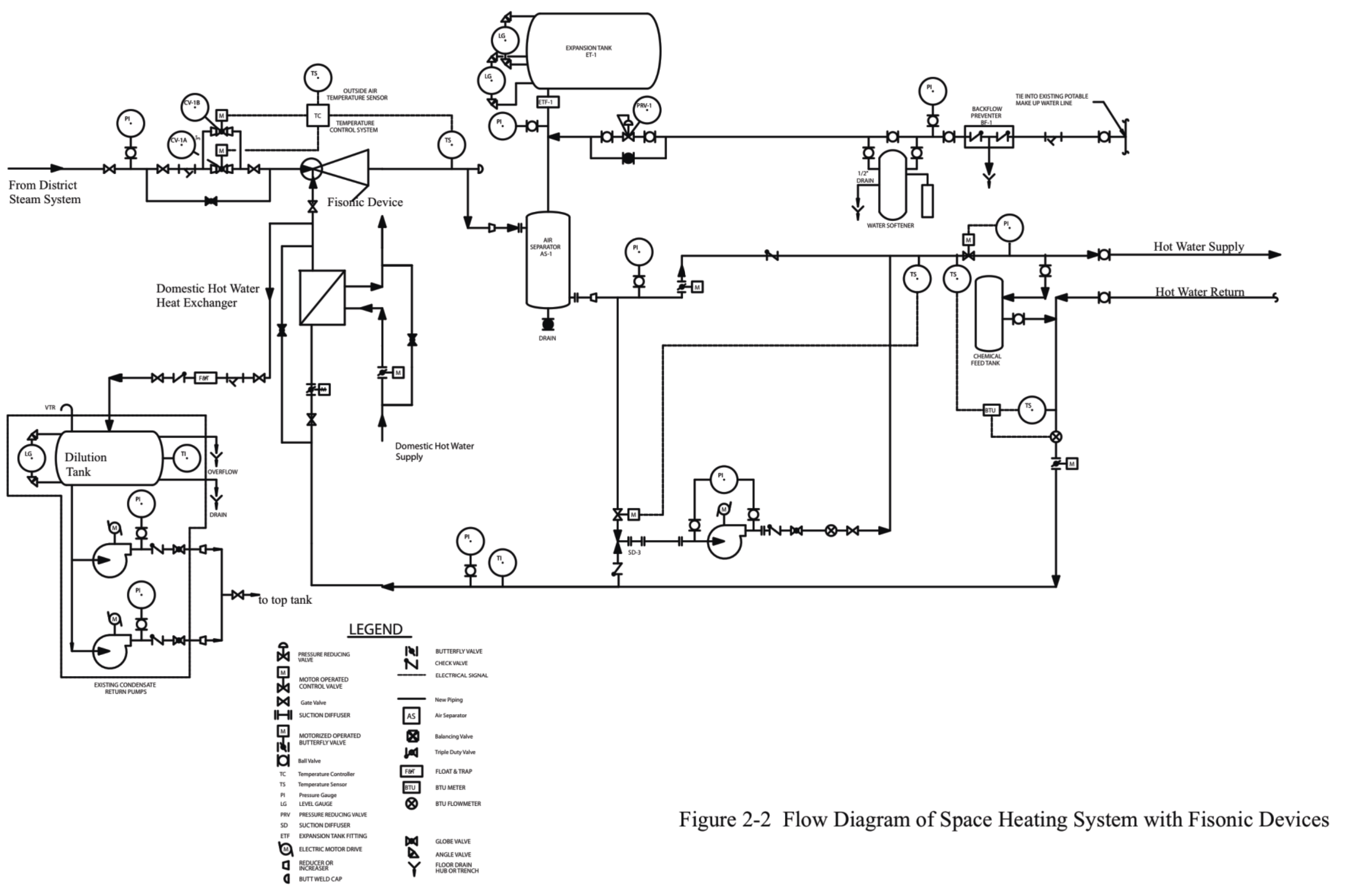


Figure 2-2 Flow Diagram of Space Heating System with Fisonic Devices

Table 2-2. Annual Operational Parameters

PARAMETERS	UNITS	Existing System	System with FDs
Annual System Heat Consumption	MMBtu	7,743	7,743
Annual Steam Consumption	Mlbs	7,722	6,458
Initial Steam Enthalpy	Btu/lb	1,199	1,199
Heat Exchanger Steam Enthalpy	Btu/lb	196	
Peak Comfort Heat Load	MMBtu/hr	3.86	3.86
Peak Steam Load	lb/hr	3,850	3,219
Hot Water Supply Temperature	F	180	180
Hot Water Return Temperature	F	160	160
Hot Water Peak Flowrate	gpm	386	386
Annual Condensate Flow	Mlbs	7,722	6,458
Annual Make-up Water Flow	Mlbs	6,692	718
Annual Make-up Water Flow	MGal	802	86
Total Condensate and Make-up Flow	Mlbs	14,414	7,176
Total Condensate and Make-up Flow	MGal	1,728	860
Annual Electric Cons. for Hot Water Pumps	kWh	62,827	12,565
Extra Building Space Requirement	Sq.Ft	200	

As one can see from Table 2-2, the annual steam consumption for the system with FDs is reduced by 16.4% when compared with the existing system (6,458 Mlbs vs. 7,722 Mlbs). This is the result of the direct contact steam condensation in the FD and condensate mixing with the hot water. The make-up water flow rate and the total water flow rate to be discharged to the sewer for the FD system are also substantially reduced (860 Mgal. vs. 1,728 Mgal). It is estimated that for the FD system the electric pump will operate about 20% of the existing system time.

Table 2-3 presents the potential O&M savings of the FD system in comparison with the existing system. The following unit costs, obtained during the building survey, were used: average steam unit cost: \$23.11/Mlb; average make-up water (city water) unit cost: \$2.67/Mgal; average sewer unit cost: \$4.28/Mgal; average electricity unit cost: \$0.15/kWh; annual cost of building space: \$50/sq.ft; maintenance cost for the existing and the FD's systems: 3% of capital cost.

Table 2-3. O&M Cost Comparison

PARAMETERS	Existing System		System with FDs		Savings, $/yr
	Cons.	Cost, $/yr	Cons.	Cost, $/yr	
Annual Steam Consumption, Mlbs	7,722	178,455	6,458	149,244	29,211
Annual Make-up Water Flow, Mcu.ft	107	2,140	11	220	1,920
Total Condensate and Make-up Flow, Mgal	1,728	7,397	860	3,682	3,715
Annual Electric Cons. for Hot Water Pumps, kWh	62,827	9,424	12,565	1,790	7,634
Extra Building Space Requirement, sq.ft	200	10,000			10,000
Maintenance		11,483		7,034	4,449
Total		218,900		161,971	56,928

The analysis indicates that the simple payback for the building retrofit with FDs is \$115,120/\$56,928 = 2 years.

Table 2-4 presents the capital cost comparison between a conventional space heating system with heat exchangers and a system with FDs for a new construction alternative. As one can see from Table 2-4 the capital cost of the space heating system with FDs is about 32% less than for a conventional system.

Table 2-4. New Conventional and Fisonic Space Heating Systems Total Capital Cost Estimates

Items	Size	Total Material and Installation Cost			
		Conven. System		System with FDs	
		Qty	$	Qty	$
Shell and Tube Heat Exchanger	4,000 MBH	2	26,600		
Fisonic Devices	4,000 MBH			2	21,000
Steam Pressure Reduction Valves		3	15,980		
Temperature Control Valves	4"	1	9,000	2	18,000
Water Pressure Control Valve	4"			1	5,330
Air Separator	1000 gpm	1	5,700	1	5,700
Expansion tank	1300 gal	1	8,100	1	8,100
Hot Water Circulating Pumps	50 HP	2	27,000	1	13,500
Hot Water Circulating Pumps	40 HP	2	24,000		
Relief Valve	3 "	1	900	1	900
Triple Duty Valves	4 "	4	3,000	1	750
Airtrol Fitting		1	720	1	720
Water Make-up Pressure Reducing Valve	2"	1	340	1	340
Suction Diffusers	4"	4	1,560	1	390
Ball Valves	4"	5	5,610	5	5,610
Gate Valves	4"	4	4,300	4	4,300
Chemical Feed Tank		1	1,370	1	1,370
Water Softener		1	5,700	1	5,700
Nonreturn Valves	4"			4	4,100
Motorized Butterfly Valves	4"	5	11,250	5	11,250
Piping, Small Valves, Supports, Insulation and Painting			44,300		27,820
Temperature and Pressure Sensors,Electric Power Supply,Connection to Graphic Screen			43,800		26,000
Subtotal			239,230		160,880
Engineering,Permitting,Drawings and Specification Preparation,Equipment Procurement,Construction Management			95,692		64,352
Contingency			47,846		32,176
Total			382,767		257,408

Section 3: Assessment of FDs for the Woolworth Building

Description of the HVAC System and Heat Loads

The Woolworth office building located at the 233 Broadway occupies 56 floors; 28 floors are now in operation. The building area is 900,000 ft². The building is supplied with steam from the Con Edison district steam system with a pressure of 170–180 psig. The steam pressure is reduced by two pressure reducing valves to 3 psig for space heating and domestic hot water. The cooling is provided by two 400 ton, two 300 ton electric driven chillers and distributed 80 ton units. The perimeter space hot water radiation is supplied from four tube and shell heat exchangers by four electric driven pumps. The space hot water temperature is modulating from 90°F to 135°F. Four instantaneous heat exchangers supply the domestic hot water load. The condensate from the heat exchangers is used to preheat the DHW and Afterward mixed with cold water and discharged into the sewer system.

The existing space heating system, which can be replaced by the FDs, is the hot water system supplying heat to the building periphery baseboards and the air conditioning hot water coils. The DHW system cannot use the FDs because steam cannot be mixed with hot water for sanitary requirements.

An economic comparison of the existing heating system of the building with the proposed heating system using FDs requires the determination of all components of the heating load. The heating load of the building consists of space heating hot water supply to the building perimeter radiators, space heating steam for ventilation, direct steam supply to the air handler heating coils, domestic hot water, and direct steam supply to the cafeteria kitchen. An assessment of the building heating loads has been performed using the monthly steam consumption obtained from Con Edison. The methodology for this analysis is described in the Appendix Section A-3.

The analysis indicated that the peak heat load of the space heating hot water system as 7.53 MMBtu/hr and the ventilation peak load as 2.85 MMBtu/hr. The total space heating load is 10.38 MMBtu/hr. The average DHW load is 1.05 MMBtu/hr. This information will be used to size and select the heat transfer equipment.

Capital Cost Estimates

The existing space heating system, which can be retrofitted by the FDs, is the hot water system supplying heat to the building periphery baseboards and the air conditioning hot water coils. The DHW system cannot use the FDs because steam cannot be mixed with hot water for sanitary requirements. The schematic diagram of the existing space heating system is similar to Figure 2-1 included in Section 2.

In the retrofitted space hot water system equipped with the FDs, when compared with the existing system, the pressure reducing valves, heat exchangers, and the hot water circulating pumps are eliminated (See Figure 2-2 in Section 2). This equipment is replaced with the FDs that provide direct contact water heating with steam and water pumping. No steam pressure reduction valves are required. The district steam pressure is reduced by the FD. In order to keep the FD discharge water flow constant a single electric driven pump is installed. The pump will operate for a limited number of hours during the year. The hot water return temperature is reduced by heating the DHW in a plate and frame heat exchanger. Afterward, the return water is discharged to the sewer.

Table 3-1 presents the capital cost estimates for the retrofit of the existing system with FDs. The capital cost includes the material and installation costs. The material costs were provided by HVAC equipment vendors and the installation costs were supplied by the New York City mechanical contractors.

Table 3-1. Capital Cost of Existing System Retrofit with FDs

Items	Capital Cost, $
Fisonic Devices	42,000
Temperature Control Valve	9,000
Pressure Control Valve	5,330
Ball Valves	5,610
Nonreturn Valves	4,100
Piping,Small Valves,Supports, Insulation and Painting	13,910
Temperature and Pressure Sensors,Electric Power Supply,Connection to Graphic Screen	13,000
Subtotal	92,950
Engineering,Permitting,Drawings and Specification Preparation,Equipment Procurement,Construction Mgnt	37,180
Contingency	18,590
Total	148,720

O&M Estimates

Table 3-2 presents the annual operational parameters of the existing and the FD space heating system. The consumption and peak load data in Table 3-2 were obtained from the heat load calculations. The steam parameters were recorded during the site survey. The make-up water flow rate was estimated from the heat balance calculation in order to provide the temperature of the total flow of condensate and make-up water to be discharged to the sewer at 150°F. The hot water peak flow rate was estimated from the water and steam heat balances.

As one can see from Table 3-2 the annual steam consumption for the system with FDs is reduced by 16.4% when compared with the existing system (14,685 Mlbs vs. 17,560 Mlbs). This is the result of the direct contact steam condensation in the FD and condensate mixing with the hot water. The make-up water flow rate and the total water flow rate to be discharged to the sewer for the FD system are also substantially reduced (1,956 Mgal. vs. 3,930 Mgal). It is estimated that for FD system the electric pump will operate about 20% of the existing system time.

Table 3-2. Annual Operational Parameters

PARAMETERS	UNITS	Existing System	System with FDs
Annual System Heat Consumption	MMBtu	17,607	17,607
Annual Steam Consumption	Mlbs	17,560	14,685
Initial Steam Enthalpy	Btu/lb	1,199	1,199
Heat Exchanger Steam Enthalpy	Btu/lb	196	
Peak Comfort Heat Load	MMBtu/hr	10.38	10.38
Peak Steam Load	lb/hr	10,352	8,657
Hot Water Supply Temperature	F	180	180
Hot Water Return Temperature	F	160	160
Hot Water Peak Flowrate	gpm	1,038	1,038
Annual Condensate Flow	Mlbs	17,560	14,685
Annual Make-up Water Flow	Mlbs	15,219	1,632
Annual Make-up Water Flow	MGal	1,825	196
Total Condensate and Make-up Flow	Mlbs	32,779	16,317
Total Condensate and Make-up Flow	MGal	3,930	1,956
Annual Electric Cons. for Hot Water Pumps	kWh	126,711	25,342
Extra Building Space Requirement	Sq.Ft	200	

Table 3-3 presents the potential O&M savings of the FD system in comparison with the existing system. The following unit costs (obtained during the building survey) were used: average steam unit cost: \$30.9/Mlb; average make-up water (city water) unit cost: \$2.67/Mgal; average sewer unit cost: \$4.28/Mgal; average electricity unit cost: \$0.15/kWh; annual cost of building space: \$50/sq.ft; maintenance cost for the existing and the FD systems: 3% of capital cost.

Table 3-3. O&M Cost Comparison

PARAMETERS	Existing System		System with FDs		Savings, \$/yr
	Cons.	Cost, \$/yr	Cons.	Cost, \$/yr	
Annual Steam Consumption, Mlbs	17,560	542,604	14,685	453,767	88,838
Annual Make-up Water Flow, Mgal	1,825	4,872	196	522	4,350
Total Condensate and Make-up Flow, Mgal	3,930	16,821	1,956	8,373	8,448
Annual Electric Cons. for Hot Water Pumps, kWh	126,711	19,007	25,342	1,790	17,216
Extra Building Space Requirement, sq.ft	200	10,000			10,000
Maintenance		13,728		9,054	4,674
Total		607,032		473,507	133,525

As one can see from Table 3-3 the O&M cost of the space heating system with FDs is about 22% less than for the existing system. The analysis indicates that the simple payback for the building retrofit with FDs is \$148,720/\$133,525=1.2 years.

Table 3-4 presents the capital cost comparison between a conventional space heating system with heat exchangers and a system with FDs for a new construction alternative. As one can see from Table 3-4 the capital cost of the space heating system with FDs is about 34% less than for a conventional system.

Table 3-4. New Conventional and Fisonic Space Heating Systems Capital Cost Estimate

Items	Size	Total Material and Installation Cost			
		Conven. System		Fisonic System	
		Qty	$	Qty	$
Shell and Tube Heat Exchanger	4,000 MBH	4	53,200		
Fisonic Devices	4,000 MBH			4	42,000
Steam Pressure Reduction Valves		2	10,653		
Temperature Control Valves	4"	1	9,000	2	18,000
Pressure Control Valve	4"			1	5,330
Air Separator	1000 gpm	1	5,700	1	5,700
Expansion tank	1300 gal	1	8,100	1	8,100
Hot Water Circulating Pumps	75 HP	2	40,500	1	20,250
Hot Water Circulating Pumps	60 HP	2	36,000		
Relief Valve	3 "	1	900	1	900
Triple Duty Valves	4 "	4	3,000	1	750
Airtrol Fitting		1	720	1	720
Water Make-up Pressure Reducing Valve	2"	1	340	1	340
Suction Diffusers	4"	4	1,560	1	390
Ball Valves	4"	5	5,610	5	5,610
Gate Valves	4"	4	4,300	4	4,300
Chemical Feed Tank		1	1,370	1	1,370
Water Softener		1	5,700	1	5,700
Nonreturn Valves	4"			4	4,100
Motorized Butterfly Valves	4"	5	11,250	5	11,250
Piping, Small Valves, Supports, Insulation and Painting			44,300		27,820
Temperature and Pressure Sensors,Electric Power Supply,Connection to Graphic Screen			43,800		26,000
Subtotal			286,003		188,630
Engineering,Permitting,Drawings and Specification Preparation,Equipment Procurement,Construction Management			114,401		75,452
Contingency			57,201		37,726
Total			457,605		301,808

Section 4: Conversion of the Louis Lefkowitz State Office Building to Hot Water with FDs

Description of the HVAC System and Heat Loads

The Louis Lefkowitz State Office Building, located at the 80 Center Street, was constructed in 1921, and has nine floors, and a total area of 400,000 ft^2. The building is in operation for 24 hours without setback control and the indoor temperature is maintained between 72 and 74°F. The Con Edison district steam pressure at the entrance of the building is 170–180 psi. The district steam pressure is reduced in two pressure reducing stations to about 3-to-5 psig and used to supply the steam radiators throughout the building. The radiators are not equipped with any thermostatic control valves. The temperature in the rooms is controlled by the window air conditioners. The steam also heats the domestic hot water in a tube and shell heat exchanger. The cooling is provided with window air conditioners.

It is proposed to retrofit the building space heating system from steam to hot water and use FDs for production of hot water. The advantages and methodology of conversion of the building from steam to hot water are presented in Appendix Section A-4.

An economic comparison of the existing heating system of the building with the proposed heating system using FDs requires the determination of the heating load. An assessment of the building heating load has been performed using the monthly steam consumption obtained from Con Edison. The methodology for this analysis is described in the Appendix Section A-4. The analysis indicated that the total space heating load of the building is 8.24 MMBtu/hr and the average DHW load is 0.11 MMBtu/hr. This information will be used to size and select the heat transfer equipment.

Capital Cost Estimates

Table 4-1 presents the capital cost estimates for conversion of the building space heating system to hot water with Fisonic Devices. All the existing steam and condensate piping and radiators remain in place. The conversion includes installation of thermostatic control valves and vents on each radiator, removal of the trap internals, installation of an expansion tank, water softener, temperature and pressure control valves, fisonic devices, and a water circulating pump for limited operation. The capital cost presented in Table 4-1 includes the material and installation costs. The material cost was provided by HVAC equipment vendors and the installation cost was supplied by the New York City mechanical contractors.

Table 4-1. Capital Cost of Existing System Retrofit to Hot Water with FDs

Items	Size	Qty	Cost, $
Fisonic Devices	5000 MBH	2	24,000
Temperature Control Valve	4"	2	18,000
Pressure Control Valve	4"	1	5,330
Air Separator	1000 gpm	1	5,700
Expansion tank	1300 gal	1	8,100
Hot Water Circulating Pump	50 HP	1	13,500
Relief Valve	3 "	1	900
Triple Duty Valves	4 "	1	750
Airtrol Fitting		1	720
Water Make-up Pressure Reducing Valve	2"	1	340
Suction Diffusers	4"	1	390
Ball Valves	4"	5	5,610
Gate Valves	4"	4	4,300
Chemical Feed Tank		1	1,370
Water Softener		1	5,700
Nonreturn Valves	4"	4	4,100
Motorized Butterfly Valves	4"	5	11,250
Thermostatic Control Valves and Vents	3/4"	1,900	266,000
Trap removal,Piping, Small Valves, Supports, Insulation and Painting			62,020
Temperature and Pressure Sensors,Electric Power Supply,Connection to Graphic Screen			55,188
Subtotal			493,268
Engineering,Permitting,Drawings and Specification Preparation,Equipment Procurement,Construction Management			123,317
Contingency			73,990
Total			690,575

O&M Cost Estimates

Table 4-2 presents the annual operational parameters of the conventional and the FD space heating system. The consumption and peak load data in Table 4-2 were obtained from the heat load calculations. The make-up water flow rate was estimated from the heat balance calculation in order to provide the temperature of the total flow of condensate and make-up water to be discharged to the sewer at 150°F. The hot water peak flow rate was estimated from the water and steam heat balances.

As one can see from Table 4-2, the annual steam consumption for the system with FDs is reduced by about 17% when compared with the existing system. The make-up water flow rate and the total water flow rate to be discharged to the sewer for the FD system are also substantially reduced by about 50%. It is estimated that for FD system the electric pump will operate about 20% of the existing system time.

Table 4-3 presents the potential O&M savings of the FD system in comparison with the conventional system. The following unit costs (obtained during the building survey) were used: average steam unit cost: $31.33/Mlb, average make-up water (city water) unit cost: $2.67/Mgal; average sewer unit cost: $4.28/Mgal; average electricity unit cost: $0.15/kWh; annual cost of building space: $50/sq.ft; maintenance cost for the existing and for the FD’s systems: 3% of capital cost.

Table 4-2. Annual Operating Parameters

PARAMETERS	UNITS	Conv Syst	Syst with FD's
Annual System Heat Consumption	MMBtu	17,991	17,991
Annual Steam Consumption	Mlbs	17,943	14,892
Initial Steam Enthalpy	Btu/lb	1,199	1,199
Heat Exchanger Steam Enthalpy	Btu/lb	196	
Peak Comfort Heat Load	MMBtu/hr	8.29	8.29
Peak Steam Load	lb/hr	8,268	6,862
Hot Water Supply Temperature	F	180	180
Hot Water Return Temperature	F	160	160
Hot Water Peak Flowrate	gpm	829	829
Annual Condensate Flow	Mlbs	17,943	14,892
Annual Make-up Water Flow	Mlbs	15,550	1,655
Annual Make-up Water Flow	Mgal	1,860	192
Total Condensate and Make-up Flow	Mlbs	33,493	16,547
Total Condensate and Make-up Flow	Mgal	4,015	1,984
Annual Electric Cons. for Hot Water P	kWh	134,931	26,986
Extra Building Space Requirement	Sq.Ft	200	

Table 4-3. O&M Cost Comparison

PARAMETERS	Existing System		System with FD'S		Savings, $/yr
	Cons.	Cost, $/yr	Cons.	Cost, $/yr	
Annual Steam Consumption,Mlbs	17,943	562,140	14,892	466,576	95,564
Annual Make-up Water Flow, Mgal	1,860	4,966	192	512	4,454
Total Condensate and Make-up Flow, Mgal	4,015	17,184	1,984	8,489	8,694
Annual Electric Cons. for Hot Water Pumps,kWh	134,931	20,240	26,986	4,048	16,192
Extra Building Space Requirement, sq.ft	200	10,000			10,000
Maintenance		41,343		20,834	20,509
Total		**655,872**		**500,460**	**155,413**

As one can see from Tables 4-1 and 4-3 the simple payback period of conversion of the building to hot water with FDs is $690,575/$155,413= 4.5 years.

Section 5: Conclusions and Plan of Action for Future Work of FDs

Study Conclusions

- The study has demonstrated that the use of FDs for Con Edison customers will reduce their annual steam consumption by about 16.4% when compared with the existing system.
- The make-up water flow rate and the total water flow rate to be discharged to the sewer for the FD system are reduced by about 50%.
- The simple payback period for retrofitting an existing hot water space heating system with FDs is under two years.
- The simple payback period for conversion of a steam heated building to hot water with FDs is about 4.5 years.

Potential Benefits of FDs

The potential benefits of use of FDs for Con Edison customers are estimated as follows:

- **Energy Savings**: assuming the implementation of Fisonic Devices by 30% of the Con Edison customers (540 buildings with current steam consumption of about 8.4 million Mlbs per year), the potential reduction in steam consumption of 16.4% will result in the following cost savings to the customers: 8.4 million Mlbs x 0.164 x $31/Mlb (current average cost of steam to the customers) = $42.7 million. The potential savings of the customers in electric consumption are: 8.4 million Mlbs x $0.80/Mlbs (estimated electric cost) = $6.7 million.
- **Water and Sewer Savings**: for the above assumptions the savings associated with cold water consumption and sewer discharge will be 8.4 million Mlbs x $0.48/Mlbs (estimated water and sewer cost) =$4 million.
- **Job Creation**: using 32 job years per $1 million of energy savings (EPRI, "Guidelines for Assessing the Feasibility of District Energy Projects," Ref.11) it is estimated that project implementation will result in creation of 1,580 job years.
- **Environmental Benefits**: using the following emission reduction factors in lbs/yr/Mlb (Ref.11): NO_x –10.74 x 10^{-3}; Particulates – 4.3 x 10^{-3}; VOC – 1.5 x 10^{-3}; CO_2 -60; the environmental benefits are estimated as the following pollution reductions in lbs/year: NO_x – 15,000; Particulates – 5,900; VOC – 2,100; and CO_2 –37,600 ton.

The potential benefits of use of FDs for the State of New York customers are estimated as follows:

> The annual primary energy consumption in NYS for buildings and industry is about 1,300 TBtu (NYSERDA Patterns and Trends, December 2002). Assuming that 5% of this energy is supplied by steam with conventional heat exchangers and electric-driven pumps, the current steam consumption can be estimated as 65 million Mlbs/year. Applying the above described methodology for estimating the energy savings with the use of FDs, the potential benefits for the NYS are estimated as follows: **Energy Savings**: $380 million; **Water and Sewer discharge savings**: $31 million; **Environmental Pollution Reductions**: NO_x – 116,000 lbs/yr, Particulates – 45,600 lbs/yr, VOC – 16,200 lbs/yr and CO_2 – 290,600 ton/yr.

Future Work

In order to start the FD commercialization the following actions are planned:

- Demonstrate the operation of FDs at Con Edison Headquarters and the Woolworth Building in New York City. The project sponsors have a high interest in a fast track demonstration project implementation.
- Taking into account the substantial potential energy savings and reduction of condensate and potable water discharge in the city sewer system, HFC will market the FDs to the Con Edison customers, city and state administration, consulting engineers and HVAC equipment vendors. All entities must be comfortable with the concept of using FDs, and all must be convinced of the cost effectiveness and benefits that it will provide.
- Many existing customers are in the process of replacing or upgrading an existing HVAC system, so local engineers and HVAC contractors are in an ideal position to market FDs by providing technical guidance. The city and state administration can also play an important role in marketing, especially when it comes to renovations of existing buildings and new construction, by passing an ordinance that would require that every building in the district steam service area considers use of FDs as a prerequisite for obtaining a construction permit.
- The HFC in close cooperation with Con Edison will conduct regular seminars aimed at providing to potential customers, consulting engineers, and HVAC equipment vendors, complete information concerning the benefits of FDs and recommending retrofit procedures and equipment.
- The involvement of Con Edison in the project and marketing activities will motivate the steam customers into ordering the FDs for their buildings. This will allow planning for the manufacturing of the FDs in NYS by the HFC. The manufacturing of the FDs in NYS can be started within 9-to-12 months after the completion of the demonstration project. After receiving customer orders for the FDs, the manufacturing facility will be financed by the HFC and a bank loan.
- The results of the demonstration project will be widely disseminated to the steam customers in New York City and NYS by publications in technical magazines and presentation to the following organizations: Building Owners and Management Association (BOMA), Manufacturing Association, and Industrial Development Agencies.

Section 6: References

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Appendix

Section A-1: Theoretical Basis of Fisonic Devices

The Fisonic Devices (FDs) are pumps with patented optimized internal geometry. The injected water is typically supplied through a narrow circumferential channel surrounding the working nozzle In this design the injected water enters the mixing chamber with high velocity in parallel with the velocity of the working stream. The mixing chamber has typically a conical shape. The FDs operate with high expansion and small compression ratios. The discharge pressure in the FDs is typically higher than the pressure of the working and injected streams.

The optimized internal geometry of the FD causes the working and the injected streams to mix and accelerate, creating transonic conditions and converting the minute fractions of the stream's thermal energy to physical trust (pump head) with the discharge pressure higher than the pressure of the mixing streams. The main reason behind this phenomenon is the high compressivity of homogeneous two-phase flows. The sonic speed in such systems is much lower than the sonic speed in liquids and in gases. As one can see from Figure A-1.1 the minimum sonic velocity takes place at the volumetric ratio of the streams of 0.5.The important feature of the FD is also the independence of the discharge flow from the changing parameters of the customer system (such as back pressure).

Water, S, m/s Gas, S, m/s

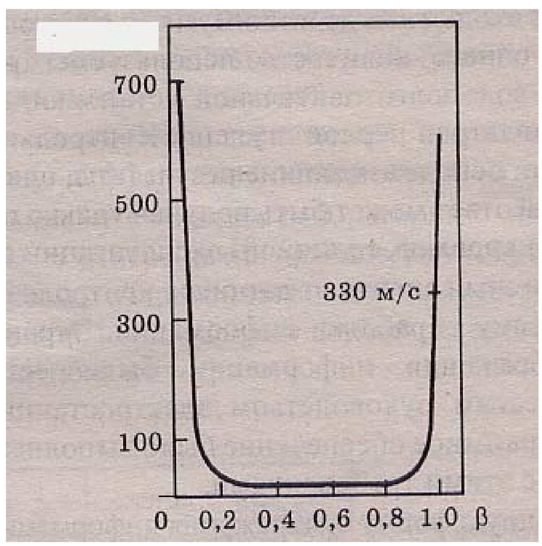


Figure A-1.1. The Dependance of Sonic Speed from Volumetric Ratio of Streams

Where:

S= sonic velocity, m/s.

β = ratio of volumetric gas to liquid plus gas composition ($\beta = \frac{V_g}{V_g + V_{ld}}$).

From Figure A-1 one can see that when there is no liquid–the ratio β equals one, if there is no gas–the ratio β equals zero. When there is 50% liquid and 50% gas (two phase flow) – the ratio β is equal 0.5 and the sonic velocity is less than 20 m/s (much lower than in gases and liquids).

The equation of sonic speed is as follows:

$$S^2 = \frac{kP}{\rho} \quad \text{………………………………………A-1.1}$$

Where:

k = isentropic exponent, equal to the ratio of specific heats; P = pressure; ρ = density of the medium.

For determining the isentropic exponent, Prof. Fisenco developed the following equation:

$$\frac{k_g(k+1)-2k}{k_g-1} = k\,\beta\left[1+\left(\frac{1}{\beta}-1\right)^2\right]-2\left(\frac{1}{\beta}-1\right)\left(\frac{1}{\varepsilon}-1\right) \quad \text{……………A-1.2}$$

Where:

k_g = isentropic exponent of gas in the mixture; ε = critical ratio of pressures; β = ratio of volumetric gas to liquid plus gas composition ($\beta = \frac{V_g}{V_g + V_{ld}}$).

The dependence of the discharge pressure after the FD (jump pressure, P_2) from the pressure before the jump inside of the FD (P_{bj}) is described by the following equation:

$$P_2 = k\,P_{bj}\,M^2 \quad \text{…………………………………...A-1.3}$$

Where:

M = Mach Number (the ratio of the flow's speed to the local sonic speed, M = W/S).

As one can see from Equation A-1.2 the isentropic exponent (k) of a homogeneous two phase flow is determined by the isentropic exponent of gas in the mixture (k_g) and the ratio of volumetric gas to liquid plus gas composition(β) and does not depend on the liquid characteristics.

For the pressure jump condition the sound velocity is related to stream velocities by the following relationship:

$S^2 = W_1 \bullet W_2$ ………………………………A-1.4

Where:

W_1 = stream velocity before the jump; W_2 = stream velocity after the jump.

For the homogeneous two phase flow where the $\rho_g << \rho_{ld}$ the $M^2 = \frac{1}{1 - \beta}$.

The work balance of the FD is described by the following equation:

$$\frac{k}{k-1} P_w V_w \left[\left(\frac{P_w}{P_i} \right)^{\frac{k-1}{k}} - 1 \right] = (P_d - P_i) V_w (u+1)$$

………………………………………..A-1.5

Where:

k = $\frac{C_p}{C_v}$; C_p = specific heat at constant pressure; C_v = specific heat at constant volume;

k = 1.3 for superheated steam; k = 1.13 for dry saturated steam; w, i, d –subscript denoting the following parameters of the working, injected, and discharge streams: P= pressure and V=specific volume; u = injection coefficient equal to the ratio of injected and working flow rates.

Table A-1.1 below presents some values of discharge pressure P^d estimated in accordance with the equation (A-1.5) for different injecting coefficients at the following conditions: the working superheated steam pressure P^w =145psi, T=395F, V=3.36 lb/cu.ft and the injected water pressure P^i =14.5 psi, T= 50F and V=0.16 lb/cu.ft.

Table A-1.1. Dependence of Discharge Pressure on the Injection Coefficient

u	20	50	100
P , psi	4,423	1,813	914

In the FDs transonic flow of a two-phase stream is achieved by the reduction of its velocity. This results in the exchange of motion impulses between the working and injected streams, thus reducing the sonic velocity in the mixing chamber. The stream at the entrance of the mixing chamber (throat) has a velocity equal to or larger than the local sonic velocity. As the result of the stream's deceleration the temperature and pressure at the exit of the mixing chamber increases. Due to the specific design geometry, the discharge pressure can be increased by few times higher than the pressure of the working media. The liquid phase in the mixing chamber has a foam type structure with a very highly turbulent surface area, therefore, the dimensions of the FD are very small when compared with conventional surface type heat exchangers.

The FD operates as follows. The stream of steam leaving the working nozzle (located at some distance of the mixing chamber) condenses during the direct contact with the injected water before the mixing chamber. During this process the temperature and velocity of the injected water are increased. In the entrance in the mixing chamber the water velocity (not uniform) is further increased and the pressure reduced. If this pressure is higher than the pressure of the saturated steam at the local temperature, in the mixing chamber the velocity profile equalizes with resulting pressure increase. In the diffuser the stream pressure is further increased.

Substantial differences in the above described process take place at small injection coefficients. The reduction of the flow rate of the injected water at the constant steam flow rate leads to the increase of the water temperature to the saturation temperature corresponding to the pressure in the mixing chamber, and, because of the shortage of water for condensation of all steam, the performance of the FD breaks-down. This mode determines the minimum injection coefficient. At this mode the operational and geometry factors influence the characteristics of the FD.

With the increase of the injection coefficient, when the flow rate of the injected water (as the result of the reduction of backpressure) is increased, the water temperature in the mixing chamber is reduced. At the same time because of velocity increase in the mixing chamber, the water pressure is reduced. The increase of the flow rate of injected water leads to the reduction of the pressure at the entrance into mixing chamber, up to the saturated pressure corresponding to the temperature of the heated water. Reduction of the backpressure doesn't cause the increase of the water flow rate because further pressure drop in the mixing chamber is impossible. This pressure drop, which determines the flow rate of the injected water, cannot be increased. Further reduction of backpressure at this conditions leads to flashing (cavitation) of the water at the mixing chamber. The cavitation of water in the mixing chamber determines the maximum (limiting) injection coefficient. It should be noted that this operational condition is the working mode of the FD. This explains the important feature of the FD – the independence of the discharge flow from the back pressure at the cavitation mode.

The specific characteristics of the FD are closely related to the geometry of the mixing chamber. The discharge pressure after the FD with a cylindrical shape mixing chamber is presented by the following equation:

$$P^{d} = P^{w} \quad T_{w1}\frac{f_{w1}}{f_3} + \frac{K_1}{\varphi_3} k_w T_{wc} \lambda_{w1} \frac{f_{wc}}{f_3} \quad \left(1 \quad 0.5\varphi_3{}^2\right) k_w \quad \frac{2}{k_w + 1}^{k_w+1/k_w \quad 1} \quad \frac{V_d}{V_w} \quad \frac{f_{wc}}{f_3}^{2} \left(1+u\right)^2 \quad +$$

$$+ \quad 1 \quad \frac{f_{wc}}{f_3} \quad P_i \qquad \text{…..........A-1.6}$$

Where:

$T^{w1} = P^{i} / P_w$; f_{w1} = cross section of the working nozzle exhaust; f_3 = cross section of the mixing chamber exhaust; K_1 = working stream velocity coefficient; φ_3 = diffuser stream velocity coefficient; T^{wc} = P^{c} / P^{w} = ratio of pressure in the critical section of the working nozzle to the working pressure; λ_{w1} = ratio of the velocity of working stream at adiabatic flow to the critical velocity; f_{wc} = cross section of critical section of the working nozzle.

From this equation one can see that when the discharge cross section of the FD is equal to the cross section of mixing chamber, the discharge pressure is independent from the pressure of injected water.

The relationship between the pressure at the entrance in the mixing chamber (P^{2}) and the injection coefficient is determined from the following equation:

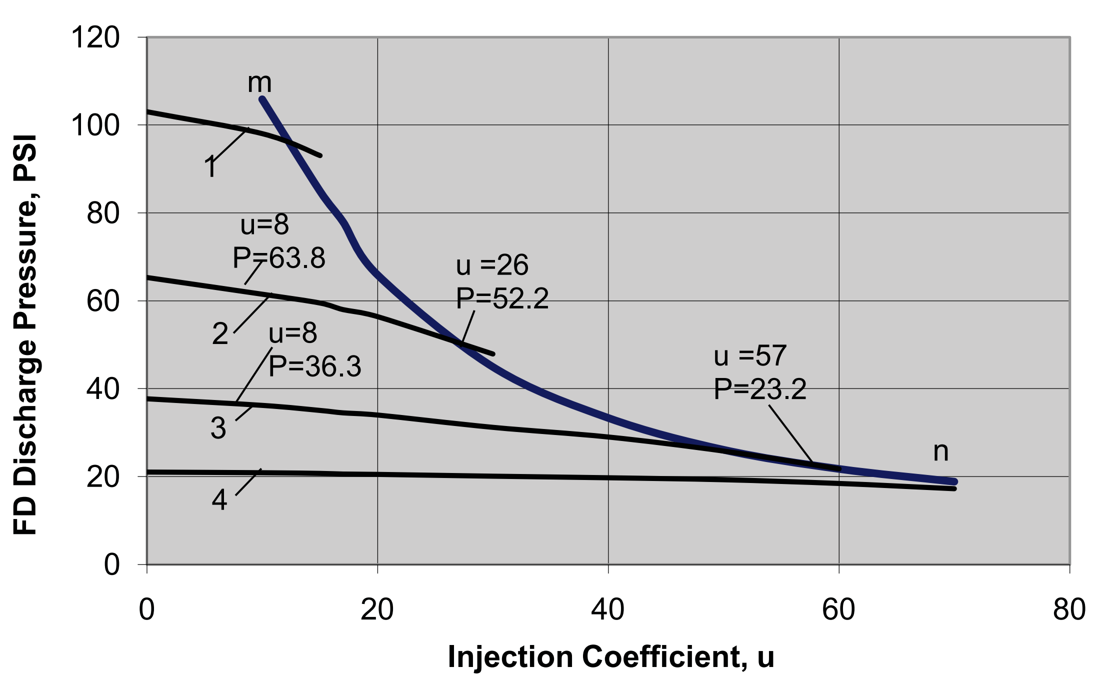
$$\frac{P_2}{P_w} = \frac{P_i}{P_w} \quad \frac{k_w}{2} \quad \frac{2}{k_w + 1}^{k_w + 1/k_w \quad 1} \quad \frac{f_{p*}}{f_2}^{2} \quad \frac{v_i}{v_w}(1+u)^2$$ ……………………………………A-1.7

Figure A-1.2 presents the dependence of the discharge pressure on the injection coefficient at different cross section ratios. The increase of cross section ratio leads to the increase of injection coefficient and reduction of the discharge pressure. In the FD the minimum and maximum injection coefficients are limited by the water boiling conditions in the mixing chamber. At these conditions the pressure in the mixing chamber will became lower than the saturation pressure (cavitation) at water temperature in the mixing chamber. Both these pressures at the given parameters of working steam, injected water and FD dimensions, depend on the injection coefficient.

At the higher temperature of the injected water the condensation rate of working steam is less intensive than at the colder temperature. At these conditions the condensation process may not be completed at the entrance chamber and part of the mixing chamber may be occupied with noncondensed working steam. As a result the cross-section area for injected water flow will be partially reduced and so the maximum injection coefficient.

One can see from Figure A-1.2 that for the given conditions and the maximum injection coefficient of 57 (curve 3), the discharge pressure after the FD is 23.2 psi and the pressure increase of the injected water is 11.6 psi. At the minimum injection coefficient of eight (curve 3), the discharge pressure after the FD is 36.3 psi and the pressure increase of the injected water is 24.7 psi. At the smaller cross section ratio the working range of injection coefficients is reduced substantially. The discharge pressure at these conditions is increasing. At the maximum injection coefficient of 26 (curve 2) the discharge pressure is 52.2 psi – pressure increase by 40.6 psi, at the minimum injection coefficient of eight (curve 2) the discharge pressure is 63.8 psi with pressure increase by 52.2 psi. Further reduction of the cross section ratio results in reduction of difference between minimum and maximum injection coefficients and at some point they became equal. At the further reduction of the cross section ratio the FD cannot be operated.

Figure A-1.2 Estimated Performance of FD



Steam Working Pressure 87 psi; Injection Pressure 11.6 psi; 1, 2, 3 and 4 – Performance Curves of the FD at Different Cross-section Ratios of: 1; 1.8; 4; and 10. **mn** Curve – Maximum Injection Coefficients.

The limiting injection coefficients can be estimated from the following equation:

$$u = \sqrt{\frac{2}{k}\,\frac{k+1}{2}}^{\frac{k+1}{k-1}} \text{ x } \frac{f_3}{f_{wc}}\sqrt{\frac{P_i - P_s}{P_w}}\sqrt{\frac{v_w}{v_i}} - 1 \quad \text{...A-1.8}$$

Where:

P^s = the saturation pressure in the mixing chamber.

The maximum injection coefficient is estimated from the following equation:

$$u = \frac{C\sqrt{P_i - P_s}}{(P_d - P_i) + (2 - \varphi_3^{2})(P_i - P_s)} - 1 \quad \text{..A-1.9}$$

Where:

$$C = \frac{2K_1}{\varphi_3}\sqrt{\frac{k_w}{k_w+1}}\sqrt{\frac{\upsilon_w P_w \lambda_{w1}}{\upsilon_i}} \quad \text{..................A-1.10}$$

υ_w and υ_i = specific volume of working and injected streams.

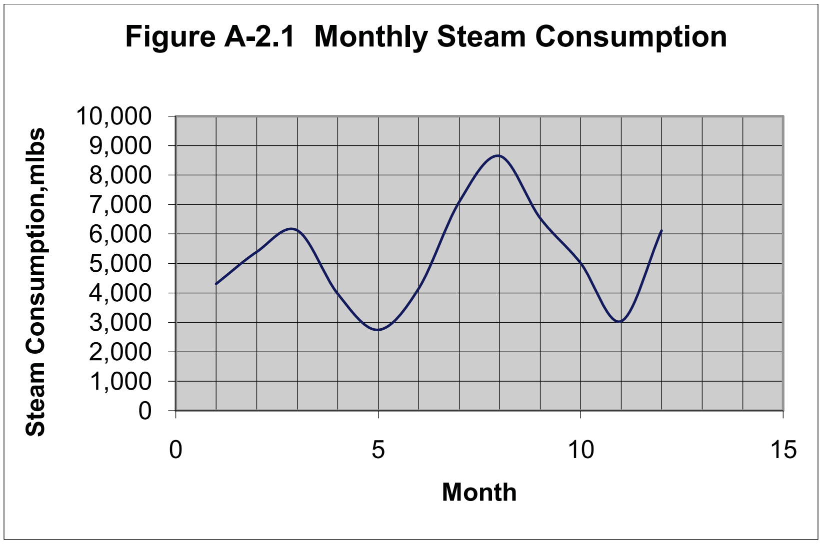
Section A-2: Assessment of Heat Loads for the 4 Irving Place Building, NY

An economic comparison of the existing heating system of the 4 Irving Place building with the proposed heating system using Fisonic Devices requires the determination of all components of the heating load. The heating load of the building consists of space heating hot water supply to the building perimeter radiators, space heating steam for ventilation (direct steam supply to the air handler heating coils), domestic hot water and direct steam supply to the cafeteria kitchen. An assessment of the building heating loads has been performed using the monthly steam consumption obtained from Con Edison (Table A-2.1). The monthly steam consumption is also presented in Figure A-2.1.

Table A-2.1. Monthly Steam Consumption

Month	mlbs	Base Cons, mlbs	Heating Steam Cons, mlbs	Cooling Steam Cons, mlbs
Dec.07	6,114	2,893	3,221	
Nov. 07	3,039	2,893	146	
Oct. 07	5,015	2,893		2,122
Sept.07	6,542	2,893		3,649
Aug. 07	8,646	2,893		5,753
Jul. 07	7,080	2,893		4,187
Jun. 07	4,145	2,893		1,252
May. 07	2,747	2,893		
Apr. 07	3,976	2,893	1,083	
Mar. 07	6,125	2,893	3,232	
Feb. 07	5,390	2,893	2,497	
Jan. 07	4,307	2,893	1,414	
TOTAL	63,126	34,716	11,593	16,963

The following calculation method was employed to determine heating demand from annual steam consumption and site-specific parameters. Based on steam consumption records for a consecutive twelve-month period, both annual and peak hourly heating loads were determined for comfort heating and internal uses such as domestic hot water and cafeteria kitchen. Parameters specific to individual building operation and function required for this methodology included the following items obtained or derived during the site visit:



- Annual steam consumption dedicated to comfort heating or other uses such as domestic hot water and cafeteria kitchen
- Average number of normal heating hours per week, usually defined as the period of time when the building is occupied and suitable for normal activities
- Average number of setback heating hours per week, usually defined as the period of time when the building is unoccupied and/or heating levels are reduced to save energy
- Indoor design temperature setting during normal heating hours, usually defined as the temperature maintained by a thermostat during normal heating hours
- Indoor temperature gain during normal heating hours is defined as the air temperature rise in a building due to internal heat sources such as people, lighting, equipment, etc. Generally expressed in degrees, it is determined by building function. A typical value of 5ºF is used for most facilities
- Indoor temperature gain during setback heating hours due to internal heat sources is defined as for item above, but applies to building operation during setback hours.

Site-specific parameters that correlate annual consumption figures with a peak heating load include:

- Outdoor winter design temperature. Based on ASHRAE weather data a value of 11ºF was used for the New York City
- Average annual heating degree-days based on a twelve-month summation for New York City. A degree-day is a unit based on temperature difference and time. For any one day when the average temperature falls below 65ºF, there are as many degree-days as the temperature difference between 65ºF and the average temperature for the day.

The following detailed heating load analysis of the 4 Irving Place building was performed employing the following methodology. Input data includes monthly steam consumption records, heating degree-days, and building operating parameters such as indoor design temperature, indoor setback temperatures, normal and setback system operating hours per week, and outside air ventilation quantities. For the 4 Irving Place building the following data was obtained during the survey:

Table A-2.2. Information Obtained from Building Survey

Parameter	Symbol	Unit	Value
Normal indoor temperature	T_{in}	°F	70
Setback indoor temperature	T_{is}	°F	60
Internal temperature gain	T_{ig}	°F	5
Normal ventilation rate	V_n	cfm	55,500
Setback ventilation rate	V_s	cfm	1,338
Total annual heat consumption for space heating & DHW and kitchen	F_a	MMBtu	46,309
Total off-heating season heat consumption	F_{os}	MMBtu	17,358
Beginning of heating season	S_b		11/1/2006
End of heating season	S_e		5/1/2007
HDD in the billing year	HDD_{actual}	HDD	4,749

The design temperature and hourly profile was obtained from weather sources. The parameters used for this building are summarized in Table A-2.3.

In the following equations, all symbols are defined in Tables A-2.2 and A-2.3.The building heat load consists of domestic hot water (DHW), cafeteria kitchen, ventilation and comfort heating loads.

Table A-2.3. Parameters Obtained from Auxiliary Sources

Parameter	Symbol	Unit	Value
Design temperature	T_d	°F	11
Seasonal steam conversion efficiency	η		100%
Specific heat of air at 40°F	Cp	Btu/lb°F	0.24
Specific volume of air at 40°F	v	ft^3/lb	12.59
HDD in the standard year	$HDD_{standard}$	HDD	4,909

The total annual useful heat consumption (Q_a) can be calculated using the total annual steam consumption (available from the records on monthly basis) and seasonal conversion efficiency as:

$$Q_a = F_a \cdot \eta \quad \text{A-2.1}$$

The total heat consumption during off-heating season will be used to estimate the portion of annual consumption that corresponds to DHW and cafeteria kitchen load. Heat consumption during the off-heating season (Q_{os}) can be estimated as:

$$Q_{os} = F_{os} \cdot \eta \quad \text{A-2.2}$$

Duration of heating season (D_s), days:

$$D_s = S_e \quad S_b \quad \text{A-2.3}$$

Then the average daily DHW consumption (QI_{av}) can be calculated as:

$$QI_{av} = Q_{os} / 365 \quad D_s \qquad \text{A-2.4}$$

The total annual DHW consumption (QI_a) is calculated as follows:

$$QI_a = QI_{av} \quad 365 \qquad \text{A-2.5}$$

The calculation of daily DHW consumption was based on the assumption that the DHW load remains constant throughout the year.

The annual heat consumption for comfort heating and ventilation (QHV), then, can be calculated using:

$$QHV = Q_a \quad QI_a \qquad \text{A-2.6}$$

This annual heat consumption corresponds to a particular year for which fuel data was obtained. This year could have a different number of degree-days than the average (standard) year for which we are designing the district energy system, and for which the temperature data is available. Therefore, the annual heat consumption for comfort heating and ventilation should be normalized to the standard year (with average outdoor temperatures for a 30-year period of record). This is evaluated as follows:

$$QHV_{norm} = \frac{QHV \quad HDD_{standard}}{HDD_{actual}} \qquad \text{A-2.7}$$

Where: QHV norm = Normalized annual heat consumption for comfort heating and ventilation, MMBtu.

Parameters calculated to this point of the analysis are summarized in the Table A-2.4.

Next, we calculate the consumption and load corresponding to ventilation. The ventilation heat load should be calculated for each temperature interval (bin) for normal and setback operation, using the following equations:

Table A-2.4. Calculated Parameters

Parameter	Symbol	Unit	Value
Total annual heat consumption	Q_a	MMBtu	46,309
Total heat cons. during off heating season	Q_{os}	MMBtu	17,358
Duration of heating season	D_s	days	181
Average daily DHW and kitchen heat consumption	QI_{av}	MMBtu	94.34
Total annual DHW and kitchen heat consumption	QI_a	MMBtu	34,433
Total annual comfort heating and vent. heat cons	QHV	MMBtu	11,876
Norm. total annual comfort heating and vent. heat cons.	QHV_{norm}	MMBtu	12,276

$$\dot{Q}V_n = V_n \frac{ft^3}{min} \quad 60 \frac{min}{hr} \quad \frac{1}{v} \frac{lb}{ft^3} \quad Cp \frac{Btu}{lb \ °F} \quad (T_{in} \quad T)(°F);$$

For normal operation A-2.8

and

$$\dot{Q}V_s = V_s \frac{ft^3}{min} \quad 60 \frac{min}{hr} \quad \frac{1}{v} \frac{lb}{ft^3} \quad Cp \frac{Btu}{lb \; °F} \quad (T_{is} \quad T)(°F) \; ;$$

For setback operation A-2.9

Where: $\dot{Q}V_n$ = Ventilation heat load for normal operation, MMBtu/hr

$\dot{Q}V_s$ = Ventilation heat load for setback operation, MMBtu/hr

T = the bin temperature.

The peak ventilation load is calculated using the formula for normal operation and:

$T = T_d$ A-2.10

The annual consumption for ventilation (QV) is calculated by integration of the ventilation heat load by the hours of occurrence during the heating season and it is summarized in the Table A-2.5.

Using Equations A-2.8 and A-2.10, the peak ventilation heat load for the building is estimated at 3.74 MMBtu/hr for the design outdoor temperature of 11°F.

The annual consumption for comfort heating (Q) is estimated using the Equation A-2.11.

$$Q = QHV_{norm} \quad QV \quad \text{A-2.11}$$

Where: QV= Annual ventilation heat consumption, MMBtu.

In order to estimate the heat loads for comfort heating, the ratio of bin to peak comfort heating load $\dot{Q}/\dot{Q}_{peak}$ should be calculated for each bin temperature for normal $\dot{Q}_n/\dot{Q}_{peak}$ and setback $\dot{Q}_s/\dot{Q}_{peak}$ operation. These calculations are performed using the following formulas:

$$\frac{\dot{Q}_n}{\dot{Q}_{peak}} = \frac{(T_{in} \quad T_{ig} \quad T)}{(T_{in} \quad T_{ig} \quad T_d)}$$; for normal operation A-2.12

and

$$\frac{\dot{Q}_s}{\dot{Q}_{peak}} = \frac{\left(T_{is} \quad T_{ig} \quad T\right)}{\left(T_{in} \quad T_{ig} \quad T_d\right)}$$; for setback operation A-2.13

The annual peak comfort heating load is calculated using the following iterative methodology. First, the initial assumed value for the annual peak comfort heating load ($\dot{Q}$ assumed peak) is selected and the values of comfort heating load ($\dot{Q}$) are calculated for each bin temperature for normal and setback operation using Equation A-2.14.

$$\dot{Q} = \dot{Q} / \dot{Q}_{peak} \quad \dot{Q}_{assumed\ \ peak}$$ A-2.14

Table A-2.5. Ventilation Heat Load

Average outside temp, °F	Observed normal hours during the	Observed setback hours during the	Ventilation load normal, MMBtu/hr	Ventilation load setback, MMBtu/hr	Ventilation consumption, MMBtu
T	h_n	h_s	$\dot{QV}_n$	QV_s	QV
-3	-	1	4.63	0.10	0
2	-	8	4.32	0.09	1
7	9	26	4.00	0.08	39
12	22	61	3.68	0.08	86
17	53	116	3.36	0.07	186
22	101	178	3.05	0.06	317
27	164	289	2.73	0.05	464
32	262	435	2.41	0.04	650
37	347	495	2.09	0.04	746
42	362	467	1.78	0.03	657
47	344	427	1.46	0.02	511
52	322	429	1.14	0.01	374
57	330	450	0.83	0.00	275
62	349	478	0.51	-	177
67	371	494	0.19	-	71
72	402	378	-	-	-
77	295	107	-	-	-
82	116	17	-	-	-
87	32	3	-	-	-
92	8	-	-	-	-
97	1	-	-	-	-
Total annual	3,892	4,859			4,553

Then, the annual consumption for comfort heating is calculated by integration of the ventilation heat load by the hours of occurrence during the heating season. Finally, such value of the peak comfort heating load is found for which the value of annual heating consumption for comfort heating calculated by integration is equal to the value calculated using Equation 11. The comfort heating load for the building is presented in Table A-2.6.

The peak comfort heating load of 3.76 MMBtu/hr satisfies the annual comfort heating consumption calculated using Equation A-2.11.

Table A-2.6. Comfort Heating Load

Average outside temp, °F	Observed normal hours during the heating season	Observed setback hours during the heating season	Normal load, %	Setback load, %	Comfort heating load normal, MMBtu/hr	Comfort heating load setback, MMBtu/hr	Comfort heating cons., MMBtu
T	h_n	h_s	$\dot{Q}_n / \dot{Q}_{peak}$	$\dot{Q}_s / \dot{Q}_{peak}$	$\dot{Q}_n$	$\dot{Q}_s$	Q
-3	-	1	126%	107%	4.86	4.15	4
2	-	8	117%	98%	4.50	3.79	30
7	9	26	107%	89%	4.15	3.43	127
12	22	61	98%	80%	3.79	3.07	271
17	53	116	89%	70%	3.43	2.72	497
22	101	178	80%	61%	3.07	2.36	729
27	164	289	70%	52%	2.72	2.00	1,025
32	262	435	61%	43%	2.36	1.64	1,333
37	347	495	52%	33%	2.00	1.29	1,333
42	362	467	43%	24%	1.64	0.93	1,029
47	344	427	33%	15%	1.29	0.57	687
52	322	429	24%	6%	0.93	0.21	392
57	330	450	15%	0%	0.57	-	189
62	349	478	6%	0%	0.21	-	75
67	371	494			-	-	-
72	402	378			-	-	-
77	295	107			-	-	-
82	116	17			-	-	-
87	32	3			-	-	-
92	8	-			-	-	-
97	1	-			-	-	-
total annual	3,892	4,859					7722

The total heating load calculations are presented in Table A-2.7.

In order to construct an annual heat load duration curve, the hourly comfort heating, ventilation and DHW and kitchen loads for each instance of occurrence should be added, and sorted out in descending order, as shown in the Table A-2.8. The hourly DHW and kitchen load is calculated as $QI_{av}/24$.

Table A-2.7. Total Heating Load

Average outside temp, °F	Observed normal hours during the heating season	Observed setback hours during the heating season	Total normal heating load, MMBtu/hr	Total setback heat load, MMBtu/hr	Total heating consumption MMBtu
T	h_n	h_s	$\dot{Q}HV_{norm\ n}$	$\dot{Q}HV_{norm\ s}$	Q
2	-	8	8.82	3.88	31
7	9	26	8.15	3.52	165
12	22	61	7.47	3.15	357
17	53	116	6.80	2.79	683
22	101	178	6.12	2.42	1049
27	164	289	5.45	2.05	1487
32	262	435	4.77	1.69	1985
37	347	495	4.10	1.32	2077
42	362	467	3.42	0.96	1686
47	344	427	2.75	0.59	1198
52	322	429	2.07	0.23	765
57	330	450	1.40	0.00	463
62	349	478	0.72	-	252
67	371	494	0.19	-	71
72	402	378	-	-	
77	295	107	-	-	
82	116	17	-	-	
87	32	3	-	-	
92	8	-	-	-	
97	1	-	-	-	
Total annual	3,890	4858			12267

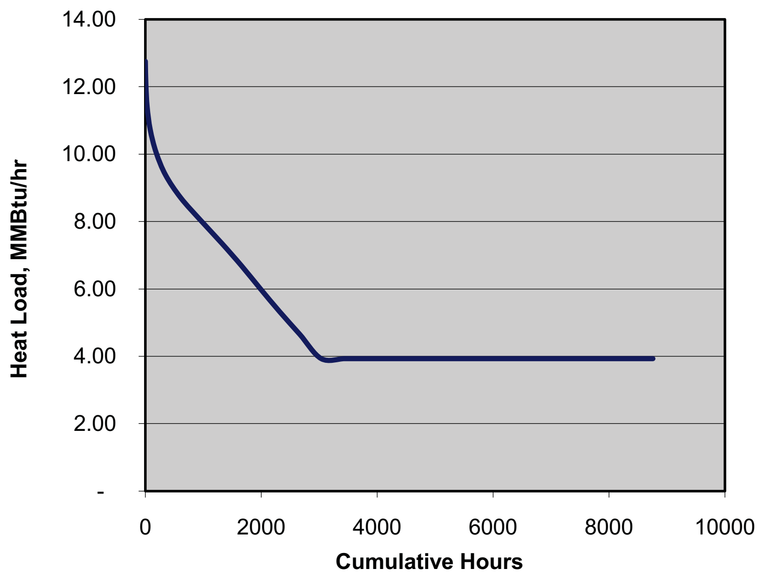
The total annual heat load duration curve for the building is presented in Figure A-2.2.

The above assessment determined the peak heat load of the space heating hot water system as 3.76 MMBtu/hr and the ventilation peak heat load as 3.74 MMBtu/hr.The average DHW and cafeteria kitchen load is 3.93 MMBtu/hr.This information was used to size and select the heat transfer equipment in Section 2 of this report.

Table A-2.8. Annual Heat Load Duration Data

Hourly Load, MMBtu/hr	Bin hours	Cumulative hours
12.75	0	0
12.08	9	9
11.40	22	31
10.73	53	84
10.05	101	185
9.38	164	349
8.70	262	611
8.03	347	958
7.35	362	1320
6.68	344	1664
6.00	322	1986
5.33	330	2316
4.65	349	2665
3.93	371	3036
3.93	402	3438
3.93	295	3733
3.93	116	3849
3.93	32	3881
3.93	8	3889
3.93	178	4067
3.93	289	4356
3.93	435	4791
3.93	349	5140
3.93	495	5635
3.93	467	6102
3.93	427	6529
3.93	371	6900
3.93	429	7329
3.93	450	7779
3.93	981	8760

Figure A-2.2 Annual Heat Load Duration



Section A-3: Assessment of Heat Loads for the Woolworth Building, NY

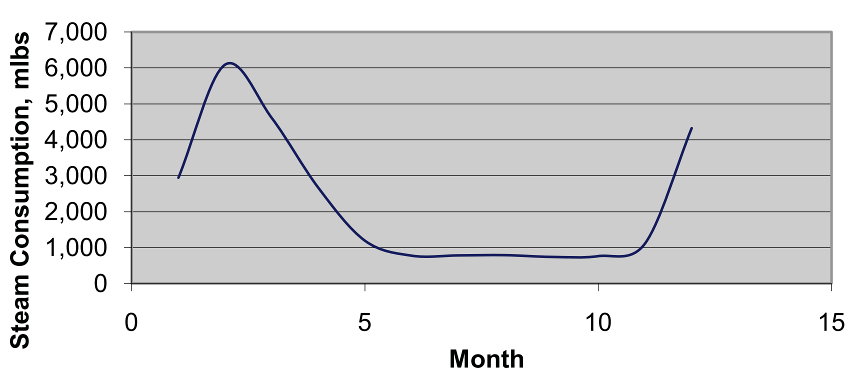
Assessment of Heating Loads

The heating load of the building consists of space heating hot water supply to the building perimeter radiators and to 80 ton units hot water coils for ventilation, and domestic hot water. An assessment of the building heating loads has been performed using the monthly steam consumption obtained from Con Edison (Table A-3.1). The monthly steam consumption is also presented in Figure A-3.1.

Table A-3.1. Monthly Steam Consumption

Month	mlbs	Base Cons, mlbs	Heating Steam Cons, mlbs
Jan.	2,944	768	2,176
Feb.	6,094	768	5,326
Mar.	4,610	768	3,842
Apr.	2,662	768	1,894
May	1,188	768	420
June	773	768	0
July	781	768	0
Aug	788	768	0
Sept.	739	768	0
Oct.	761	768	0
Nov.	1,113	768	345
Dec.	4,325	768	3,557
TOTAL	26,778	9,216	17,560

Figure A-3.1 Annual Steam Consumption



The calculation method employed to determine heating demand from annual steam consumption and site-specific parameters was described in details in Section A-2. For the Woolworth building the following data obtained during the survey is presented in Table A-3.2.

Table A-3.2. Information Obtained from Building Survey

Parameter	Symbol	Unit	Value
Normal indoor temperature	T_{in}	°F	70
Setback indoor temperature	T_{is}	°F	60
Internal temperature gain	T_{ig}	°F	5
Normal ventilation rate	V_n	cfm	42,291
Setback ventilation rate	V_s	cfm	1,057
Total annual heat consumption for space heating & DHW	F_a	MMBtu	26,778
Total off-heating season heat consumption	F_{os}	MMBtu	3,840
Beginning of heating season	S_b		11/1/2006
End of heating season	S_e		6/1/2007
HDD in the billing year	HDD_{actual}	HDD	4,666

Table A-3.3. Parameters Obtained from Auxiliary Sources

Parameter	Symbol	Unit	Value
Design temperature (F	T_d	°F	11
Seasonal boiler efficiency	η		100%
Specific heat of air at 40°F	Cp	Btu/lb°F	0.24
Specific volume of air at 40°F	v	ft^3/lb	12.59
HDD in the standard year	$HDD_{standard}$	HDD	4,909

Parameters calculated to this point of the analysis are summarized in the Table A-3.4 below.

Table A-3.4. Calculated Parameters

Parameter	Symbol	Unit	Value
Total annual heat consumption	Q_a	MMBtu	26,778
Total heat cons. during off heating season	Q_{os}	MMBtu	3,840
Duration of heating season	D_s	days	212
Average daily DHW heat consumption	QI_{av}	MMBtu	25.10
Total annual DHW heat consumption	QI_a	MMBtu	9,161
Total annual comfort heating and vent. heat cons	QHV	MMBtu	17,617
Norm. total annual comfort heating and vent. heat cons.	QHV_{norm}	MMBtu	18,535

The results of the ventilation heat load calculations are presented in Table A-3.5.

Table A-3.5. Ventilation Heat Load

Average outside temp, °F	Observed normal hours during the	Observed setback hours during the	Ventilation load normal, MMBtu/hr	Ventilation load setback, MMBtu/hr	Ventilation consumption, MMBtu
T	h_n	h_s	$\dot{Q}V_n$	$\dot{Q}V_s$	QV
-3	-	1	3.53	0.08	0
2	-	8	3.29	0.07	1
7	9	26	3.05	0.06	30
12	22	61	2.80	0.06	66
17	53	116	2.56	0.05	142
22	101	178	2.32	0.05	242
27	164	289	2.08	0.04	353
32	262	435	1.84	0.03	496
37	347	495	1.60	0.03	568
42	362	467	1.35	0.02	500
47	344	427	1.11	0.02	389
52	322	429	0.87	0.01	285
57	330	450	0.63	0.00	209
62	349	478	0.39	-	135
67	371	494	0.15	-	54
72	402	378	-	-	-
77	295	107	-	-	-
82	116	17	-	-	-
87	32	3	-	-	-
92	8	-	-	-	-
97	1	-	-	-	-
Total annual	3,892	4,859			3,469

From Table A-3.5 the peak heating load attributed to ventilation is estimated as 2.85 MMBtu/hr.Then, the annual consumption for comfort heating is calculated by integration of the ventilation heat load by the hours of occurrence during the heating season. Finally, the comfort heating load for the building is calculated and presented in Table A-3.6.

From Table A-3.6 the peak heating load for space heating without ventilation is estimated as 7.53 MMBtu/hr.

Table A-3.6. Comfort Heating Load

Average outside temp, °F	Observed normal hours during the heating season	Observed setback hours during the heating season	Normal load, %	Setback load, %	Comfort heating load normal, MMBtu/hr	Comfort heating load setback, MMBtu/hr	Comfort heating cons., MMBtu
T	h_n	h_s	$\dot{Q}_n / \dot{Q}_{peak}$	$\dot{Q}_s / \dot{Q}_{peak}$	$\dot{Q}_n$	$\dot{Q}_s$	Q
-3	-	1	126%	107%	9.49	8.09	8
2	-	8	117%	98%	8.79	7.39	59
7	9	26	107%	89%	8.09	6.70	248
12	22	61	98%	80%	7.39	6.00	529
17	53	116	89%	70%	6.70	5.30	970
22	101	178	80%	61%	6.00	4.60	1,422
27	164	289	70%	52%	5.30	3.91	2,000
32	262	435	61%	43%	4.60	3.21	2,600
37	347	495	52%	33%	3.91	2.51	2,600
42	362	467	43%	24%	3.21	1.81	2,008
47	344	427	33%	15%	2.51	1.12	1,340
52	322	429	24%	6%	1.81	0.42	764
57	330	450	15%	0%	1.12	-	369
62	349	478	6%	0%	0.42	-	146
67	371	494			-	-	-
72	402	378			-	-	-
77	295	107			-	-	-
82	116	17			-	-	-
87	32	3			-	-	-
92	8	-			-	-	-
97	1	-			-	-	-
total annual	3,892	4,859					15065

The total heating load calculations are presented in Table A-3.7.

In order to construct an annual heat load duration curve, the hourly comfort heating, ventilation and DHW and kitchen loads for each instance of occurrence should be added, and sorted out in descending order, as shown in the Table A-3.8. The hourly DHW load is calculated as $QI_{av}/24$.

Table A-3.7. Total Heating Load

Average outside temp, °F	Observed normal hours during the heating season	Observed setback hours during the heating season	Total normal heating load, MMBtu/hr	Total setback heat load, MMBtu/hr	Total heating consumption MMBtu
T	h_n	h_s	$\dot{Q}HV_{norm\ n}$	$\dot{Q}HV_{norm\ s}$	Q
2	-	8	12.08	7.46	60
7	9	26	11.14	6.76	276
12	22	61	10.20	6.06	594
17	53	116	9.26	5.35	1112
22	101	178	8.32	4.65	1668
27	164	289	7.38	3.95	2351
32	262	435	6.44	3.24	3098
37	347	495	5.50	2.54	3166
42	362	467	4.56	1.84	2509
47	344	427	3.62	1.13	1730
52	322	429	2.68	0.43	1048
57	330	450	1.74	0.00	577
62	349	478	0.81	-	281
67	371	494	0.15	-	54
72	402	378	-	-	
77	295	107	-	-	
82	116	17	-	-	
87	32	3	-	-	
92	8	-	-	-	
97	1	-	-	-	
Total annual	3,890	4858			18521

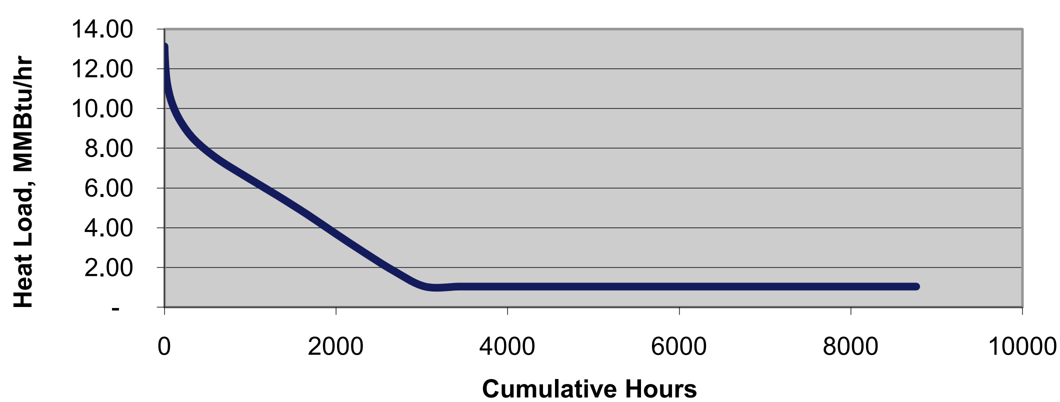
The total annual heat load duration curve for the building is presented in Figure A-3.2.

The above assessment determined the peak heat load of the space heating hot water system as 7.53 MMBtu/hr and the ventilation peak load as 2.85 MMBtu/hr.The total space heating load is 10.38 MMBtu/hr.The average DHW load is 1.05 MMBtu/hr.This information was used to size and select the heat transfer equipment in Section 3 of this report.

Table A-3.8. Annual Heat Load Duration Data

Hourly Load, MMBtu/hr	Bin hours	Cumulative hours
13.12	0	0
12.18	9	9
11.24	22	31
10.30	53	84
9.37	101	185
8.43	164	349
7.49	262	611
6.55	347	958
5.61	362	1320
4.67	344	1664
3.73	322	1986
2.79	330	2316
1.85	349	2665
1.05	371	3036
1.05	402	3438
1.05	295	3733
1.05	116	3849
1.05	32	3881
1.05	8	3889
1.05	178	4067
1.05	289	4356
1.05	435	4791
1.05	349	5140
1.05	495	5635
1.05	467	6102
1.05	427	6529
1.05	371	6900
1.05	429	7329
1.05	450	7779
1.05	981	8760

Figure A-3.2 Annual Heat Load Duration



Section A-4: Conversion of the Louis Lefkowitz Building to Hot Water with FDs

This section presents the methodology of conversion of the building from steam to hot water operation with Fisonic Devices and assessment of heat loads.

Advantages of Hot Water versus Steam

Many buildings in Manhattan connected to the Con Edison district steam system distribute steam throughout the building for space heating. JTC extensive experience of converting steam heated buildings to hot water in the US and overseas demonstrates that hot water operation can provide substantial energy savings to the building owners. Retrofit of steam heated buildings to hot water or air is sporadically performed during rehabilitation or renovation of buildings. A consistent development of a steam to hot water conversion technology with documented results is not available. Moreover, the advantages of such technology are not documented and widely known in the engineering field.

The advantages of hot water operation versus steam are as follows:

1. Substantial reductions in heat losses in the building distribution system. A hot water system also eliminates the heat losses from steam traps, valve stems, packing gland leakage, and flash losses.
2. Better temperature control. A hot water system operates with variable supply temperatures and flow rates closely following the demand. A steam system is subject to fluctuating outlet temperatures totally dependent upon steam pressure. Hot water yields a more uniformly distributed temperature. The worst source of inefficiency in steam heated buildings is uneven heating. To minimize complaints from cooler areas of the buildings, owners are forced to grossly overheat other areas, leading tenants to open their windows for relief. Monitoring of steam heated buildings demonstrated that the primary causes of the uneven heating were large differences in steam arrival times. The radiators farthest from the steam supply source often received steam 15 to 25 minutes later than the radiators closest to the source, due to the large thermal mass of the distribution system relative to the steam input rate, poor venting, and low pressure operation. The system operates in short bursts, which continually fill the near radiators but only fill the far radiators once every few cycles. To make more distant areas warm the areas close to the steam source must be grossly overheated.
3. The circulating hot water provides a better heating medium than steam for two reasons. Water retains its heat and stays in the radiator when the steam source shuts down. This provides more even heating and great overall comfort. When the system starts up again, a steam source must raise the temperature over the boiler point, requiring more steam than a hot water system that must raise the water temperature only a relatively few degrees. Also, when the hot water system starts up, warmed water immediately starts circulating through the radiators bringing instant heat. In addition, hot water preserves the life of the system which is greatly extended when the heating system is converted to hot water because of the elimination of corrosive condensates.
4. Closed system. Closed hot water system has practically no make-up requirements. The only unavoidable losses are minor leakage at pump glands and valve steams. A steam system can require from 1 to 100% make-up depending on the type of condensate return system and water quality. Losses from atmospheric leakage and condensate can necessitate significant make-up involving a chemical treatment facility and deaeration. Con Edison steam system has a 100% make-up that is provided at steam generating stations.

5. High pipe corrosion rate, particularly the condensate lines. A hot water system provides a relatively corrosion-free environment after initial chemical treatment; a steam system's condensate return is notorious for corrosion problems.
6. Superior heat storage characteristics. In a hot water system, a large volume of water in constant circulation provides a continuous heat reservoir available to handle peak demands or swings in load. In a steam system, meeting a sudden system surge depends upon how fast the steam unit can produce steam and how rapidly the user can pull on the system without causing the pressure and temperature to decay. Thermal storage is a function of the heat capacity per unit volume, and a hot water system can store up to 50 times as much heat as a steam system.
7. Substantial release of organic dust (dust frying) on the steam heated surfaces that have a temperature of 220-250°F – much higher than the hot water heating elements (160°F). In order to keep the IAQ acceptable a much higher ventilation rate is required (about 30%). Because of the negative impact on the indoor air quality, use of steam heated systems is prohibited in Europe for residential, commercial and institutional buildings.
8. Water hammer noise problems.
9. Flexibility of piping distribution systems. Since a hot water system uses forced circulation, the piping lines can follow the contours of the building, as the heating medium remains liquid. Steam and condensate return lines require a certain degree of pitch for drip legs, traps, etc.
10. Lowered maintenance. In a hot water system, the heat medium stays in a liquid state. Thus there is not need for pressure reducing stations, condensate receivers, traps, drip legs, strainers, etc., as with a steam system. Maintenance problems associated with steam traps alone are often significant.
11. Safety. Hot water is less dangerous than steam.
12. Operational experience demonstrates that the retrofit of steam-based systems to hot water operation may offer system efficiency improvements in the range of 20 to 25%, silent operation, substantial improvement of comfort conditions of the occupants, improvement of indoor air quality, reduction of the ventilation rate and cost of the heating system. See, for example, the published test results: "The final area we explored for steam heated buildings is conversion to hot water heat. This has produced dramatic savings, averaging 27% of total gas use (3,930 terms) for two-pipe steam buildings (TPS) and 18% (4,410 terms) for single pipe steam (SPS) buildings," by Martha J. Hewett, May Sue Lobenstein and Susan K. Nathan "The Science and Art of Retrofitting Low-Rise Multifamily Buildings," ASHRAE Journal, May 1994.

Conversion of Steam Systems to Hot Water

The following types of steam systems are typically used:

1. Radiation

- **One-pipe Cast Iron Radiators** – Steam is supplied to the bottom of cast iron radiators with a single pipe that is also used to for the condensate return.
- **Two-pipe Cast Iron Radiators** – Steam is supplied to the top of cast iron radiators and a separate pipe is used for the condensate return. The radiators are equipped with steam traps, which allow for passage of condensate but not steam.
- **Finned Tube Radiators/Convectors** – Steam is supplied to heat transfer tubes or pipes with an extended surface of fins, disks or ribs, and a separate pipe with a steam trap removes the condensate.

2. Air Handling Systems

- **Coils in Air Handling Units** – Steam is supplied to heating coils of the air handling units and a separate pipe with a steam trap removes the condensate.

- **Coils in Terminal Units** – Steam is supplied to coils of the terminal equipment located in each heating zone, such as fan coils, unit ventilators, unit heaters, terminal reheat in ducts, etc. and a separate pipe with a steam trap removes the condensate.

Joseph Technology Corporation, Inc (JTC) extensive experience in the US and overseas has indicated that in most cases the existing steam heating systems can be successfully used for hot water circulation without replacing the radiators, terminal equipment, and distribution piping. Before the actual retrofit the following analysis and testing should be performed:

- Hydraulic analysis to determine if the condensate piping has the capacity to allow the circulation of the required hot water flow rate
- The steam and the condensate piping are in good condition and can withstand the static pressure of the building height
- The radiators sections are connected at the top and the bottom
- Hydro test of the steam system to determine the location of existing leaking points
- Presents of shut-off, control valves and vents on each radiator or terminal equipment.

Typically the conversion of the steam system to hot water operation involves the following steps:

- Buildings with one-pipe steam systems are equipped with air vents, return condensate lines, thermostatic control valves on all radiators (to provide the individual control), expansion tanks, steam to hot water heat exchangers, hot water circulation pumps, controls and connections to the make-up water system.
- Buildings with two-pipe steam systems are equipped with air vents, thermostatic control valves, expansion tanks, steam to hot water heat exchangers, hot water circulation pumps, controls and connections to the make-up water system. Existing steam traps are removed or disabled.

For the system with the FDs, no heat exchangers and electric driven circulating pumps (except for one electric driven pump) will be required.

Assessment of Building Heating Loads

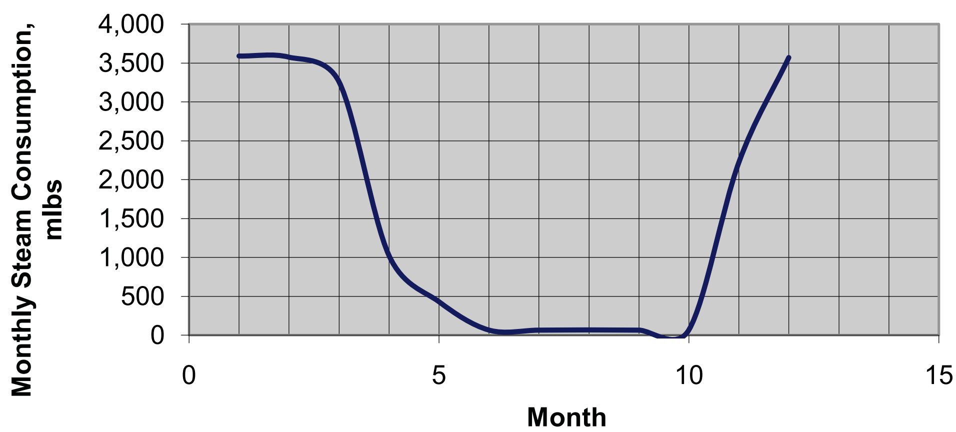
An economic comparison of the existing heating system building with the proposed heating system using Fisonic Devices (FDs) requires the determination of all components of the heating load. The heating load of the building consists of a space heating steam heat supply to the building perimeter radiators, partially by window air conditioners, and domestic hot water (generated in tube and shell heat exchanger).An assessment of the building heating loads has been performed using the monthly steam consumption obtained from Con Edison (Table A-4.1 and Figure A-4.1).

The calculation method employed to determine heating demand from annual steam consumption and site-specific parameters was described in details in Section A-2.

Table A-4.1. Monthly Steam Consumption

Month	Total,mlbs	DHW, mlbs	Space Heating, mlbs
Dec.07	3,569	64	3,505
Nov. 07	2,219	64	2,155
Oct. 07	66	64	**2**
Sept. 07	64	64	**0**
Aug. 08	67	64	**3**
Jul. 08	64	64	**0**
Jun. 08	65	64	**1**
May. 08	430	64	366
Apr. 08	1,025	64	961
Mar. 08	3,259	64	3,195
Feb. 08	3,574	64	3,510
Jan. 08	3,589	64	3,525
Total	**17,991**	**768**	**17,223**

Figure A-4.1 Annual Steam Consumption



For the subject building the data obtained during the survey is presented in Table A-4.2. Parameters obtained from auxiliary sources are presented in Table A-4.3.

Table A-4.2. Information Obtained from Building Survey

Parameter	Symbol	Unit	Value
Normal indoor temperature	T_{in}	°F	70
Setback indoor temperature	T_{is}	°F	60
Internal temperature gain	T_{ig}	°F	5
Normal ventilation rate	V_n	cfm	18,796
Setback ventilation rate	V_s	cfm	470
Total annual heat consumption for space heating & DHW	F_a	MMBtu	17,991
Total off-heating season heat consumption	F_{os}	MMBtu	320
Beginning of heating season	S_b		11/1/2006
End of heating season	S_e		6/1/2007
HDD in the billing year	HDD_{actual}	HDD	4,666

Table A-4.3. Parameters Obtained from Auxiliary Sources

Parameter	Symbol	Unit	Value
Design temperature (Ref. 1-2)	T_d	°F	11
Seasonal boiler efficiency	η		100%
Specific heat of air at 40°F	Cp	Btu/lb°F	0.24
Specific volume of air at 40°F	v	ft^3/lb	12.59
HDD in the standard year	$HDD_{standard}$	HDD	4,909

Parameters calculated to this point of the analysis are summarized in the Table A-4.4.

Table A-4.4. Calculated Parameters

Parameter	Symbol	Unit	Value
Total annual heat consumption	Q_a	MMBtu	17,991
Total heat cons. during off heating season	Q_{os}	MMBtu	320
Duration of heating season	D_s	days	212
Average daily DHW heat consumption	QI_{av}	MMBtu	2.09
Total annual DHW heat consumption	QI_a	MMBtu	763
Total annual comfort heating and vent. heat cons	QHV	MMBtu	17,228
Norm. total annual comfort heating and vent. heat cons.	QHV_{norm}	MMBtu	18,125

The results of the ventilation heat load calculations are presented in Table A-4.5.

Table A-4.5. Ventilation Heat Load

Average outside temp, °F	Observed normal hours during the	Observed setback hours during the heating season	Ventilation load normal, MMBtu/hr	Ventilation load setback,	Ventilation consumption, MMBtu
T	h_n	h_s	$\dot{Q}V_n$	$\dot{Q}V_s$	QV
-3	-	1	1.57	0.03	0
2	-	8	1.46	0.03	0
7	9	26	1.35	0.03	13
12	22	61	1.25	0.03	29
17	53	116	1.14	0.02	63
22	101	178	1.03	0.02	107
27	164	289	0.92	0.02	157
32	262	435	0.82	0.02	220
37	347	495	0.71	0.01	253
42	362	467	0.60	0.01	222
47	344	427	0.49	0.01	173
52	322	429	0.39	0.00	127
57	330	450	0.28	0.00	93
62	349	478	0.17	-	60
67	371	494	0.06	-	24
72	402	378	-	-	-
77	295	107	-	-	-
82	116	17	-	-	-
87	32	3	-	-	-
92	8	-	-	-	-
97	1	-	-	-	-
Total annual	3,892	4,859			1,542

From Table A-4.5 the peak heating load attributed to ventilation is estimated as 1.27 MMBtu/hr.Then, the annual consumption for comfort heating is calculated by integration of the ventilation heat load by the hours of occurrence during the heating season. Finally, the comfort heating load for the building is calculated and presented in Table A-4.6. From Table A-4.6 the peak heating load for space heating without ventilation is estimated at 8.29 MMBtu/hr.The total heating load calculations are presented in Table A-4.7.

In order to construct an annual heat load duration curve, the hourly comfort heating, ventilation, and DHW and kitchen loads for each instance of occurrence should be added, and sorted out in descending order, as shown in the Table A-4.8. The hourly DHW load is calculated as $QI_{av}/24$.The total annual heat load duration curve for the building is presented in Figure A-4.2.

The above assessment determined the peak heat load of the space heating hot water system as 6.97 MMBtu/hr and the ventilation peak load as 1.27 MMBtu/hr.The total space heating load is 8.24 MMBtu/hr.The average DHW load is 0.11 MMBtu/hr.This information was used to size and select the heat transfer equipment in Section 4 of this report.

Table A-4.6. Total Heating Load

Average outside temp, °F	Observed normal hours during the heating season	Observed setback hours during the heating season	Normal load, %	Setback load, %	Comfort heating load normal, MMBtu/hr	Comfort heating load setback, MMBtu/hr	Comfort heating cons, MMBtu
T	h_n	h_s	$\dot{Q}_n / \dot{Q}_{peak}$	$\dot{Q}_s / \dot{Q}_{peak}$	$\dot{Q}_n$	$\dot{Q}_s$	Q
-3	-	1	126%	107%	10.44	8.90	9
2	-	8	117%	98%	9.67	8.14	65
7	9	26	107%	89%	8.90	7.37	273
12	22	61	98%	80%	8.14	6.60	583
17	53	116	89%	70%	7.37	5.83	1,068
22	101	178	80%	61%	6.60	5.07	1,565
27	164	289	70%	52%	5.83	4.30	2,202
32	262	435	61%	43%	5.07	3.53	2,862
37	347	495	52%	33%	4.30	2.76	2,861
42	362	467	43%	24%	3.53	2.00	2,210
47	344	427	33%	15%	2.76	1.23	1,475
52	322	429	24%	6%	2.00	0.46	841
57	330	450	15%	0%	1.23	-	406
62	349	478	6%	0%	0.46	-	161
67	371	494			-	-	-
72	402	378			-	-	-
77	295	107			-	-	-
82	116	17			-	-	-
87	32	3			-	-	-
92	8	-			-	-	-
97	1	-			-	-	-
total annual	3,892	4,859					16580

Table A-4.7. Annual Heat Load Duration Data

Average outside temp, °F	Observed normal hours during the heating	Observed setback hours during the heating	Total normal heating load, MMBtu/hr	Total setback heat load, MMBtu/hr	Total heating consumption MMBtu
T	h_n	h_s	$\dot{QHV}_{norm\ n}$	$\dot{QHV}_{norm\ s}$	Q
2	-	8	11.13	8.17	65
7	9	26	10.26	7.40	285
12	22	61	9.38	6.63	611
17	53	116	8.51	5.86	1130
22	101	178	7.63	5.09	1676
27	164	289	6.76	4.32	2356
32	262	435	5.88	3.55	3084
37	347	495	5.01	2.78	3112
42	362	467	4.13	2.01	2433
47	344	427	3.26	1.24	1648
52	322	429	2.38	0.46	967
57	330	450	1.51	0.00	498
62	349	478	0.63	-	221
67	371	494	0.06	-	24
72	402	378	-	-	
77	295	107	-	-	
82	116	17	-	-	
87	32	3	-	-	
92	8	-	-	-	
97	1	-	-	-	
Total annual	3,890	4858			18109

Table A-4.8. Annual Load Duration Data

Hourly Load, MMBtu/hr	Bin hours	Cumulative hours
11.22	0	0
10.35	9	9
9.47	22	31
8.60	53	84
7.72	101	185
6.85	164	349
5.97	262	611
5.09	347	958
4.22	362	1320
3.34	344	1664
2.47	322	1986
1.59	330	2316
0.72	349	2665
0.09	371	3036
0.09	402	3438
0.09	295	3733
0.09	116	3849
0.09	32	3881
0.09	8	3889
0.09	178	4067
0.09	289	4356
0.09	435	4791
0.09	349	5140
0.09	495	5635
0.09	467	6102
0.09	427	6529
0.09	371	6900
0.09	429	7329
0.09	450	7779
0.09	981	8760

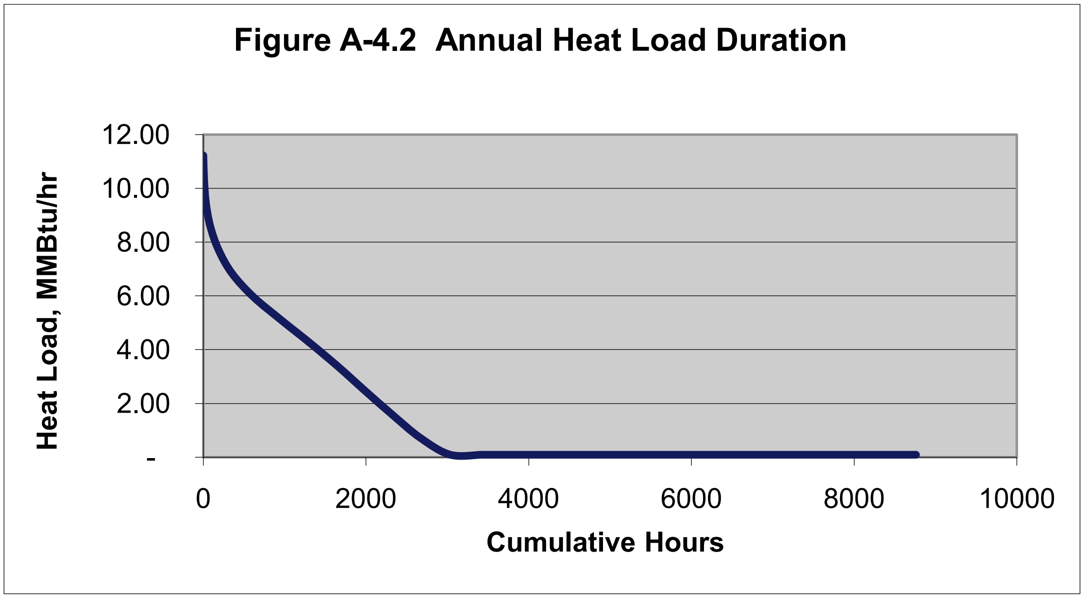


Figure A-4.2 Annual Heat Load Duration
Heat Load, MMBtu/hr
12.00
10.00
8.00
6.00
4.00
2.00
-
0
2000
4000
6000
8000
10000
Cumulative Hours

Section A-5: Assessment of FDs for Other Applications

The FDs can be used for many heat transfer and pumping application. Some of them are as follows:

1. Replacement of surface type heat exchangers for space and district heating.
2. Replacement of electric driven pumps.
3. Waste heat recovery systems.
4. Space cooling applications
5. Power plant feed water systems.
6. Deaeration processes.

Applications 1 and 2 have been evaluated in Sections 2, 3 and 4 of this report. The other applications are briefly discussed below.

Waste Heat Recovery Systems

A diagram of waste heat recovery system with FDs is presented in Figure A-5.1. The FD can use any available steam or hot water as a working media and increase the temperature level and pressure of the waste stream of energy.

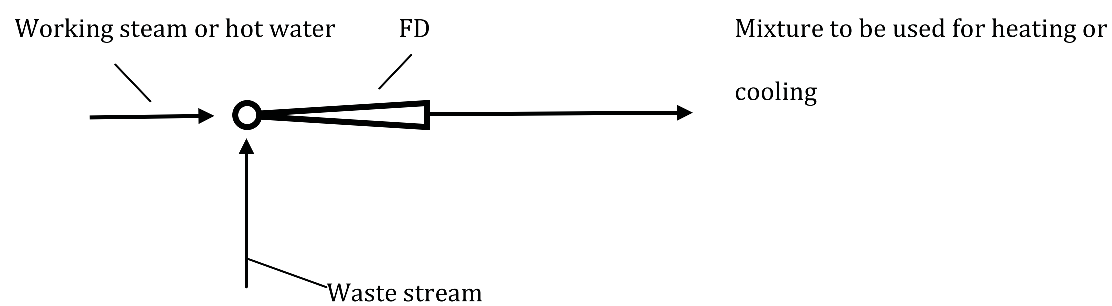


Figure A-5.1. Diagram of Waste Heat Recovery with a FD

Utilization of the Condensate for Space Heating

Con Edison customers with hot water space heating systems can reduce the building steam consumption by using the district steam condensate. The condensate from the street manholes can be pumped by the FD into the space heating system and Afterward discharged into the dilution tank (Figure A-5.2). The payback period for such an application is about two years.

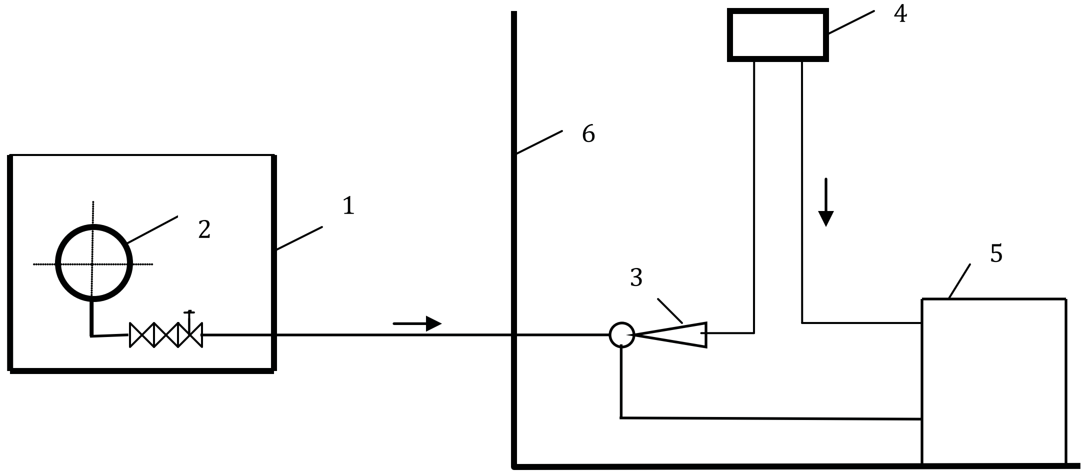


Figure A-5.2 Diagram of Condensate Use in a Space Heating System

1 – Street Manhole; 2 – District Steam Pipe; 3 – Fisonic Device; 4 – Space Heating Load; 5 – Dilution Tank; 6 – Customer Basement Wall

Use of Waste Heat in Absorption Chillers

Con Edison customers equipped with steam driven chillers typically discharge the steam condensate from the chiller's condenser (with a temperature of 210-220F) into a dilution tank without using the energy contained in the condensate. A feasibility evaluation of using the condensate for generation of chilled water has been performed for the 4 Irving Place building (Figure A-5.3). The peak cooling load of the building is 2,400 ton and the annual steam consumption for cooling (see Section A-5) is 16,963 Mlbs. The hourly steam consumption for cooling is: 2400ton x 11lb of steam per ton of cooling = 26,400 lb/hr.In addition the condensate from DHW system with a flow rate of 3,930 lb/hr can also be used.

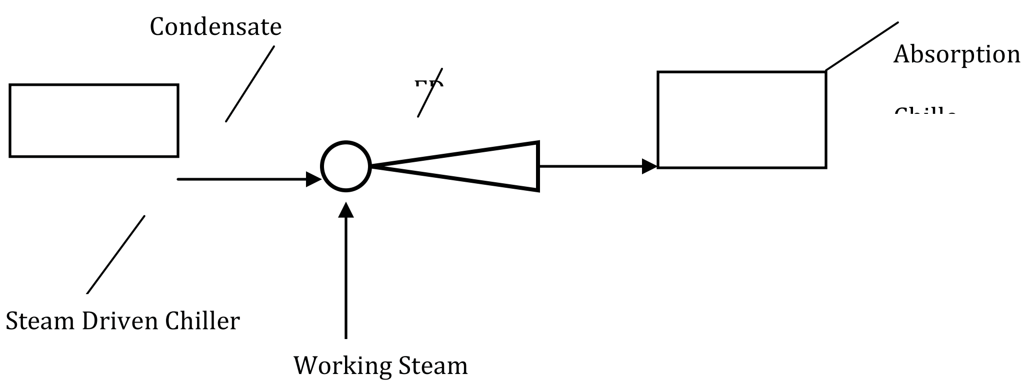


Figure A-5.3. Space Cooling Application with FDs

Discussions with absorption chiller manufacturers have indicated that in a single stage absorption chiller the condensate temperature can be reduced by about 70F with a heat rate for chilled water generation of 15,900 Btu per ton. Therefore the installed capacity of the absorption chiller is estimated as (26,400+3,930) lb/hr x (184.2 – 114.0) Btu/lb / 15,900 Btu/ton = 134 ton. Taking into the account the steam flow rate necessary to operate the FD (about 3 lb/1000 lb of condensate), the total installed capacity of the absorption chiller is estimated as 140 ton. The annual full load operating hours of the steam driven hours is 16,693 Mlbs / (2400 ton x 11 lb/ton) = 632 hours. The potential chilled water generation by the absorption chiller will be 632 hours x 140 ton = 88,480 ton-hours/year. The potential steam savings from the utilization of waste heat of the condensate is estimated as 88,480 ton-hr/yr x (11 – 3) lb/ton = 707,840 lb/yr or 707.8 Mlbs x $23.11/mlbs (Section 2) = $16,357/yr. The capital cost of installation of the absorption chiller is estimated at $210,000 and the simple payback as $210,000/$16,357/yr = 12.8 years, which makes this option not attractive.

The economics of this application can be improved if the condensate from the Con Ed district steam lines after manholes traps could be pumped by a FD into a close-by located building and used in an absorption chiller. In this case no cold water for reducing the condensate temperature to 150F in the manhole will be required.

Use of FDs in Steam Based Power Plants

In this application the surface type feed water heaters of the electric generating plant are replaced with fisonic devices that provide the direct contact heating and pumping of the feed water (Figure A-5.4).

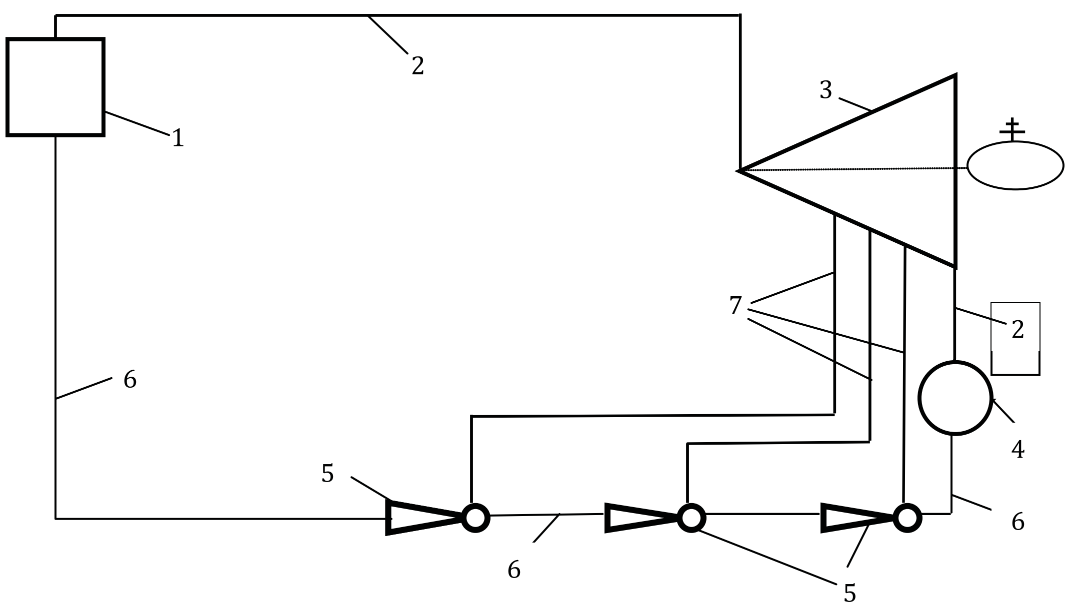


Figure A-5.4. Diagram of a Power Plant with FDs

1 – Steam Generator; 2 – Steam; 3 – Steam-Turbine Generator; 4 – Condenser; 5 – FDs; 6 – Feed Water; 7 – Extraction Steam.

A heat balance diagram of a 250 MW reheat steam turbine is presented in Figure A-5.5. As one can see from this figure the steam turbine is equipped with four closed type low pressure heat exchangers(heaters 4,5,6 and 7) a direct contact deaerating heater and two high pressure closed type heaters. The thermal temperature difference between the steam saturation temperature in the low pressure heaters and the exiting feed water temperature is 5F. The thermal temperature difference results in thermal cycle deficiency. The replacement of the low pressure heaters with FDs offers the opportunity to improve cycle efficiency and reduce the cost of feed water heaters, and reduce the O&M cost. The preliminary comparison of the two systems is presented below.

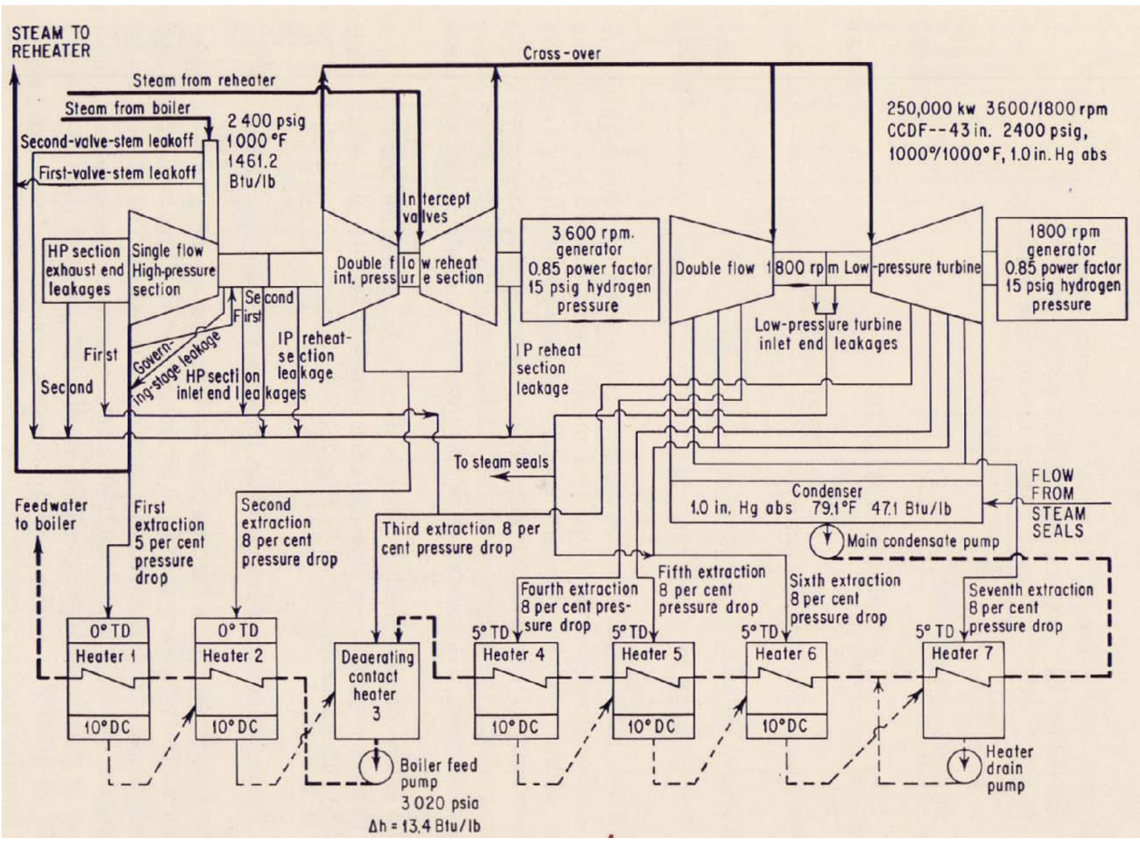


Figure A-5.5. Heat Balance Diagram of a Typical General Electric Reheat Steam Turbine

The net turbine heat rate of the system presented in Figure 4 is 7641 Btu/kWhr (Ref. 8). Use of the same system with direct contact FD feed water heaters allows improving the cycle efficiency by 0.5% or 38.2 Btu/kWhr (Ref. 9). With the current cost of coal of about $4/MMBtu the potential fuel savings of the power plant with FDs versus the conventional plant will be: 38.2 Btu/kWhr *$4.0 MMBtu/kWhr *250,000kW *8760 hr/yr * 0.7 (capacity factor) =$234,242 per year. The capital cost of replacing the closed type feed water heaters with FDs (taking into account the additional devices to protect the turbine from potential water induction) is estimated at $800,000.Taking into account the O&M cost (3% from the capital cost) the simple payback period of replacing the existing system with FDs is about 3.1 years.

Use of FDs for Deaeration

The use of the FD in the deaeration process is presented in Figure A-5.6. In this system the FD is replacing the standard deaeration column reducing the size of the deaerator and increasing the efficiency of the deaeration process.

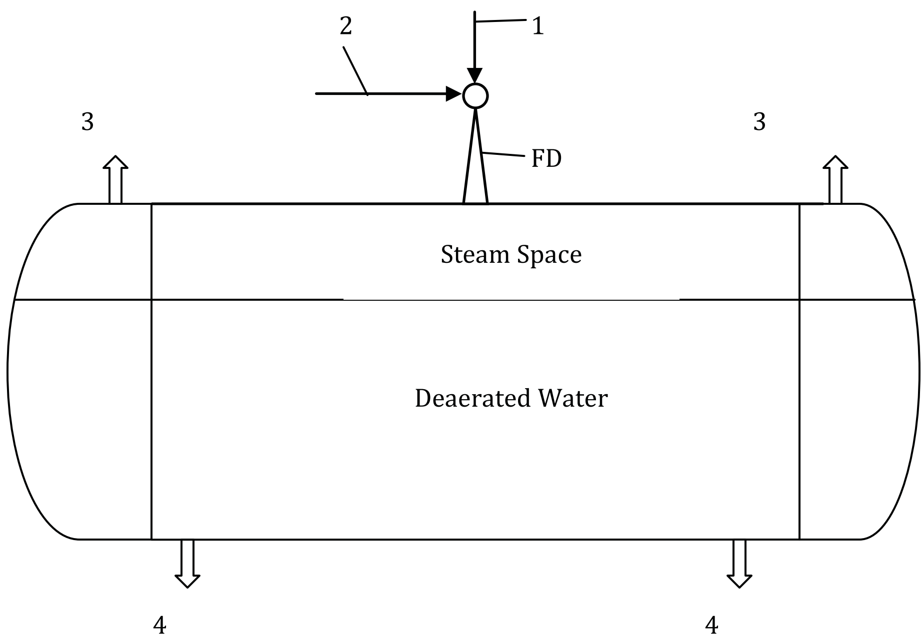


Figure A-5.6. Deaerator with a FD

1 – Working Steam; 2 – Make-up Water and Condensate; 3 – Steam Vapor with Non-condensable gases; 4 – Exit of Deaerated Water.

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Use of Fisonic Devices in Con Edison Service Territory

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