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ABSTRACT

Experiments were conducted to evaluate four techniques to improve the energy efficiency of electrically-heated domestic clothes dryers. Reduced air flow rate and heater input led to energy savings around 8%, while recirculation of a portion of the exhaust air back into the clothes dryer reduced energy consumption by approximately 18%. These two measures are attractive because of their low cost. Two modes of using an air-to-air heat exchanger for heat recovery were considered. The first is to preheat the inlet air with heat from the exhaust air, which resulted in 20 to 26% energy savings. The second mode is 100% recirculation of air through the dryer and a heat exchanger and condensation of water from this air in the heat exchanger by using indoor air as a heat sink. This resulted in 100% heat recovery (i.e., all heat was rejected to indoors) but the energy consumption of the dryer was increased by up to 6%. To maximize energy savings, a clothes dryer with a heat exchanger can be equipped to operate in the preheating mode in the summer and in the recirculation/condensation mode in the winter. The last measure investigated recirculation, through a heat pump (i.e., dehumidifier), also resulted in a 100% heat recovery and, in addition, up to a 33% reduction in dryer energy consumption, but this technique also yielded long drying times.

INTRODUCTION

Approximately 9% of the primary energy used in the household sector is consumed by the laundry process and 30% of this is used in the drying process. The number of clothes dryers in 1978 was 45 million and the ratio of electrical clothes dryers to gas-fired clothes dryers was about 2.5 to 1 (1). Field studies on residential electric clothes drying were performed by Pacific Gas and Electric Company (PG&E) (2) and Oklahoma Gas and Electric Company (OG&E) (3) in 1965 and 1971, respectively. The PG&E survey indicated an annual energy consumption of 1,400 kWh for an average family size of 4.5 people. The OG&E survey indicated an energy demand of 1,320 kWh/year for an average family of 4.8 people. If the energy consumption per load is 3.0 kWh, there will be approximately 450 drying loads per year, on the average. At a typical electricity price of $0.075/kWh, the cost for operating a domestic electric clothes dryer would be around $100 a year.

Several measures have been undertaken by manufacturers to reduce the energy consumption of clothes dryers. In many of the dryers marketed in Europe, the direction of rotation of the drum reverses periodically throughout the drying cycle to prevent clothes from rolling up into a configuration that is difficult to dry. Also available in Europe are clothes dryers that recirculate the exhaust air through an air-to-air heat exchanger (where water is removed by condensation using the indoor air as a heat sink). An advantage of these units is that no exhaust to outdoors is needed. However, if the clothes dryer is operated in a small room, the room air temperature can rise substantially during operation which results in a decrease in the condensation (drying) rate. An inexpensive energy saving measure is to vent the exhaust air to indoors during the winter and a variety of manufacturers market valve systems for this purpose. This technique may be suitable in some situations, but the latent heat in the vented air is not recovered unless moisture condenses on indoor surfaces. Excessive condensation of moisture and high humidity may result especially in air-tight residences. In addition, if a gas-fired dryer is vented to indoors, high indoor concentrations of potentially harmful combustion products (e.g., nitrogen dioxide and carbon monoxide) may result. One other category of energy saving measures is to modify the consumers' behaviors. The high average number of drying loads per year (518) reported in the 1971 study by OG&E suggests that only a small amount of clothes are dried per load. Less frequent drying of larger loads could save significant energy and some homeowners may have already modified their behavior in this manner.

In the remainder of this report, analytical and experimental analyses of various energy saving measures are presented. More detailed information is available in a Lawrence Berkeley Laboratory Report (4).

ANALYTICAL DESCRIPTION OF THE DRYING PROCESS

The following simple balances refer to the schematic of an electrically heated clothes dryer in Figure 1. We assume that the sensible heat change of the clothes, the mechanical power input, and the heat transfer to the surroundings are negligible. A mass balance for the air and water vapor yields

$$\dot{m}_{air}\ (1 + X_{\infty}) + \dot{m}_{H_2O} = \dot{m}_{air}\ (1 + X_{out}). \tag{1}$$

An energy balance for air passing the heating element gives

$$\dot{m}_{air}\ h_{\infty} + P = \dot{m}_{air}\ h_{in}, \tag{2}$$

and an energy balance for the air passing through the dryer drum yields

$$\dot{m}_{air}\ h_{in} + \dot{m}_{H_2O}\ h_{H_2O} = \dot{m}_{air}\ h_{out} + \dot{Q}_{evap.}. \tag{3}$$

DESCRIPTION OF ENERGY SAVING TECHNIQUES

Reduced Air Flow Rate

The drying process in a clothes dryer involves forced convection heat transfer and mass transfer with a phase change. Water is transported from the inside of the wet clothes to their surface by capillary action and diffusion and then into the surrounding hot air. The air removes moisture through convection to the exhaust conduit and provides the heat for evaporation. The rationale behind reducing the air flow rate is to manipulate the kinetics of the drying process so that the increase in evaporating rate, due to a higher air temperature, yields a better energy efficiency despite the reduced convection. Also, since the fresh air is generally drawn from indoors and the exhaust air is vented to outdoors, reduced air flow rate results in a smaller draft in the building and decreased space heating or cooling loads; however, the savings due to a reduced draft are small (4).

Recirculation of Exhaust Air

An easy and inexpensive way to recover exhaust heat is to recirculate a portion of the exhaust air back into the clothes dryer. This procedure increases the average operating temperature and absolute humidity; two effects which counteract each other (i.e., the evaporation rate decreases with the increase of humidity but increases with an increase in temperature). To recirculate exhaust air, modifications to the ductwork are required and more efficient lint filters have to be installed to prevent lint accumulation and subsequent combustion. Current safety codes in some areas do not permit recirculation.

Heat Recovery, Utilizing an Air-to-Air Heat Exchanger

Preheating Mode: One technique to save energy when using an air-to-air heat exchanger is to preheat the inlet air with the exhaust air. To overcome the additional resistance to airflow, an additional or stronger blower is required. Condensation and, thus, latent heat recovery will occur under some operating conditions and the inlet air can be drawn from outdoors, thus decreasing the draft in the building.

The rate of heat transfer in the heat exchanger is

$$\dot{Q} = \epsilon\, C_{min}\, (T_{HS} - T_{CS}) \tag{4}$$

where C_{min} is the smaller of the heat capacities of the two airstreams passing through the heat exchanger, and ϵ is the temperature effectiveness of the heat exchanger. Since the hot air stream exiting the dryer contains more moisture, which can condense, the cold airstream will have a lower heat capacity; thus, $C_{min} = C_c = \dot{m}_c\, Cp_c$ and in the case of no condensation

$$\dot{Q} = \epsilon\, C_c(T_{HS} - T_{CS}) = C_c(T_{CR} - T_{CS}) = C_h(T_{HS} - T_{HR}). \tag{5}$$

For the case where the drying time is unchanged from that in the baseline operating mode (i.e., the same inlet temperature), a new value for the heating element power can be obtained from the equation,

$$P = \dot{m}_{air}\, (Cp_{air} + X_{\infty}\, Cp_{H_2O})\, (T_{in} - T_{CR}) \tag{6}$$

where T_{CR} is calculated from equation (5) for a given temperature effectiveness. Assuming typical operating conditions, energy savings of 30% are predicted.

The savings from using the heat exchanger will be approximately $30 per year. The costs for the heat recovery equipment is estimated to be $100 if the product is mass produced. This includes: a heat exchanger core, one additional blower, additional ductwork, a more efficient lint filter (i.e. a fine nylon mesh), and improved seals around drying chamber and door. With a real discount rate of 6%, the simple payback time will be approximately 4 years.

Recirculation/Condensation Mode: A second arrangement to recover heat with an air-to-air heat exchanger is a system where 100% of the air is recirculated through the dryer and the heat exchanger, and a second open-loop airstream recovers sensible and latent heat by cooling the dryer air and condensing the moisture contained in it. Room air is used for the second airstream and the recovered heat is useful in the winter as space heat. No vent to outdoors is required, all exhaust heat is recovered, and no moisture in the form of vapor is added to indoors. The heat removed from the hot air is approximately

$$\dot{Q}_h = \dot{m}_h (Cp_h + X_{out}\ Cp_{H_2O})\ (T_{HS} - T_{HR}) + \dot{m}_{H_2O}\ h_{fg}. \tag{7}$$

With negligible heat transfer through the case of the heat exchanger

$$\dot{Q}_h = \dot{Q}_c = C_c (T_{CR} - T_{CS}) = \epsilon\ C_{min}\ (T_{HS} - T_{CS}). \tag{8}$$

The two heat recovery modes can be incorporated into a clothes dryer equipped with an air-to-air heat exchanger and a summer/winter switch (Figure 2). The preheating mode is employed in the summer and the recirculation/condensation mode in the winter.

Recirculation With Condensation, Utilizing a Heat Pump

Another method to recover both sensible and latent heat is recirculation of all the warm air in a closed loop through the dryer and a heat pump (Figure 3). No electric heating element is needed. The exhaust air from the dryer passes through the evaporator coil of the heat pump where it cools to below its dewpoint temperature, sensible and latent heat are extracted, and moisture is removed in the form of condensate. The heat is transferred back to the less-moist airstream at a higher temperature in the condenser coil. One problem is that the maximum condenser temperature of existing heat pumps is too low for this application, thus, long drying times result because the compressor cycles on and off due to a high temperature safety control. If higher condenser temperatures around 80 °C are achievable without too low of a coefficient of performance (take COP = 2.5), a heat pump with a capacity of around 3 kW (10,000 Btu/h) could dry a standard load of clothes in 60 minutes. The reduction in dryer energy consumption would be approximately 60% and, in addition, no heat need be vented to outdoors. However, equipment costs, and maintenance and reliability considerations may limit this approach.

EXPERIMENTAL STUDY

The clothes dryer used during tests is a typical model with a power rating of 5400 W at 240 V. The heating element is rated at 5000 W, the electric motor is rated at 190 W (1/4 hp) output, and the air flow rate is given as 82 l/s (173 cfm). The drum volume is 163 l (5.75 ft^3).

The test procedure consisted of measuring total energy consumption, electric heater energy and power, amount of evaporated water, inlet and outlet temperatures, and humidities, temperature of the hot air after the heating element, air flow rates, and the drying cycle time including heating time and cool down time. The drying cycle was terminated manually. The evaporation rate was obtained by calculating the ratio of the amount of evaporated water (determined gravimetrically) to the overall drying time. Tests were conducted with a standard load of clothes containing 50% synthetic and 50% cotton fibers that weighed 3.18 kg (7 lb) when dry. They were wetted to a moisture content of 70% of the dry weight prior to each test. During the drying cycle, the moisture was reduced to approximately 3-5% of the bone-dry weight (1). During all tests a plastic sheet envelope was attached around the dryer housing to simulate tighter seals at dryer drum and door and a fine nylon mesh filter was installed in addition to the standard lint filter.

RESULTS

Table 1 describes the various tests and compares the energy requirements and the drying times of different energy saving measures. Figure 4 relates graphically the energy consumed during the various test modes.

Baseline Experiments

The baseline tests simulated operation in both the heated and unheated sections of a house and included tests with reduced heater power. The air flow rate was approximately 57 l/s (120 cfm) which is significantly less than the 82 l/s (173 cfm) reported by the manufacturer despite the fact that the test system was not substantially more restrictive to flow than a typical installation. The total drying time was approximately 40 min for the tests with 230 V at the heater (P_{heater} = 4.55 kW, ambient temperature T_{∞} = 20 °C), which included a 2 min cool down time, (i.e., the cool down time was the period of time before termination of the cycle, during which the heater was off). The tests with a lower inlet temperature (T_{∞} = 10 °C), simulating dryer operation in the unheated section of the house, lasted approximately 45 min and consumed about 13% more electricity. For tests with a reduced heater input (190 to 200 V, P_{heater} = 3.1 to 3.45 kW), drying times increased to 52-60 min with insignificant energy savings.

Experiments With Reduced Air Flow Rate

In tests with reduced air flow rates and electric heater inputs, drying times were 51-59 min compared to 40 min for the baseline. Energy savings of approximately 8% were achieved. Approximately 5% energy savings occurred when the air flow rate was reduced from 57 l/s (120 cfm) to approximately 39 l/s (83 cfm) and the heater power was unchanged.

Recirculation Experiments

For tests with recirculation, the voltage at the heating element was maintained at 230 V, and the air flow rate through the drying chamber was approximately 57 l/s (120 cfm). The recirculation ratio, defined as the amount of recirculated air divided by the total air flow rate, was 49 to 72%. Due to the use of a second blower upstream of the dryer, a slight pressurization of the drying chamber resulted. A mass balance calculation, indicated that approximately 8% of the evaporated moisture had leaked into the surrounding airspace.

The drying time for these tests was 43 ± 1.5 min, which included a cool down time of 5 min. This drying time is not significantly greater than the drying time for the baseline tests. Energy savings were 10.4 to 18.5% depending on the recirculation ratio. The optimum recirculation ratio, considering drying time and operating temperatures, was around 67%.

The cost to retrofit for recirculation a residential clothes dryer with an accessible air inlet is estimated to be about $30 if the labor is performed by the homeowner. (Only some ductwork and an improved lint filter are required.)

The simple payback time, with a real discount rate of 6% and yearly energy savings of $18, is a little less than two years.

Heat Recovery Experiments

The counterflow heat exchanger (5) used in these tests has a heat transfer area of 10.7 m^2 (115 ft^2), an effectiveness of about 70%, and a pressure drop of approximately 65 Pa (0.26" H_2O) at a flow rate of 57 l/s (120 cfm). An additional fan upstream of the heat exchanger provided equal flow rates at the inlet and outlet of the dryer (i.e., minimized leakage at the dryer drum).

Preheating Mode: For tests in the preheating mode, fresh air with a temperature of approximately 10 °C was preheated with the exhaust air. The voltage at the electric heater was 230 V, and the air flow rate was approximately 55 l/s (116 cfm). The drying time was 39 to 40 min including a 4 to 5 min cool down time. The energy savings when comparing with baseline tests with T_∞ = 10 °C were about 26%. The heat exchanger effectiveness was approximately 60%, which is less than indicated previously (5).

Recirculation/Condensation Mode: For tests in the recirculation /condensation mode, the heater voltage was 190 to 230 V, the air flow rate through the dryer and heat exchanger was approximately 57 l/s (120 cfm) and the flow rate of the of the air that passed in an open loop through the heat exchanger was approximately 60 l/s (126 cfm). The drying time varied from 60.5 to 65.0 min and the energy consumption was 1.4 to 6.4% higher than the baseline energy consumption, depending on the electric heater power. Approximately 25% of the evaporated water escaped to the surroundings due to the fact that the drying chamber was slightly pressurized. The heat exchanger effectiveness ranged from 70 to 80% -- higher than in the preheating mode due to increased latent heat recovery. The indoor air that was used as a heat sink was returned back to indoors as dry warm air at a temperature of 45 to 49 °C.

Based upon the results, a brief analysis of the economics of a clothes dryer equipped with an air-to-air heat exchanger and a summer/winter switch (preheating mode in the summer and recirculation/condensation mode in the winter) is presented. We assume, the clothes dryer is located in an electrically heated house, the heating season lasts 5 months, and heat losses from the clothes dryer housing to the surroundings are negligible. Furthermore, the increase in energy consumption during the recirculation/condensation mode is assumed negligible. The additional cost is estimated to be $100 if the system is mass produced. With 25% heat recovery in the summer and 100% heat recovery in the winter, the savings will be approximately $56/year. Assuming a real discount rate of 6%, the simple payback time will be approximately 2 years.

Experiments with a Heat Pump

The heat pump used for the experiments is a residential dehumidifier with a rated capacity of 16.6 l/day (35 pints/day) at standard test conditions (T_∞ = 80 °F, RH_∞ = 60%). This rated capacity corresponds to a refrigeration load of approximately 1.9 kW (6500 Btu/h). The refrigerant used is R500.

Four tests were performed, one with the dehumidifier alone and three with a dehumidifier and an air-to-air heat exchanger in series. The heat exchanger was positioned upstream of the dehumidifier and its function was to remove part of the sensible heat contained in the exhaust air. The average power input of the compressor was 650 to 850 W, the flow rate through the dryer was about 61 l/s (130 cfm) for the tests with a heat exchanger and 85 l/s (180 cfm) when the heat exchanger was not used. The flow rate of the cooling air through the heat exchanger ranged from 17 to 53 l/s (36 to 113 cfm). In the test without a heat exchanger, the compressor cycled on and off due to thermal overloading, and the drying time was 150 min. For the other tests, the flow rate of the cooling air was adjusted so that no or very little cycling occurred which reduced the drying time to 145 to 120 min. The test with no heat exchanger consumed about 33% less energy than the baseline, although it lasted approximately 3.75 times as long. With the additional heat exchanger and a low flow rate of cooling air, energy savings were about 29% and the drying time was 3 times as long as for the baseline. These tests indicate a significant potential for energy savings, however, the dehumidifier would have to be redesigned so that it will operate with higher condenser temperatures.

SUMMARY

The four basic energy saving measures that were studied experimentally are: (1) reduced air flow rate; (2) recirculation of exhaust air; (3) heat recovery, utilizing an air-to-air heat exchanger; and (4) recirculation with condensation, utilizing a heat pump.

Baseline tests indicated that the actual air flow rate through the dryer was smaller than reported by the manufacturer. About half of the heat contained in the exhaust air appeared as sensible heat. Approximately 8% energy savings were achieved by reducing the air flow rate and heater power.

Recirculation of exhaust air resulted in a 10-18% decrease in energy consumption, depending on the recirculation ratio. The optimal recirculation ratio was around 67%. The costs to retrofit for recirculation are low and payback times can be as short as two years if the labor is performed by the homeowner. This energy saving measure could also be incorporated into new equipment by the manufacturers.

A 26% reduction in energy consumption was achieved with heat recovery, utilizing an air-to-air heat exchanger in the preheating mode. High equipment costs (approximately $100) yield an estimated payback time of four years for this energy saving measure. However, 100% heat recovery can be achieved by using an air-to-air heat exchanger, when the exhaust air is 100% recirculated and the moisture condensed, using indoor air as a heat sink. The drying times in this mode were approximately 50% higher than the baseline drying times. For a clothes dryer equipped with an air-to-air heat exchanger and designed to operate in the preheating mode in the summer and the recirculation/condensation mode in the winter, the estimated payback time is approximately 2 years in an electrically heated house.

The last experiments were performed using a heat pump. A residential dehumidifier was coupled to a clothes dryer and this integration resulted again in a closed system with 100% heat recovery and a 33% reduction in dryer energy consumption. However, the drying time is unacceptably long with currently available heat pumps.

Summarizing, it can be said that significant energy savings could result from the various energy saving measures investigated and several of the measures may be cost effective. However, in addition to further laboratory experiments, a better understanding of the cost, marketability, reliability and consumer acceptance of various measures is desirable.

ACKNOWLEDGEMENTS

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NOMENCLATURE

Symbols

C	= capacity rate	P	= power
C_p	= specific heat at constant pressure	$\dot{Q}$	= energy transfer rate
h	= enthalpy	RH	= relative humidity
h_{fg}	= heat of evaporation	T	= temperature
$\dot{m}$	= mass flow rate	$\dot{V}$	= air flow rate
$\dot{m}_{H_2O}$	= evaporation or condensation rate	X	= absolute humidity
		ϵ	= heat exchanger effectiveness

Subscripts

CR = cold return	HR = hot return	in = inlet
CS = cold supply	HS = hot supply	out = outlet
c = cold	h = hot	∞ = ambient space

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Table 1. Comparison of Test Conditions, Energy Savings, and Drying Times

Test Mode (Number of Tests)	Test Conditions	Specific Energy Consumption (kWh/kgH_2O)	Energy Savings (%)	Drying Time (min)
Baseline 1 (5 tests)	$P_{heater} = 4.55$ kW $P_{fan} = 0.25$ kw $\dot{V} = 56$ ℓ/s $T_\infty = 20^oC$	1.444	-	40
Baseline 2 (2 tests)	as above except $T_\infty = 10^oC$	1.628	-12.7	45
Baseline 3 (3 tests)	$P_{heater} = 3.1$-3.45 kW $P_{fan} = 0.25$ kW $\dot{V} = 56$ ℓ/s $T_\infty = 20^oC$	1.431	0.9	52-60
Reduced Air Flow Rate & Heater Input (5 tests)	$P_{heater} = 3.08$-3.43 kW $P_{fan} = 0.25$ kW $\dot{V} = 29$-40 ℓ/s $T_\infty = 20^oC$	1.324	8.3	51-59
Recirculation (5 tests)	$P_{heater} = 4.5$ kW $P_{fan} = 0.25$ kW $\dot{V} = 55$ ℓ/s R = 49-72% $T_\infty = 20^oC$	1.227 (1.294 - 1.177)	15.0 (10.4 - 18.5)	43 43
HX* - Preheat (4 tests)	$P_{heater} = 4.5$ kW $P_{fan} = 0.35$ kW $\dot{V} = 55$ ℓ/s $T_{inlet} = 7$-11^oC	1.197	26.5**	39
HX - Recirculation 1 (4 tests)	$P_{heater} = 4.55$ kW $P_{fan} = 0.35$ kW $\dot{V} = 57$ ℓ/s $T_\infty = 20^oC$	1.537	-6.4***	60.5
HX - Recirculation 2 (3 tests)	$P_{heater} = 3.15$-3.5 kW $P_{fan} = 0.35$ kW $\dot{V} = 57$ ℓ/s $T_\infty = 20^oC$	1.464	-1.4***	63
Dehumidifier (1 test)	$P_{compr} = 0.837$ kW $P_{fan} = 0.3$ kW $\dot{V} = 85$ ℓ/s $T_\infty = 20^oC$	0.970	32.8***	150
Dehumidifier / HX	$P_{compr} = 0.655$-0.745 kW $P_{fan} = 0.4$ kW $\dot{V}_{RC} = 61$ ℓ/s $\dot{V}_{open} = 17$-53 ℓ/s $T_\infty = 20^oC$	1.024-1.215	15.9-29.1***	120-145

* HX = heat exchanger.

** Compared to Baseline 2. Energy savings are estimated to be 20% for $T_\infty = 20^oC$.

*** All energy consumed by the dryer can be added to the indoor air to reduce heating loads.

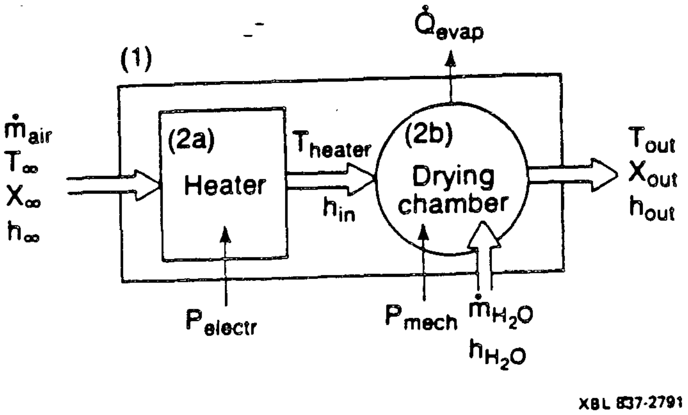
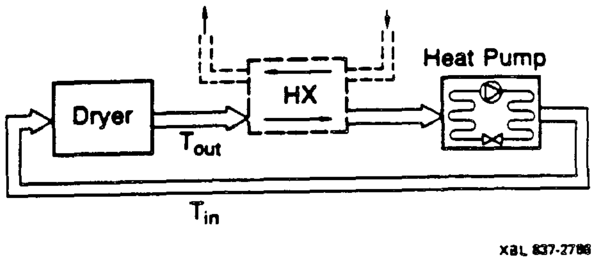


Figure 1 (left)- Schematic diagram of clothes dryer.

Figure 2- Schematic diagram of clothes dryer with an air-to-air heat exchanger and summer/winter switch. The preheating mode is employed in the summer and the recirculation/condensation mode in the winter.



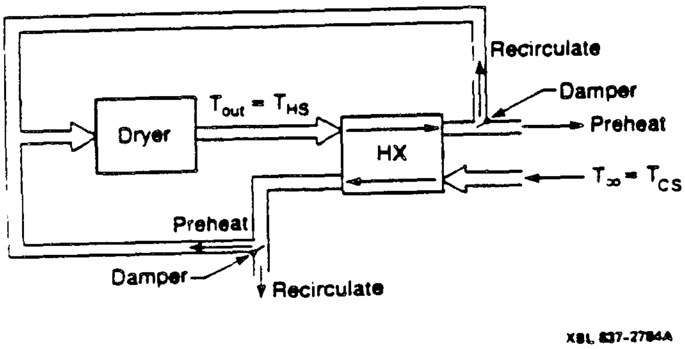


Figure 3 (above)- Schematic diagram of a clothes dryer with a heat pump and an optional air-to-air heat exchanger.

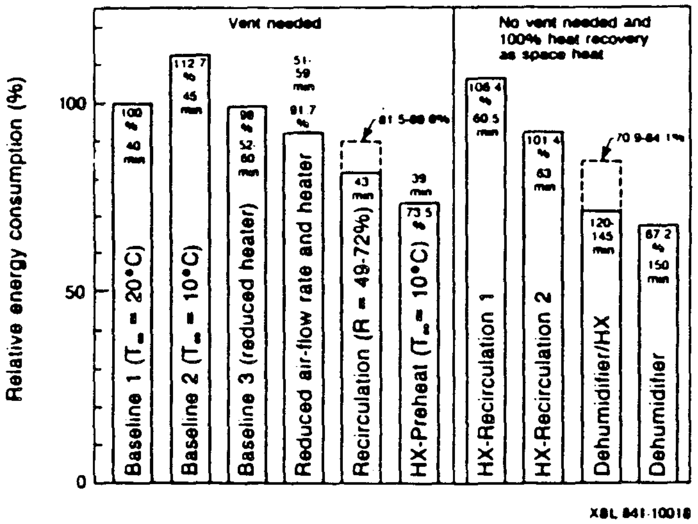


Figure 4- Comparison of experimentally determined energy requirements and drying times.