Heat Pump/Thermal Storage in Combinations for Enhancing Passive Solar Heating of Canadian Houses

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EXECUTIVE SUMMARY

With the increased use of large areas of south facing, high R-value glazings, passive solar gains can play an increased role in supplanting domestic heating loads. At the same time, comfort control must be addressed to avoid overheating.

Traditional outdoor air source heat pumps can provide heating and cooling but have poor heating performance at cold outdoor temperatures and do not utilize waste heat generated indoors in summer and winter (i.e. from solar gains, exhaust air and grey water recovery and space cooling). However, heat pump technology can supply heating loads at high efficiencies as long sufffficient waste heat sources (at 0^{o}C or higher) are available. In a passive solar residence, occupant activities, ventilation and solar gains provide ample low grade heat sources, but are typically not coincident with building loads. Thermal storage strategies can ensure that the waste heat is captured and stored for supply to eventual space and hot water requirements.

Heat-pump-based integrated mechanical systems combined with thermal storage have the potential to both increase solar utilization and provide adequate space cooling. Usually offered as a single package, integrated mechanical systems combine a number of otherwise discrete pieces of equipment. Functions that are integrated include space heating, space cooling, ventilation exhaust air heat recovery, and domestic water heating. The benefits are reduced initial capital costs and installation labour, and increased energy efficiency.

This study examines five residential-scale heat pump/thermal storage combinations designed to increase in passive solar gain utilization and economic feasibility. Using a modified version of the detailed hour-by-hour computer program, ENERPASS, these systems were modelled for thermal performance. Thermal storage modelling included a large hot water tank, a reversible airflow rock store and ice phase-change storage, both as an indoor tank and an exterior ground storage.

The house used for modelling is a 200 m^2 two-storey dwelling. South-glazing-to-interior-floor-area ratios (GFR) of 7% to 20% were examined. Three thermal envelopes were defined: a "NBC" house insulated at the National Building Code levels; an Energy-Efficient (EE) house with higher insulation levels, significantly improved windows and basement slab insulation; and a Super-Energy-Efficient (SEE) house with a further increase in insulation levels, RSI 1.25 windows and an attached sunspace. The attached sunspace is located on the first floor of the south facade and constitutes 50% of the south-facing glazing.

Of the systems studied, the most cost-effective heat pump/thermal storage configurations are fully integrated mechanical systems with either isolated ice storage or with phase-change ground storage. The cost effectiveness is slightly better than HRV's on a payback basis (less than 3 years) but the cost savings realized after 5 years (including incremental capital costs) are more than 2 times as large. These systems extract heat from ventilation air, grey water, and ground, as well as utilizing excess solar gains. As such, their performance is not highly affected by varying the solar aperture.

On the other hand, they are able to control overheating with glazing-to-floor ratios up to 20%, thereby allowing the designer significant architectural freedom.

The payback of a simple DHW heat pump extracting heat from space to offset water heating used in combination with a heat recovery ventilator has a payback of just over 5 years. This is a low-cost package yielding significant ventilation and water heating energy savings ($500 per year) while providing partial air conditioning. This combination may be adequate for many houses, particularly if the site is exposed and conscientious venting and some solar shading is practiced in summer.

There seems to be promise in developing an improved air conditioning heat pump with large warm water storage that also has a ventilation heat recovery function. If the cost of such a system is similar to other systems studied, a payback of about 4 years would be achieved.

The total 5-year cost savings (incl. incremental costs) for the three more promising configurations when compared to conventional systems are as follows (electricity $16/GJ, gas $10/GJ)

- DHW heat pump w/HRV: $1100 (elec. base case only)
- Integrated Mechanical System: $1600-2400
- Int. Mech. System w/ground storage: $2000-2600

Calculation procedures for the energy benefits were developed in terms of manufacturers' specifications under typical operating conditions. The energy benefits are subtracted from annual total (space + DHW) heating consumption established from thermal modelling with a conventional mechanical system.

Preheating of ventilation air by drawing it via the one-storey attached sunspace used in this study contributes an equivalent heat recovery effectiveness of 30% or typically 9 GJ or $145. This is additional to any exhaust air heat recovery that may be realized by an exhaust-only ventilating heat pump.

The most interesting result of this study is the phase-change ground storage modelling of a volume of saturated soil or man-made aquifer in contact with surrounding soil. Computer simulations yielded optimized annual storage capacities amounting to only 8% of the annual total (space + DHW) heating loads. This requires significantly less yard area for locating the storage, and heat exchange may be achieved simply by burying 1 or 2 large solar pool collectors or a ground heat exchanger coil similar to the helical direct expansion coil developed at NRC.

The computer modelling confirmed that improved window technology and increased solar gains allow architectural freedom to design large south facing window areas without an energy penalty. Several heat pump/thermal storage combinations are able to control summer and winter overheating while making use of increased solar gains. When combined with other gains, such as ventilation heat recovery, grey water heat recovery and ground heat extraction, they can yield impressive cost feasibility. Field trials of these systems, in particular the artificial aquifer ground storage, should be pursued. Concurrently, work should be pursued on the development of a standard procedure for rating these types of systems and determining their energy credits.

RÉSUMÉ

Le recours accru aux grands vitrages à facteur d'isolation élevé sur la face sud des bâtiments permet aux systèmes solaires passifs de faire un apport considérable au chauffage domestique. Il faut cependent contrôler l'ambiance afin d'éviter la surchauffe.

Les thermopompes à air extérieur ordinaires chauffent et refroidissent, mais leur rendement en mode chauffage est faible par temps froid et elles ne mettent pas à contribution la chaleur résiduelle qui provient de l'intérieur, en hiver comme en été (apport solaire, air usé, eaux ménagères et climatiseur). Toutefois, l'on offre actuellement des thermopompes capables d'assurer le chauffage des locaux très efficacement si les sources de chaleur disponibles suffisent à la tâche (0° ou plus). Dans un bâtiment à système solaire passif, l'activité humaine, la ventilation et l'apport solaire constituent autant de sources abondantes de chaleur de faible intensité, mais sont rarement à la hauteur de la charge de chauffage du bâtiment. En recourant à diverses techniques de stockage thermique, ou peut capter la chaleur et la conserver, puis l'utiliser plus tard pour chauffer les locaux ou l'eau.

Les systèmes domestiques intégrés à thermopompe et stockage thermique permettent de mieux exploiter l'énergie solaire aussi de refroidir adéquatement les locaux. Ces systèmes, généralement offerts sous forme d'ensembles, réunissent plusieurs éléments autonomes. Les fonctions suivantes sont comprises : refroidissement et chauffage des locaux, ventilation, récupération de chaleur à même l'air usé et chauffage de l'eau domestique. Ces systèmes sont moins chers à l'achat, moins chers à installer sur le plan de la main d'oeuvre et offrent un rendement énergétique supérieur.

La présente étude porte sur cinq systèmes combinés à thermopompe et stockage thermique de format résidentiel conçus pour tirer meilleur parti des apports solaires passifs et pour réaliser une rentabilité accrue. Ils ont été conçus pour maximiser le rendement thermique à l'aide d'une version modifiée du logiciel horaire détaillé ENERPASS. A l'étape de la modélisation, on a inclus un grand réservoir d'eau chaude, le stockage sur galets à circulation d'air réversible et le stockage souterrain sur glace à changement de phase, sous forme de réservoir intérieur et de stockage géothermique externe.

L'habitation-type utilisée aux fins de la modélisation était un bâtiment résidentiel de 200 m^2 à deux étages. L'on a testé des rapports de 7 % à 20 % entre la surfae de plancher et celle des vitrages orientés vers le sud. Trois enveloppes thermiques ont été définies : une habitation "CNB" (isolée selon les normes du Code national du bâtiment), une habitation à haut rendement énergétique beaucoup mieux isolée au niveau des fenêtres et de la dalle de fondation, et enfin une maison à haut rendement énergétique à fenêtres RSI 1,25 et aire ensoleillée intégrée. L'aire en question est au premier étage, du côté sud, et comporte

la moitié des surfaces vitrées exposées au sud.

Parmi les systèmes étudiés, les combinaisons thermopompe-stockage thermique les plus efficaces sont les systèmes intégrés à stockage isolé sur glace ou stockage souterrain à changement de phase. Ils sont légèrement plus rentables que les ventilateurs à récupération de chaleur sur la base du délai de recouvrement (moins de trois ans), mais les économies directes qu'ils permettent, sur cinq ans (les coûts en capital différentiels étant compris) sont plus de deux fois plus grandes. Ces systèmes puisent la chaleur à même l'air de ventilation, les eaux ménagères et le sol, tout en utilisant les apports solaires excédentaires. C'est pourquoi leur rendement n'est pas très affecté par la variation des surfaces de captage solaire. D'autre part, ils permettent de limiter la surchauffe même lorsque le rapport vitrage-surface de plancher atteint 20 %, ce qui donne plus de latitude pour ce qui est de l'architecture.

Il est possible de récupérer en un peu plus de cinq ans le coût d'un simple système de chauffage de l'eau domestique à thermopompe puisant la chaleur dans les locaux pour compenser la perte au chauffe-eau, avec ventilateur récupérateur. Il s'agit d'un ensemble à prix modique qui permet de réaliser des économies sensibles au plan de la ventilation et du chauffage de l'eau (500 $ par an), en plus d'aider à refroidir l'air. Ce système pourrait suffire à bien des habitations, surtout celles qui sont implantées sur des sites exposés et si les occupants veillent à la ventilation et à un certain contrôle de l'ensoleillement en été.

Il semblerait utile de mettre au point un climatiseur de modèle amélioré à thermopompe doté d'un grand réservoir d'eau tiède capable d'assurer une partie de la fonction de récupération de chaleur à même la ventilation. Si le coût de ce genre était à peu près comparable à celui des autres, on pourrait s'attendre à ce qu'il se rentabilise en quatre ans environ.

Les économies totales possibles sur cinq ans (y compris coûts différentiels) que permettraient les trois systèmes les plus prometteurs, comparativement aux systèmes traditionnels, s'établissement comme suit (électricité à 16 $ le gigajoule, gaz naturel à 10 $ le gigajoule) :

- chauffe-eau domestique avec VRC : 1 100 $ (scénario de base électricité seulement)
- système domestique intégré : 1 600 $ - 2 400 $
- système intégré avec stockage souterrain : 2 000 $ - 2 600 $

La méthodologie de calcul des avantages énergétiques a été établie d'après les fiches techniques des fabricants, en fonction de conditions d'exploitation typiques. Les économies d'énergie sous soustraites de la consommation annuelle totale de chaleur (chauffage des locaux et de l'eau domestique) établie à partir de la modélisation thermique fondée sur un système domestique traditionnel.

Dabs le cas étudié, le préchauffage de l'air de ventilation par séjour dans l'aire ensoleillée intégrée d'un étage de haut fait un apport de 30 % (en général 9 GJ ou 145 $) à l'efficacité de la récupération de chaleur, ceci en sus de la récupération de l'air d'échappement que permettrait une thermopompe consacrée exclusivement à cet usage.

La composante la plus intéressante de cette étude à été le modélisation du stockage souterrain à changement de phase d'une quantité de terre saturée ou d'une nappe aquifère artificielle en contact avec le terrain environnant. D'après les simulations numériques, les capacités de stockage annuelles optimisées de tels systèmes n'atteindraient que 8 % de la charge de chauffage annuelle totale (chauffage des locaux et de l'eau domestique). Cette méthode de stockage occupe sensiblement moins d'espace sur le terrain et permet d'opérer l'échange de chaleur en aménagement simplement un ou deux grands réservoirs solaires souterrains ou un serpentin souterrain semblable à l'appareil hélicoïdal à expansion directe mis au point par le CNRC.

La modélisation a confirmé que le perfectionnement technologique des fenêtres et l'accroissement de l'apport solaire permettent à l'architecte de concevoir de grandes surfaces vitrées orientées vers le sud sans compromettre le rendement énergétique. Plusieurs groupes thermopompe-stockage thermique peuvent régulariser la surchauffe, été comme hiver, tout en permettant d'exploiter davantage les apports solaires. Ces systèmes, lorsqu'ils s'ajoutent aux autres sources de chaleur comme la récupération au niveau de la ventilation, des eaux ménagères et du sol, deviennent fort rentables. Les essais de ces systèmes dans le réel, surtout l'essai du stockage en nappe aquifère artificielle, devraient continuer. Il conviendrait en même temps de poursuivre l'élaboration d'une procédure normalisée d'évaluation de ce genre de système et de détermination de leur capacité d'économie d'énergie.

1.0 BACKGROUND

As house envelopes become more energy-efficient and high R-value glazings make significant inroads in residential construction, the potential for improved passive solar strategies may be great, yet remains largely unexplored. While house designs motivated strictly by passive solar features are relatively few in number, designers and homeowners are generally intrigued by large, sunbathed expanses of glass, as evidenced by the significant numbers of houses that are being built with areas of 20 m^2 or more (see Fig. 1). Increasingly this glazing will consist of high R-value glazing (i.e. RSI 0.6 or better), thus retaining larger solar gains. Combining this with reduced space heating demands, the frequency with which excess gains occur will increase. Typically, the choice is to dump these unutilized gains or to sacrifice comfort by allowing large excursions of indoor temperature.

To increase utilization of excess gains designers have traditionally tried to incorporate heavy construction materials within the building structure. These include masonry walls, multiple layers of drywall or water columns. Remote storage in the form of air-coupled rockstores and large water tanks has also been employed. Such storage strategies, unless exposed to radiation in direct gain areas, rely on sensible temperature excursions of the house air to deliver energy to the mass with resulting comfort problems. This conundrum makes phase-change materials attractive which promise to store large amounts of energy at room temperature. However, high cost and stability of the phase change materials have impeded their use on any significant scale. At present, phase-change drywall (i.e standard drywall soaked with fatty acids) shows good promise when applied to the south zone of a house. However, its capacity to deal with large solar gains is limited by the surface area of installed drywall and its storage performance.

On the mechanical system side, passive solar and low energy housing has spurred the development of energy-efficient mechanical ventilation systems, primarily heat recovery ventilators. More recently, exhaust air heat pumps have become available which chill stale outgoing house air thus recovering energy, primarily for the purpose of supplanting domestic hot water energy consumption. Increased airtightness in new house construction has prompted the housing industry to take a close look at ventilation requirements with a real possibility of mandating detailed mechanical ventilation standards in the near future. This scenario is exerting pressure on the HVAC industry to develop new, integrated and energy-efficient ventilation equipment, some of which will include heat pump technology.

Ground source heat pumps have established themselves in residential construction, supplying space heating and cooling and often hot water heating. Even though costs of installing ground heat exchange piping are continuing to drop, and innovative, low-cost ground heat exchangers are being developed (e.g. at NRC), the system cost is still high and installation is usually restricted to large custom homes. Passive solar low energy houses have the potential, however, to significantly reduce the required peak heating demand, thus reducing ground piping costs substantially.

By marrying the energy-efficient features of heat pumps and mechanically-coupled isolated storage, integration of all the thermal functions in the home (space heating/cooling, water heating, ventilation with heat recovery and grey water heat recovery) can realize new levels of energy efficiency at lower cost over individually purchased components.

The purpose of this study is to explore the concept of integrating heat pumps with a variety of thermal storage configurations, thereby extending temperature differentials of sensible storage or utilizing water as a low-cost phase change material. A primary motivation is to increase the passive solar contribution towards space heating and DHW loads and the utilization of the transmitted solar gains as the aperture area increases. The resulting benefits are increased energy efficiency and greater cost effectiveness.

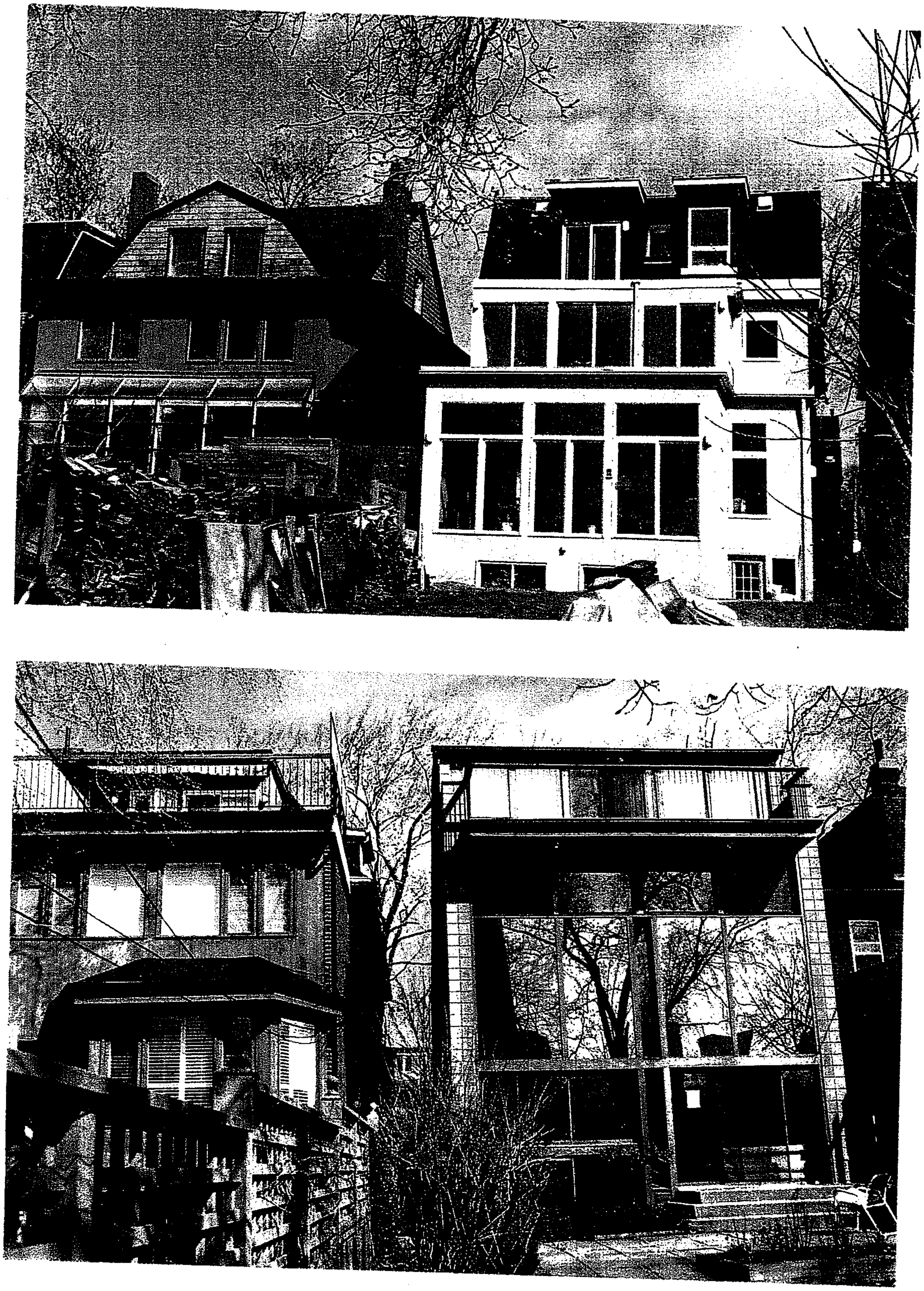


Figure 1. Typical Houses with Large Amounts of South-facing Glazing

2.0 APPROACH

After reviewing the available literature on heat pumps and thermal storage, a systematic method of defining candidate systems was chosen. A generic heat pump system schematic was constructed (see Fig. 2) with thermal storage on both the heat extraction (evaporator) and heat delivery (condenser) side, such that all model systems in the study are subsets of this main generic model. Both the cold and warm thermal storage could be sensible or phase-change storage media. After examining all the possible combinations with respect to capital costs and potential energy benefits, more concrete operational schematics were developed with defined heat pump performance, storage types and control regimes. These operational models form the basis of the computer modelling.

Five model systems, described below, were simulated in a passive solar house with three levels of energy efficiency and different ratios of south-facing glazing. The computer simulation program, ENERPASS (Ref. 1), was modified with several new thermal models, such as a ground storage model with ice phase-change, a reversible airflow rockstore and an interposed mass wall between a sunspace and the main living space. Some other models, e.g. a DHW preheat tank, DHW heat pumps and an integrated mechanical system with isolated ice phase-change storage, were already available within ENERPASS from previous work (Ref. 2 and 5). The system configurations were initially evaluated for Ottawa climate data and systems with promising energy cost performance are validated for Winnipeg and Vancouver.

From the results of the computer modelling, energy cost savings were compared with estimated incremental system capital costs to establish the economic feasibility of the systems. The extensive simulation work enables procedures for sizing the thermal storage and heat pump capacity to be developed.

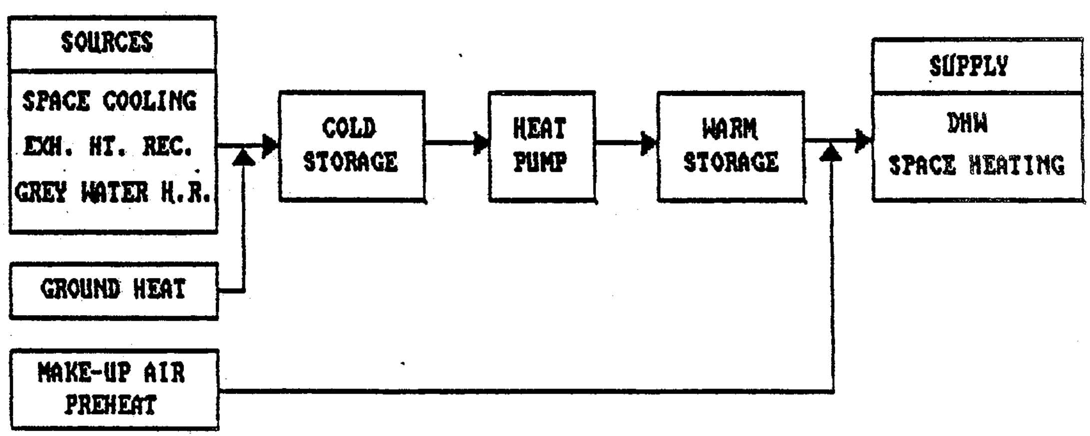


Figure 2 GENERIC HEAT PUMP/THERMAL STORAGE MODEL

The following five systems were chosen as promising for in-depth study.

2.1 System 1: Indoor Air Heat Pump/DHW Supply (DHW/HP)

This heat pump consists of an air-source evaporator and a refrigerant/water coaxial condenser (see Fig. 3A). It has a nominal output of 3 kW at a Coefficient of Performance (C.O.P.) of 3.5 (i.e. 2.5 units of cooling + 1 unit of electrical input supplies 3.5 units of heating output). The evaporator extracts heat from space air in the south living zone or sunspace when excess gains can be utilized (i.e. above 23°C in the living zone, above 15°C in the sunspace). The energy extracted from space air is transferred via the heat pump to a 250 L water tank which serves to preheat the mains inlet water for the DHW tank. Previous studies of DHW heat pumps have been done by Enermodal (Ref. 2) and the author (Ref. 3,4), including a test installation with continuous monitoring. This system is included more for completeness than as a new concept.

2.2 System 2: Indoor Air Heat Pump/Space and DHW Supply (IAHP)

This system is essentially an air-conditioning heat pump that dumps waste heat to a large warm water tank whenever the cooling setpoint (26°C) is exceeded (see Fig. 3B). It can be configured as an air-conditioning-only heat pump or an air conditioning and ventilation heat recovery heat pump. In the first version the air conditioning function extracts heat from space at a nominal rate of 5 kW (cooling C.O.P. = 2.75) and stores the gains in a 10 000 L tank as a preheat to a standard hot water tank. The back-up heater is sufficiently large to maintain 55°C water for supply to space and DHW loads. The second version has an air conditioning function sized to extract the maximum amount of ventilation energy without frosting the coil (1.5 kW cooling at a C.O.P = 2.75). The system extracts heat from ventilation air when the space cooling function is not required. With an exhaust-only ventilation strategy, fresh air can be drawn into the house passively via a sunspace thus taking advantage of ventilation air preheat.

2.3 System 3: Integrated Mechanical System/Ice Phase Change Storage (IMS)

An integrated mechanical system was designed for NRC (Ref. 5) and prototyped with EMR funding (Ref. 6). The prototype has been recently tested at Ontario Hydro and a limited number of units are being produced for field installations. At time of writing one field installation is operating well in a low energy house in Toronto and a second installation is imminent in the Advanced Demonstration House in Brampton, Ontario.

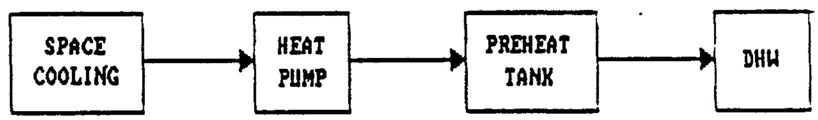


Figure 3A INDOOR AIR HEAT PUMP/DHW SUPPLY

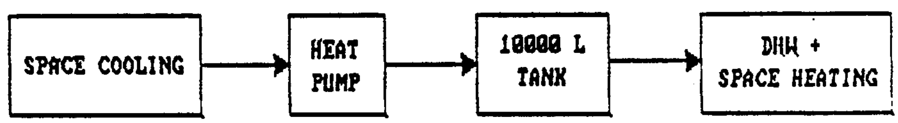


Figure 3B INDOOR AIR HEAT PUMP/SPACE AND DHW SUPPLY

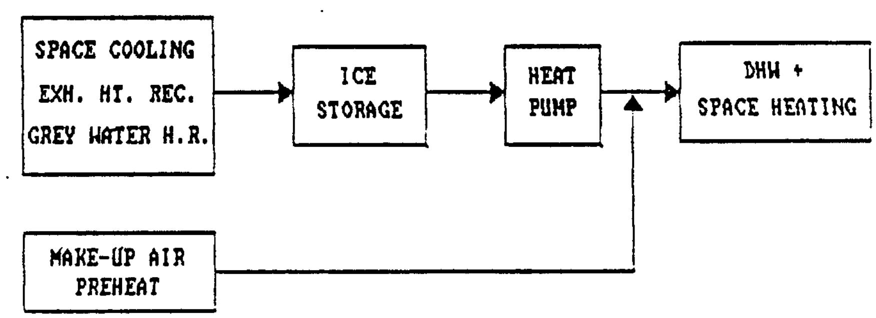


Figure 3C INTEGRATED MECHANICAL SYSTEM W/ ICE STORAGE

The integrated mechanical system provides space and DHW heating, space cooling, ventilation with heat recovery and grey water heat recovery (see Fig. 3C). The system extracts energy from exhaust air, space and grey water and adds it to a 450 L storage tank. The evaporator, in turn, extracts energy from the storage tank via a heat exchanger and tries to freeze the water in the storage tank.

The ice tank is designed as a diurnal phasechange storage and has a storage capacity of 40 kWh when compared to the maximum ice condition. The extracted energy combined with compressor input energy (nominally 5 kW at a C.O.P. of 2.25) is transferred to the condenser through which water from the hot water tank is circulated and heated. Space heating is supplied from the hot water tank via a hydronic coil for forced air heating or directly via hydronic radiators. If the fresh air is drawn into the house passively in the exhaust-only configuration, the system can also take advantage of fresh air preheat via a massive sunspace.

2.4 System 4: Integrated Mechanical System/Ground Storage with Phase Change (IMS/GRD)

The basic difference between this system and System 3 above is that the evaporator side storage is replaced by a seasonal ground storage (see Fig. 3D). The moisture in the ground may freeze and store additional energy when the ground temperature drops below $0^{o}C$. Similiar in concept to the ice tank, the ground storage is a large "tank" of soil/water mixture (44% water by volume) or a man-made aquifer (see Fig. 4). Its walls are in contact with the ground, except for the top which is assumed to be highly insulated. Cold water from the ground storage tank (usually ranging from $0^{o}C$ to $10^{o}C$) is circulated to pick up gains from exhaust air, space cooling and grey water. The brine (i.e. water/ethylene glycol solution) circulating through the heat exchanger and via the evaporator extracts energy from the ground storage. As the "tank" temperature drops or rises, energy is either extracted or added to the surrounding soil. The energy extracted by the evaporator is used to supply DHW and space heating via a hot water tank as in System 3.

2.5 System 5: Heat Pump-Coupled Rockstore

Rockstores which are mechanically coupled to space air for the purpose of moderating temperature swings have been used on numerous occasions. A rockstore employing a reversible airflow damper for efficient charging and discharging was used in the Toller-Tener House in Ottawa (Appendix C). In this way, the excess solar gain is delivered to the top of the storage during the day and, by reversing the airflow at night, heat is extracted from the warmest part of the storage, i.e. again the top (see Fig. 5). System 5 (see Fig. 3E) uses the heat pump to extend the sensible range of the rock store by "superheating" it on charging (space cooling mode) and "supercooling" it on discharge (heating mode). Rather than using reversible refrigerant valves, which are expensive and tend to degrade performance, the reversible airflow strategy described above and a unidirectional heat pump are employed.

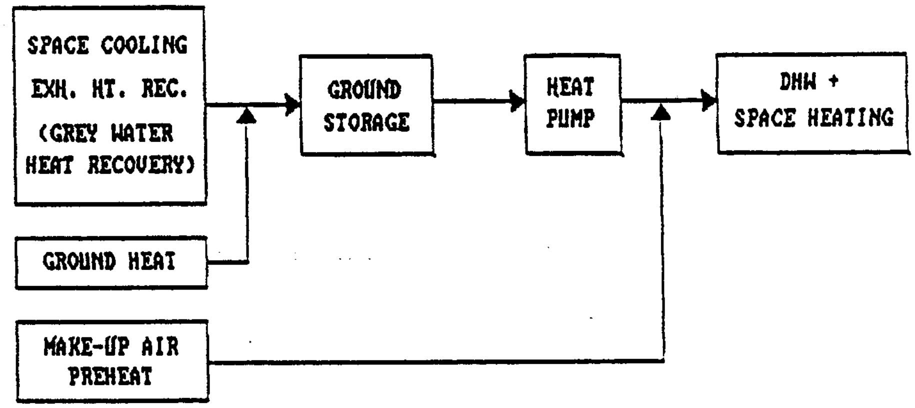


Figure 3D INTEGRATED MECHANICAL SYSTEM W/ GROUND (+ICE) STORAGE

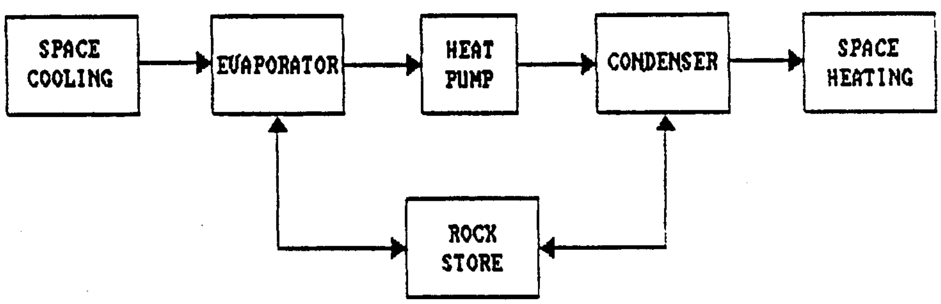


Figure 3E HEAT PUMP COUPLED ROCK STORE

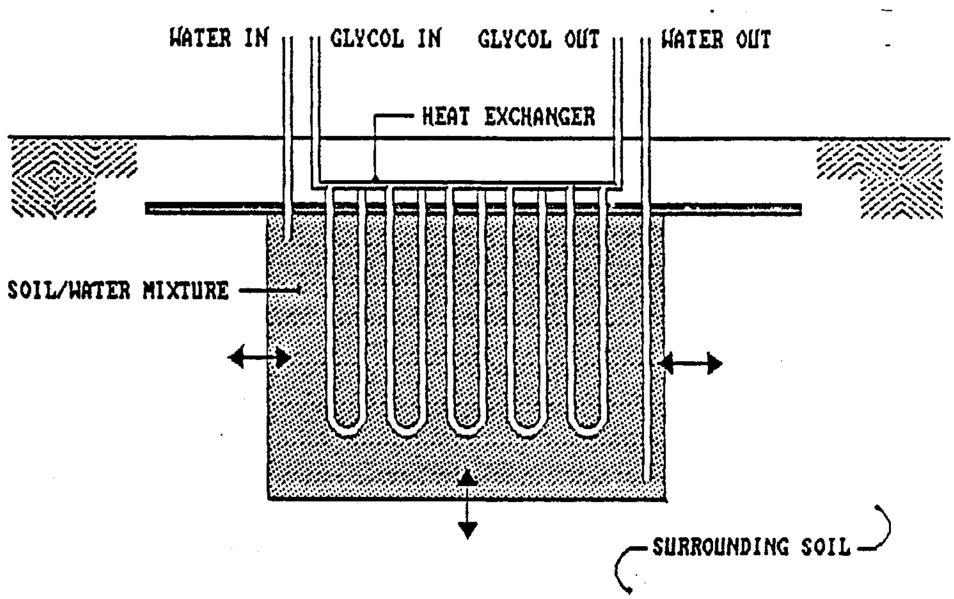


Figure 4 GROUND STORAGE CONFIGURATION

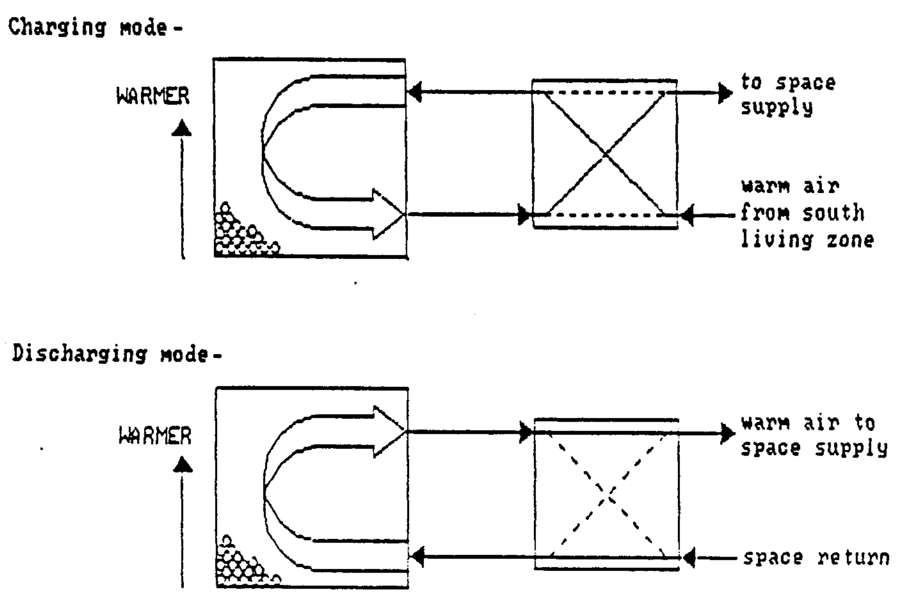


Figure 5 REVERSIBLE AIRFLOW THROUGH ROCK STORE

The computer simulation program, ENERPASS, was modified with several new thermal models, such as a ground storage model with ice phase-change, a reversible airflow rockstore and an interposed mass wall between a sunspace and the main living space.

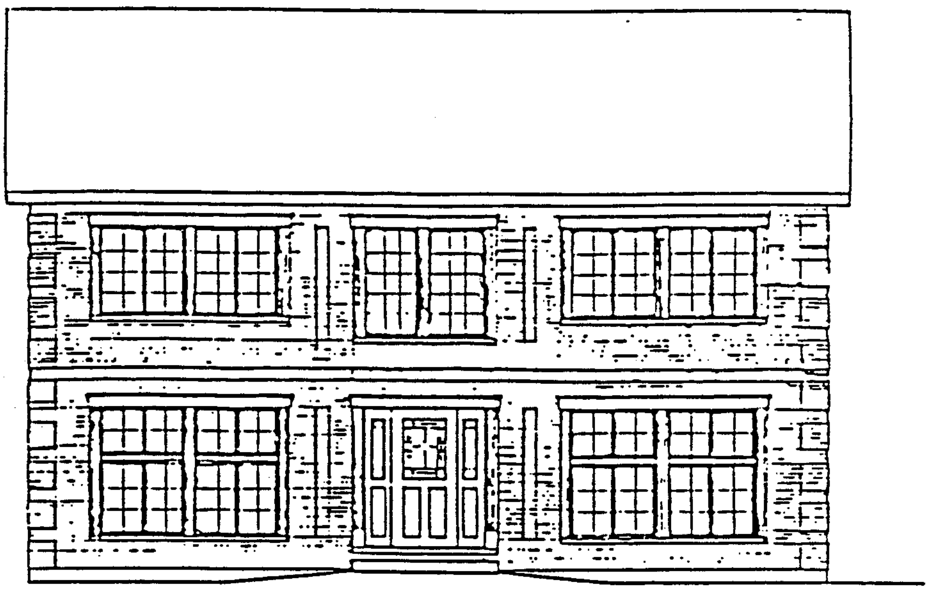
3.0 THERMAL PERFORMANCE MODELLING

3.1 Model Houses

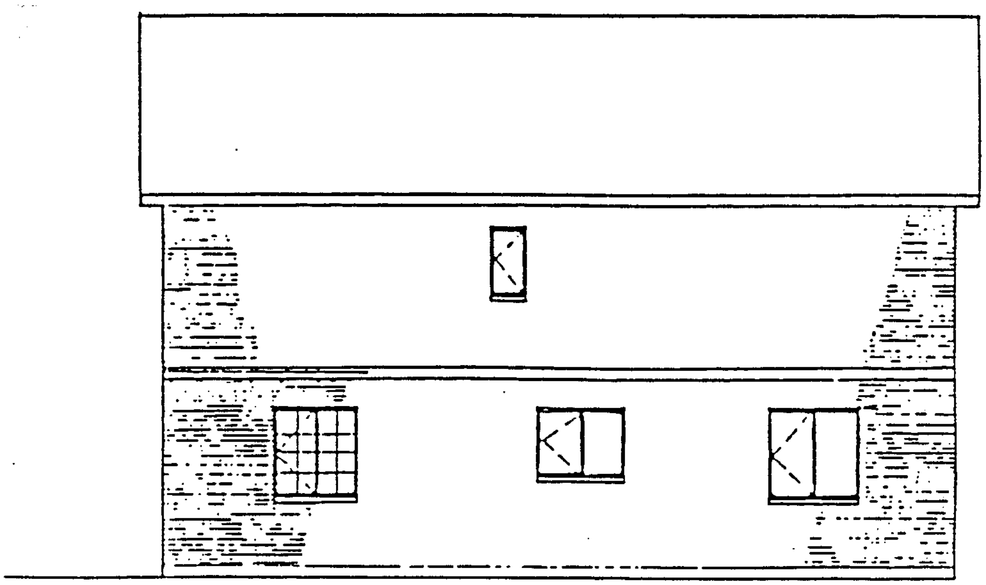
The reference house design is the "NRC House" which has been used in numerous Canadian energy-efficient and passive solar simulation studies (see Fig. 6A and 6B). It has been modified to include some east windows and more significant west windows, the latter being useful for additional gains late in the day for increased night heating supply. This base case house has a south glazing-to-interior floor area (GFR) of 7%. The window areas, including 25% frame area, for this house are given in Figure 7.

The base case mechanical system is an electric forced-air furnace with hot water delivered via an electric water heater. Mechanical ventilation of 58 L/s is provided by a balanced central exhaust system with or without heat recovery depending on comparisons made. This ventilation rate is 0.3 ACH, the same as the preliminary F326 ventilation standard. The new standard also has a room-by-room requirement; however, for this size of house the 0.3 ACH requirement would typically take precedence. For the SEE house (see below) ventilated in an exhaust-only mode, the 58 L/s of ventilation air can also be drawn into the living zone via the sunspace thus taking advantage of solar preheat.

Three types of housing were defined: an "NBC" house insulated at the National Building Code levels and using double-glazed windows; an Energy-Efficient or "EE" house with higher insulation levels, significantly improved windows (double-glazed, Low-E/argon fill) and basement slab insulation; and a Super-Energy-Efficient or "SEE" house with a further increase in insulation levels, RSI 1.25 windows (triple-glazed, double Low-E/argon fill) on the north, east and west, insulation between basement and main living space and an unheated attached sunspace (see summary, Table 1). The attached sunspace is located on the first floor of the south facade and constitutes 50% of the south facing glazing. To maintain high solar gains all south glazing and the east and west glazing of the sunspace are double-glazed, low-E/argon units which have 75% higher transmissivity than the RSI 1.25 glazing. Note that the EE and SEE house types used in this study are not to be confused with any government or utility-run energy-efficient housing programme.



FRONT ELEVATION



REAR ELEVATION

Figure 6A. Reference House Elevations

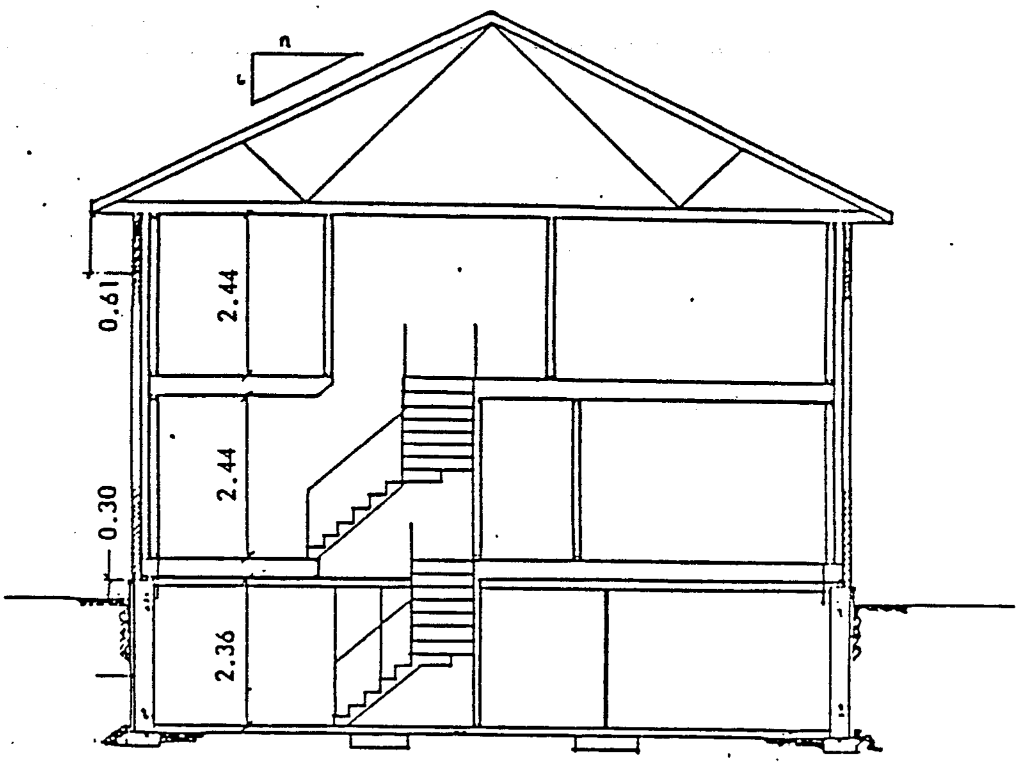
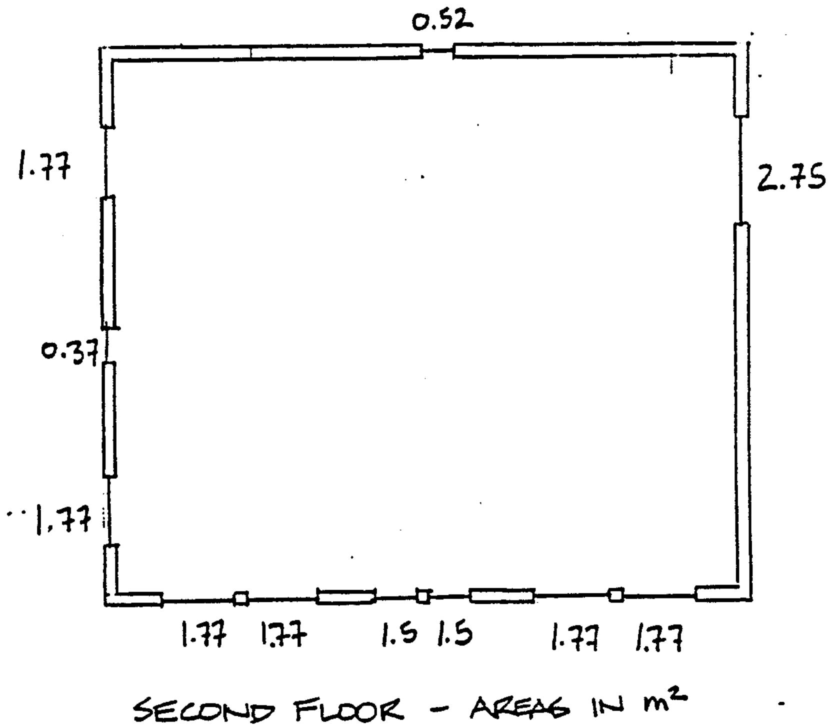


Figure 6B. Reference House Section



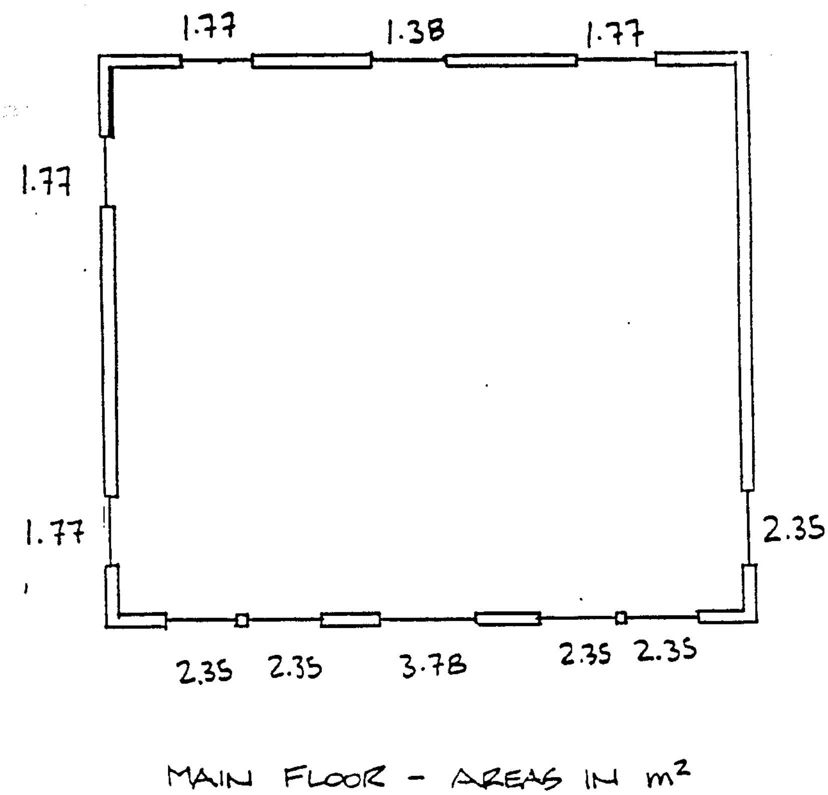


Figure 7. Reference House Window Areas (GFR = 7%)

HOUSE TYPE	NBC	EE	SEE
INSULATION LEVELS			
Walls (RSI)	3.6	6.4	7.0
Ceilings (RSI)	5.6	9.0	10.0
Glazing: South (RSI/TRANS.)	0.35/0.76	0.70/0.62	0.70/0.62
Other (RSI/TRANS.)	0.35/0.76	0.70/0.62	1.25/0.43
Frame (RSI)	0.70	0.70	0.70
Floor between Basement/Main Floor	0.84	0.84	3.6
Basement Walls	2.0	2.0	2.0
Basement Floor	0.2	1.32	1.32
INFILTRATION (ACH)	0.02	0.02	0.02
VENTILATION RATE (L/s)	58	58	58
(ACH)	0.3	0.3	0.3
SUNSPACE	No	No	Yes
Design loads (kW) Ottawa	16	11.5	8
Winnipeg	18	12.5	9.5
Vancouver	10.5	7.5	5.5

TABLE 1 - THERMAL PARAMETERS OF HOUSE TYPES USED IN PERFORMANCE SIMULATIONS

3.2 Heat Pump Performance

While heat pump performance characteristics are affected by many variables, e.g. refrigerant cycle design, condenser and evaporator sizing, source and supply media (air or water), it was decided for the purposes of this study to generalize heat pump performance as much as possible. In other words, the subject of study is heat pump/thermal storage combinations rather than heat pump performance.

From a review of various manufacturers' literature, the basic variables affecting coefficient of performance (C.O.P.), in the order of importance, are:

1. Evaporator temperature
2. Condenser temperature
3. Heat exchange fluid (liquid or air) at evaporator and condenser

The four types of heat pumps are air-to-air, liquid-to-air,, liquid-to-liquid and air-to-liquid. The liquid can be either water or an anti-freeze solution ("brine"). For this study Systems 1 and 2 employ air-to-liquid heat pumps supplying hot water, Systems 3 and 4 are liquid-to-liquid heat pumps, and an air-to-air heat pump is used with the rock store (System 5). In general, heat pumps are most sensitive to the evaporator (or source) temperature. Assuming that an air- or water-source evaporator is sized optimally, both heat pumps will have similar performance. Heat pumps which have the condenser delivering energy to hot water are optimized for entering water temperatures of 50°C to 60°C while air condensers deliver to space temperature (20°C). As well, liquid condensers see a large range of temperatures (15°C-60°C) depending on the condition of the DHW tank. There is some justification, then, to separate air supply and water supply heat pumps based on entering condenser temperatures. Figure 7 summarizes the above discussion as a series of lines. Note that the definition of C.O.P. includes all parasitic energy such as pumps, except space heating fan energy:

$$COP = TOTAL\ OUTPUT/TOTAL\ INPUT$$

where TOTAL INPUT = Compressor Input + Evaporator Fan/Pump + Condenser Pump (if applicable)

This also makes the definition consistent with steady state combustion efficiency for furnaces.

Further it is assumed that the output of heat pumps can be scaled up or down without any effect on C.O.P. Thus a unit output relationship can be defined with a value of 1 kW at 0°C and a slope of 0.016 kW/°C rise in evaporator temperature (see Fig. 8). This approximates manufacturers' data. The performance data used for the various systems shown in Table 2 was derived from the relationships shown in Figure 8. Beyond an evaporator inlet temperature of 20°C the C.O.P. and input energy values level off to constant values.

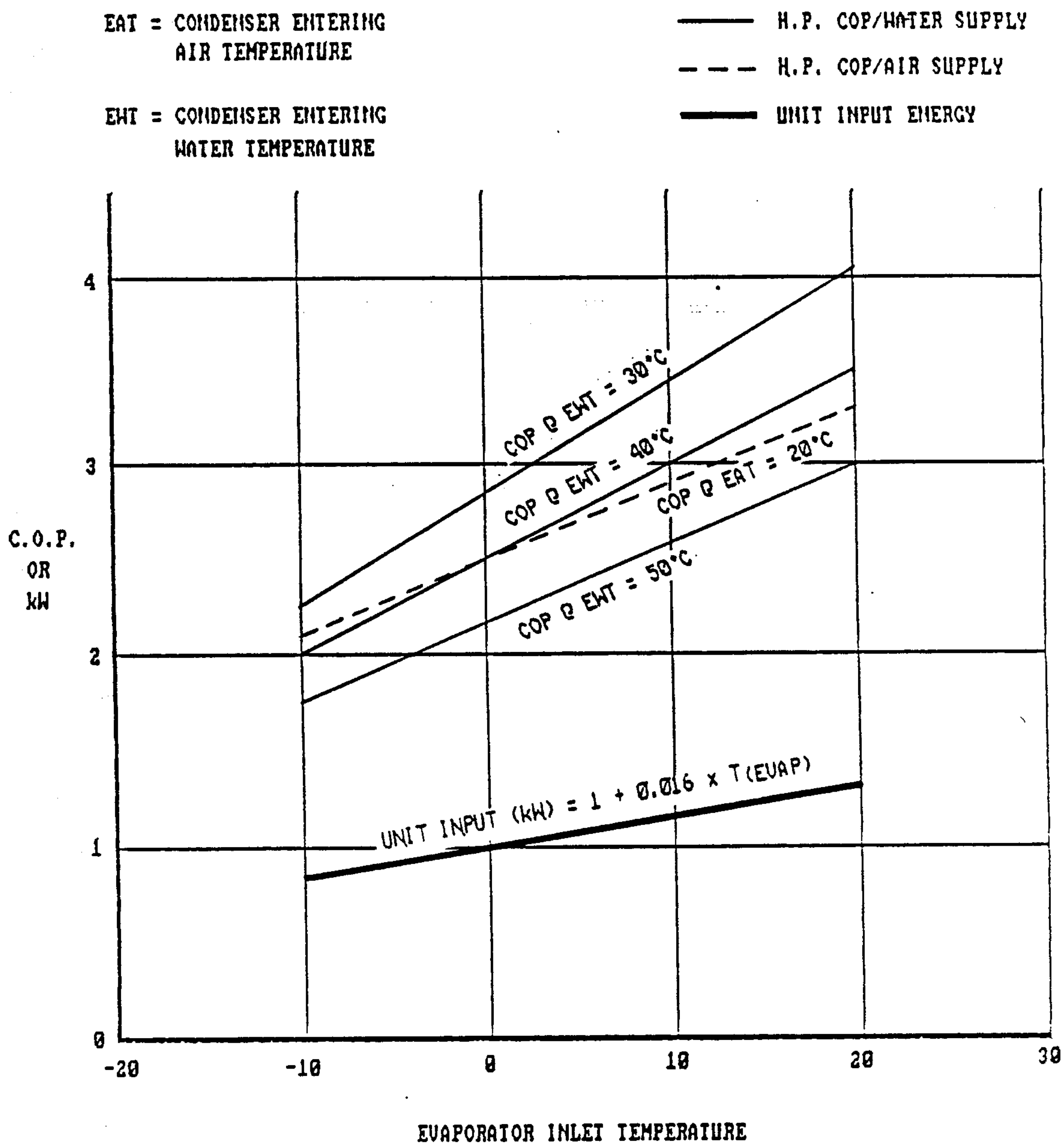


Figure 8 C.O.P. RELATIONSHIPS FOR MODEL HEAT PUMPS

EVAP. INLET TEMPERATURE °C	HEATING C.O.P.	HEATING OUTPUT kW
10	3.0	2.6
15	3.25	2.8
20	3.5	3.0
25	3.75	3.2
30	4.0	3.4

Input Energy at 0°C = 0.75 kW

TABLE 2A - INDOOR AIR HEAT PUMP PERFORMANCE (System 1)
(Average Condenser Inlet Temperature = 40°C)

EVAP. INLET TEMPERATURE °C	HEATING C.O.P.	HEATING OUTPUT kW	
		A	B
10	3.0	4.5	1.6
15	3.25	5.25	1.8
20	3.5	6.0	2.1
25	3.75	6.8	2.4
30	3.75	6.8	2.4

Input Energy at 0°C = 1.3 kW (A)
= 0.5 kW (B)

TABLE 2B - INDOOR AIR HEAT PUMP PERFORMANCE (System 2)
(Average Condenser Inlet Temperature = 40°C)

EVAP. INLET TEMPERATURE °C	TANK TEMP. °C	HEATING C.O.P.	HEATING OUTPUT kW
-10	-5	2.0	4.0
-5	2	2.25	5.0
0	9	2.5	6.0
5	15.25	2.75	7.1
10	24	3.0	8.4
15	31.75	3.25	9.7

Input Energy at 0°C = 2.4 kW (System 3 only)

TABLE 2C - INTEGRATED MECHANICAL SYSTEM (Systems 3 & 4)
(Average Condenser Inlet Temperature = 40°C)

EVAP. INLET TEMPERATURE °C	HEATING C.O.P.	HEATING OUTPUT kW
10	2.9	4.7
15	3.1	5.4
20	3.3	6.1
25	3.5	6.9
30	3.5	7.0

Input Energy at 0°C = 1.4 kW

TABLE 2D - ROCKSTORE HEAT PUMP PERFORMANCE (System 5)
(Average Condenser Inlet Temperature = 20°C)

3.3 System Models for Computer Simulations

3.3.1 ENERPASS Program

The systems identified in Section 2.0 are modelled using the computer residential energy simulation program ENERPASS (Ref 1). This program utilizes annual hour-by-hour weather data and calculates energies and temperatures in the model house anywhere from once to 10 times an hour depending on requirements for convergence. The model building can be split into a number of zones, each capable of modelling basement conditions. Standard features include a variety of mechanical system options, such as various recirculation and ventilation strategies, heat pump DHW heating and heat pump space heating and cooling. A ground source heat pump model is included; however, this assumes that the ground is an infinite heat sink and heat can be added or extracted from the ground without impacting the ground temperature regime. This model was not considered adequate for our purposes. As mentioned earlier, an integrated mechanical system model was available from work performed prior to this study (Ref. 5).

The DHW draw profile used in ENERPASS was as follows (total daily load = 100%):

Hour	%	Hour	%	Hour	%	Hour	%	Hour	%	Hour	%
1	0	5	0	9	2.2	13	2.2	17	11.1	21	2.2
2	0	6	0	10	20.0	14	0	18	20.0	22	0
3	0	7	4.4	11	2.2	15	0	19	11.1	23	0
4	0	8	11.1	12	4.4	16	4.4	20	4.4	24	0

The latest version, ENERPASS 3.0, allows locating a mass wall between zones, so that direct gains collected by the mass wall are not only re-radiated to the sunspace itself, but also conducted through the interposed mass wall to the adjacent living zone. This reflects reality more closely and allows careful modelling of fresh air preheating via a massive sunspace.

Further new modelling developed for this study are the ground source model and the reversible airflow rockstore coupled to the evaporator and condenser of an air-to-air heat pump. All of the system models are described below in more detail.

3.3.2 Base Case System

The base case mechanical strategy assumes an electric forced air furnace and an electric 272 L domestic hot water (DHW) tank, both assumed to deliver heating at an efficiency of 100%. For this feasibility assessment only one DHW load, 250 L/day or 19 GJ/yr, similar to R-2000 requirements, was used in the modelling. The forced air recirculation fan has a capacity of 300 L/s which includes the 58 L/s of ventilation air. Ventilation is assumed to be exhausted centrally and supplied to the south zone and north zone at 35 L/s and 23 L/s respectively. Heat recovery can be achieved either by exhaust only heat extraction via a heat pump or as a balanced ventilation system utilizing a 75% effective heat recovery ventilator. The HRV was modelled as a sensible heat recovery device operating year round, either preheating or cooling fresh air depending on indoor-outdoor temperature difference. If a sunspace is available all of the fresh air is drawn through the sunspace and taking advantage of preheating before delivery to the south and north zones. Heat recovery for this strategy was provided in an exhaust-only mode. Lower energy benefits would be realized by preheating the fresh air delivered to an HRV since preheat will reduce the amount of heat recovered via the core. Note that for the SEE house a second base case (SEE/P) has been defined that includes preheating of ventilation air via the sunspace (see Table 4).

When outdoor temperatures allow (i.e. are cooler than indoor temperatures) a passive cooling strategy at 3.0 ACH for the south and north zones is operated when the cooling setpoint of 26°C is reached. A central air conditioning system with a large cooling capacity (20 kW) at a cooling C.O.P of 2.0 operates if passive cooling is insufficient. Passive cooling in the sunspace was provided at 6 ACH at the cooling setpoint of 26°C. House humidity was not modelled explicitly, and only sensible cooling was provided.

3.3.3 System 1: Indoor Air Heat Pump/DHW Supply (DHWHP) - see Figure 9

The evaporator of the heat pump is located in a direct gain space (i.e. south zone) of the house or in a sunspace. In the direct gain configuration (NBC and EE houses) the heat pump extracts excess gains once the cooling setpoint has been reached. Since in this case the available cooling capacity is limited (i.e. a nominal cooling capacity of 2.5 kW), the cooling setpoint was set relatively low at 23°C both to "store" cooling by keeping indoor temperatures low and to make available as many excess gains as possible. Passive cooling (i.e. venting with outdoor air) is assumed to operate as a second stage cooling system at a setpoint of 26°C. The heat is delivered to a 250 L preheat tank with a setpoint identical to the DHW tank (55°C). When the preheat tank reaches the setpoint, the heat pump cannot operate. Any deficit in the DHW tank is supplied by the back-up element. For the sunspace configuration the only difference is that the cooling setpoint is significantly lower (15°C). This can be justified in that sunspaces commonly experience large excursions in temperature and are usually enjoyed when solar radiation is available to maintain comfort. Again, the preheat tank setpoint will limit the run-time of the heat pump. Since the ENERPASS program does not allow variable condenser temperatures, an average preheat tank temperature of 40°C is assumed. Heat pump C.O.P. performance is then defined by the curve in Fig. 8 denoted by EWT = 40°C, where the evaporator inlet temperature is the space air temperature. See Table 2A for heat pump performance data.

3.3.4 System 2: Indoor Air Heat Pump/Space and DHW Supply (IAHP) - see Figure 10

Version A is designed to capture excess passive solar gains by storing them in a seasonal hot water preheat tank. The heat pump extracts heat from space air on cooling demand (26°C) and delivers heat to the preheat tank which in turn supplies the preheated water to a conventional DHW tank. Maximum setpoints for both tanks is 55°C; however, the storage temperature ranges typically from 15°C to the setpoint. Table 2B gives the heat pump performance parameters for an average condenser temperature of 40°C. A 5 kW (1.4 ton) air conditioning capacity and 6.8 kW heating capacity at an indoor temperature of 25°C is assumed. Version B supplies modest cooling or ventilation heat recovery at a nominal 1.5 kW cooling for both functions. For the SEE house preheating of fresh air via the massive sunspace can be utilized. While the schematic, for purposes of modelling, suggests a pressurized preheat tank, in actual applications a lower cost, non-pressurized tank with a water-to-water heat exchanger and slightly different plumbing would be used.

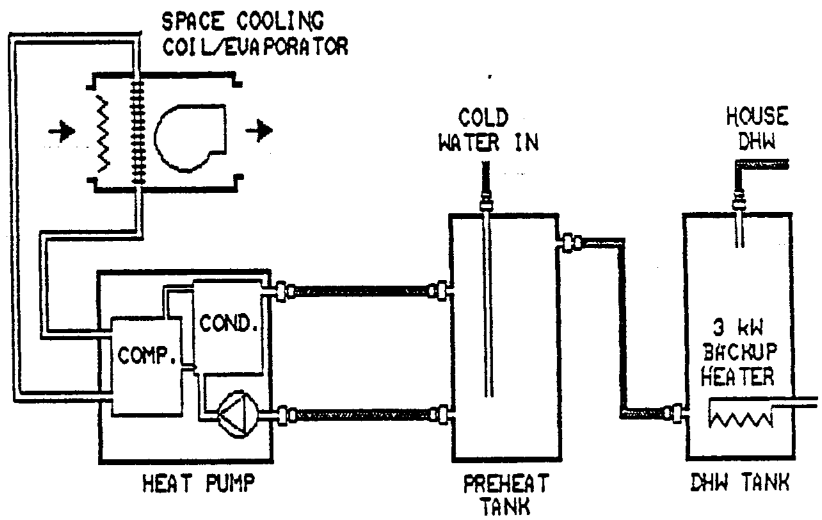


Figure 9 INDOOR AIR HEAT PUMP/DHW SUPPLY

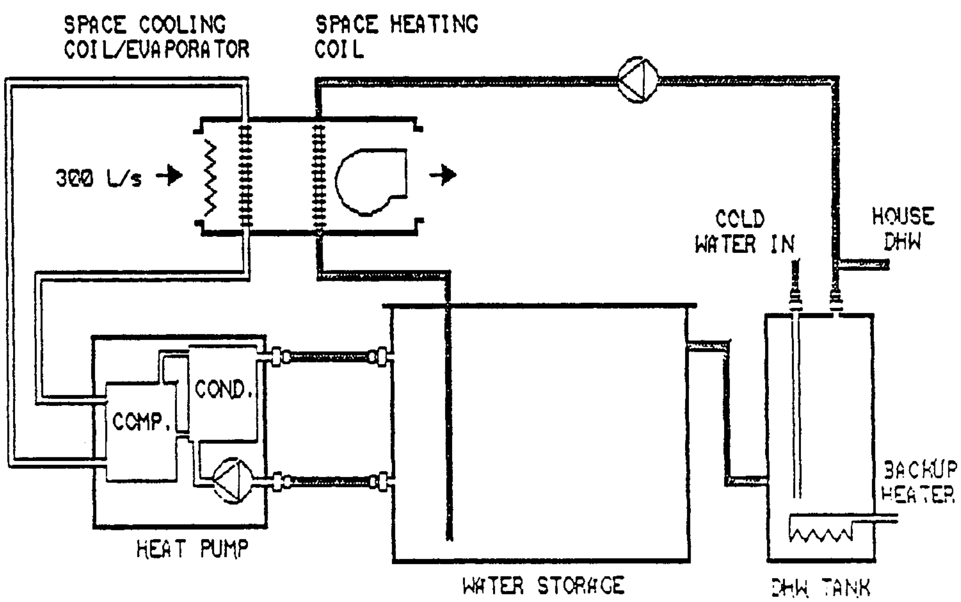


Figure 10 SYSTEM 2: INDOOR AIR HEAT PUMP/SPACE AND DHW SUPPLY

3.3.5 System 3:
Integrated Mechanical System/Ice Phase-Change Storage (IMS) - see Figure 11

This heat pump was designed to capture ventilation, grey water and space cooling gains which supplies about 40 GJ towards space and DHW heating. As such, it has an output sufficient to deliver the gains to the hot water tank. For the SEE house this corresponds to 5 kW at a thermal storage/ice tank temperature of $0^{o}C$. Typically, if the total house heating load exceeds 50 GJ back-up heating is provided to supply the difference.

Brine circulates from the evaporator through a brine-to-water heat exchanger located in the 450 L water/ice tank. It consists of 40 m of 10 mm copper tubing, outside of which ice forms with increasing diameters. The computer model tracks the heat exchange coefficient as thickness of ice increases, the amount of ice built and the tank water temperature. Ice formation is limited to 75% of the volume of the tank so that some water always remains in the tank. To avoid complex dynamic modelling of freeze-thaw cycles of the ice, heat is extracted by the heat exchanger while gains are delivered to the tank by a water loop thus melting the ice from the outside diameter. In this way the model maintains solid ice around the heat exchanger, which is consistent with the computer heat transfer model.

During the heating season heat is recovered continuously from exhaust air. Water from the ice tank is circulated via the ventilation heat recovery coil which extracts heat at a rate of 75% of the temperature difference between the ventilation air from space and the ice tank temperature. Chilled water is also circulated continuously through a coaxial grey water heat exchanger. Energy from grey water is recoverable at the DHW usage schedule with a hot/cold water ratio of 30/70. This determines the grey water delivery temperature. The grey water heat exchanger has a heat transfer effectiveness of 65% defined on the temperature difference between the grey water and the circulating water (at the tank temperature). If the space cooling setpoint is exceeded, chilled water is circulated via the space cooling coil further raising the energy of the storage tank. The space cooling coil extracts heat at an effectiveness of 75% of the temperature difference between the space air and the circulating water (at the tank temperature).

In summer, when the only heating load is DHW demand, the exhaust and grey water heat recovery functions are disallowed, e.g. similar to thermostat cooling switchover, thereby maintaining cold conditions in the storage and maximizing space cooling capacity.

Heat pump operation is controlled by an aquastat at the DHW tank. Space heating is delivered by circulating hot water from the DHW tank when the space thermostat calls for heat. When the ice tank is 75% frozen, the heat pump operation is shut down and back-up comes on. Heat pump performance is defined at the evaporator by the ice tank temperature and at the condenser by an average hot water tank temperature of $40^{o}C$. Table 2C gives the performance of the heat pump.

3.3.6 System 4:
Integrated Mechanical System/Ground Storage with Phase-Change (IMS/GRD)

The operation of this system is the same as given in Fig. 11 with the exception that the diurnal ice tank is replaced by a seasonal soil/water tank in contact with ground temperatures (see Fig. 4). Modelling of this ground storage is achieved by defining an additional basement zone filled with water and soil and insulated from the rest of the house. The model is a large version of the ice tank modified to take into account the specific heat values above and below freezing and heat of fusion for a given soil/water ratio. A typical saturated soil is 44% water and 56% solids by volume or 23% water and 77% solids by weight. As with the ice tank, to keep the phase change modelling realistic, heat is extracted via the brine heat exchanger and gains are delivered by water circulation. As temperatures of the ground storage tank rise and fall heat is extracted from or lost to deep ground temperatures according to the Mitalas basement model. In this model, the water is allowed to fully freeze on discharging to take full advantage of phase-change storage.

The heat pump capacity, volume of storage and the heat exchange heat transfer coefficient vary according to the space heating requirements (see sizing discussion in Section 3.4). While C.O.P.'s are the same as System 3, Table 2C (i.e. average condenser inlet temperature = 40°C), the output varies in proportion to the heating demand of the house being modelled.

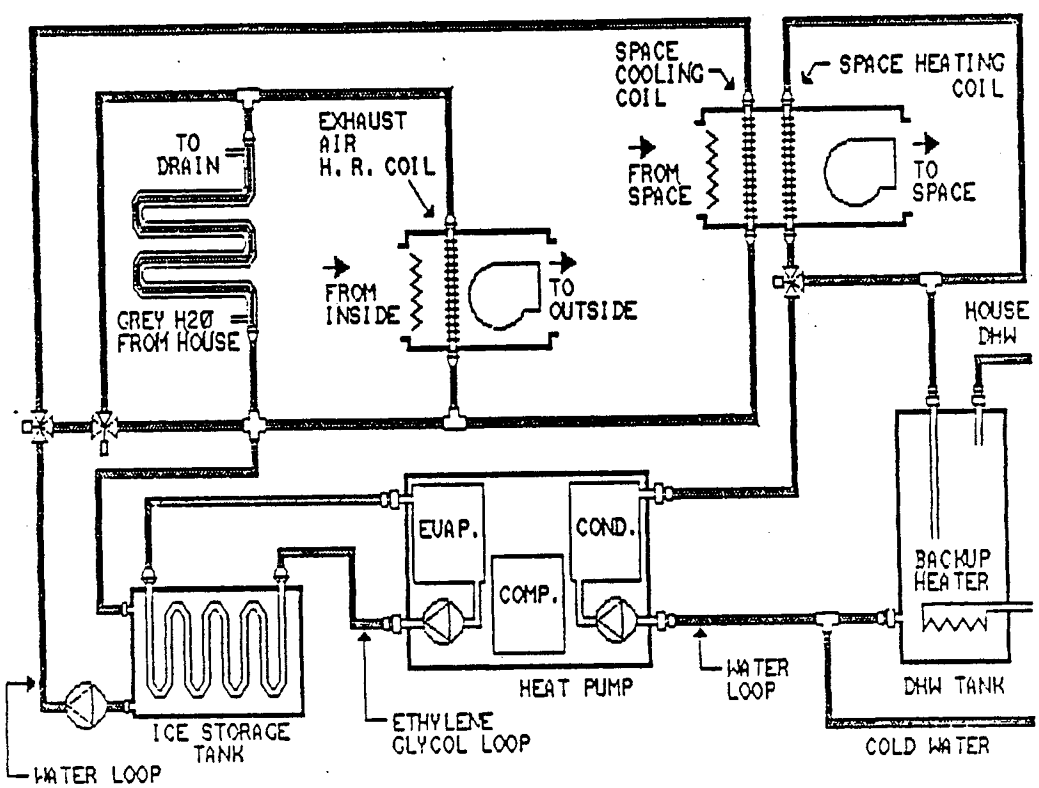


Figure 11 INTEGRATED MECHANICAL SYSTEM/ICE PHASE CHANGE STORAGE

3.3.7 System 5:
Heat-Pump-Coupled Rockstore
- see Figure 12

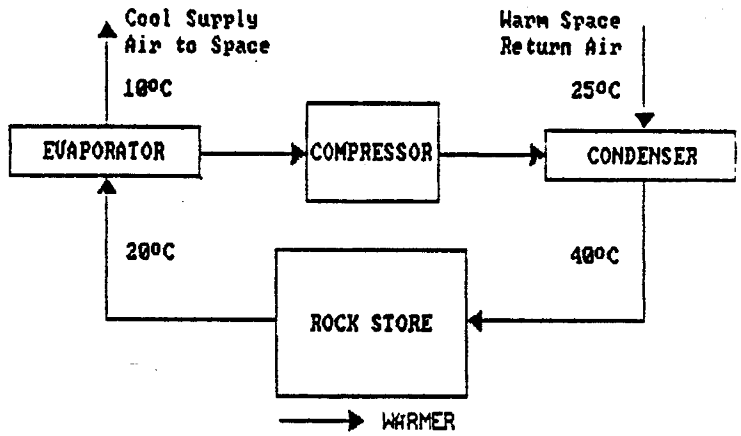
This heat pump delivers space cooling and heating by extending the sensible temperature range of the rockstore. In the cooling mode the motorized reversible damper is set such that the warm return air from space (26°C - 30°C) passes over the condenser and is heated up. This warms the right side of the rockstore, storing heat for space heating at night. The evaporator is supplied with cooler air from the depleted side of the storage and chills it to supply the cooling. A warm wave front passes through the rockstore, the extent of which depends on the amount of excess gains (plus compressor input) delivered by the cooling function. Since the thermal regime in the rockstore does not reverse, this means that some, or most, of the rock store is lower than the outlet temperature. If the storage is fully charged, i.e. the cooled air becomes too warm to supply adequate cooling (above 25°C), heat pump operation ceases and the storage is bypassed by the the air recirculation loop. When the storage outlet drops below 15°C, cooling to space is provided strictly by air circulating via the rock storage. To ensure maximum cooling capability, passive cooling of the rockstore is utilized when the temperature in the middle of the rockstore is greater than 25°C. In summer the by-pass mode is used to allow outdoor air to pass over the condenser, thus operating the heat pump as a conventional air conditioner.

In the heating mode, the airflow is reversed and moderately warm air from the right side of the rockstore is raised in temperature by the condenser for delivery to space heating demand. If heat pump operation cannot meet the load any deficit is supplied by a duct heater in the supply duct. When the storage is depleted (ie. below outlet temperature of 18°C) and the heat pump is basically heating in resistance mode, operation is terminated and back-up supplies the full heating load.

The rock store is modelled as a stratified thermal storage with 10 nodes. In this way the thermal wavefront can be tracked as it progresses and recedes in the rockstore.

Table 2D gives the performance characteristics of the air-to-air heat pump used with the rockstore. The condenser inlet temperatures can range from 25°C or more in the cooling mode down to 15°C when the storage is being depleted in the heating mode. Therefore an average condenser inlet temperature of 20°C has been assumed. Similarly, the rockstore supplies temperatures ranging from 15°C to 25°C, or an average of 20°C, to the evaporator, even though temperatures returning from space are greater than 26°C. The cooling capacity at 25°C is 5 kW (1.4 tons) which is equivalent to a 6.9 kW heating capacity at a C.O.P of 3.5.

Cooling (Charging) Mode -



Heating (Discharging) Mode -

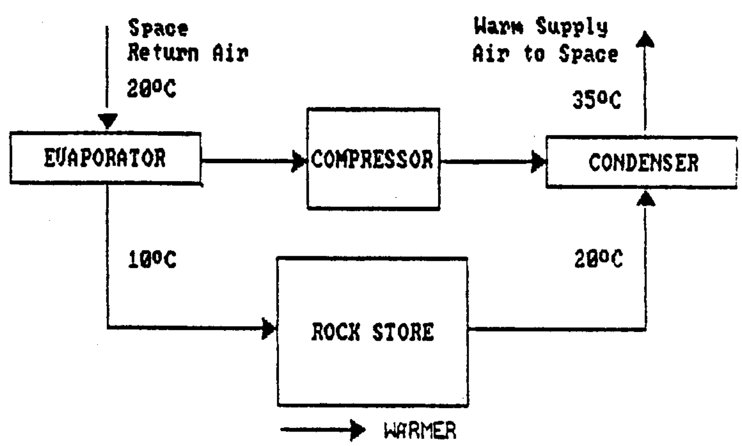


Figure 12 COOLING AND HEATING MODES OF HEAT PUMP COUPLED ROCK STORE

3.4 Results of Computer Simulations

The different house types were modelled on ENERPASS 2.1 with glazing-to-floor area ratios (GFR) of 7%, 10%, 15% and 20%. For the first three glazing areas the frame area of the south-facing windows comprised 25% of the total window area. The 20% south glazing area, including the sunspace, utilized a frame-to-window ratio of 11.5%. These areas are summarized in Table 3. The simulation results of the base case houses for a heating season from October to April in Ottawa are given in Table 4. The definitions of the various energies listed are the same as commonly used by other researchers:

Net Load - space heating load after useful internal gains but before addition of transmitted solar gains

Space Heating Load - the amount of energy a conventional furnace supplies (ie. after subtracting internal and solar gains)

Transmitted Solar Gains - the net solar gains delivered to the building interior during the heating season whether utilized or dumped

For the runs in Table 4 the percent of utilized solar gains to transmitted solar gains was calculated. Based on these results, it was decided not to concentrate further study on houses with large utilizations (e.g. 85% or greater) and, as a result, the 7% GFR NBC and EE houses were dropped.

When dealing with heating systems that also heat water and utilize integrated storage, particularly if seasonal, it is important to realize that while the total transmitted solar gains are defined on the October to April heating season, solar gains outside this period may be used to offset heating loads. Thus it is conceivable to have utilization rates of greater than 100%, as may happen, for example, when summer solar gains used for heating hot water are included.

For preliminary computer runs of the various systems, a GFR of 15% was chosen as the typical large glazing area for NBC, EE and SEE houses.

GFR	South Window Area m^2	South Glazing Area m^2	Frame/ Window Ratio
7%	19.6	14.7	25%
10%	27.6	20.7	25%
15%	41.3	31.0	25%
20%	46.7	41.3	11.5%

TABLE 3 - WINDOW AREAS CORRESPONDING TO GLAZING-TO-FLOOR AREA RATIO (GFR)

Table 5 gives the results of the computer modelling for Ottawa for a heating setpoint of 21°C.

The definitions for the values in Table 5 are as follows:

TOTAL HEATING ENERGY =	Space Energy + DHW Energy Consumption - for the Ottawa base case this is the sum of the space load from Table 4 plus the Ottawa DHW load of 19 GJ.
SAVING =	Difference between Base Case Total Heating Energy and System Total Heating Energy.
SOLAR TRANSMITTED =	Transmitted Solar Gains during Oct.- Apr. Heating Season (59% of Annual Transmitted Solar Gains in Ottawa).
SOLAR UTILIZED =	Amount of gains actually used to displace heating load. % utilized is utilized solar gain divided by seasonal transmitted solar gains (Oct.- Apr.).
TOTAL GAINS =	Sum of all gains used to displace heating load; from solar, exhaust air, greywater, and ground.

The base cases have been run with summer and winter passive cooling (priority) and central air conditioning (as required) at a cooling set point of 26°C. The base case runs use a conventional mechanical system consisting of an electric furnace and an electric water heater. The EE and SEE houses were modelled with and without a 75% HRV. The base case SEE house was also modelled with preheat of fresh ventilation air via the sunspace (denoted as See/P). Table 5 separately lists the air conditioning energy required.

HOUSE	GFR	NET LOAD	SPACE HTG. LOAD	UTILIZED SOLAR GAINS	TRANSMITTED SOLAR GAINS	% UTIL.
	%	GJ	GJ	GJ	GJ	%
NBC	7%	110.9	80.4	30.5	33.0	92.4%
	10%	119.3	82.6	36.7	42.3	86.8%
	15%	133.0	88.6	44.4	58.2	76.3%
	20%	143.5	89.1	54.4	74.2	73.3%
EE w/o HRV	7%	77.9	53.0	24.9	26.9	92.6%
	10%	82.7	53.7	29.0	34.6	83.8%
	15%	90.8	55.1	35.7	47.5	75.2%
	20%	94.4	53.9	40.5	60.5	66.9%
EE w/ HRV	7%	54.4	31.2	23.2	26.9	86.2%
	10%	59.1	32.7	26.4	34.6	76.0%
	15%	67.1	34.7	32.4	47.5	68.2%
	20%	70.7	34.1	36.6	60.5	60.5%
SEE w/o HRV	10%	71.5	47.2	24.3	38.3	63.4%
	15%	75.4	45.5	29.9	51.4	58.2%
	20%	79.0	44.6	34.4	64.4	53.4%
SEE w/HRV	10%	47.3	24.7	22.6	38.3	59.0%
	15%	51.0	23.7	27.3	51.4	53.1%
	20%	54.7	23.2	31.5	64.4	48.9
SEE/P w/Preheat	10%	67.1	39.1	28.0	38.3	73.0%
	15%	70.6	36.7	33.4	51.4	65.0%
	20%	74.1	35.3	32.8	64.4	50.9%

TABLE 4 - VARIATION OF WINDOW AREAS IN HOUSES WITH CONVENTIONAL MECHANICALS (Oct.- Apr. heating season in Ottawa)

		TOTAL HEATING ENERGY GJ	SAVING GJ	SOLAR TRANS. GJ	SOLAR UTIL. GJ/%	TOTAL* GAINS GJ	AVG. JULY TEMP. °C	A/C ENERGY GJ
NBC	BASE	107.6	-	58.2	44.4/76	-	24.7	9.1
	DHWHP	97.6	10.0	58.2	54.4/95	-	25.7	-
	IAHP A	95.3	12.3	58.2	56.7/97	-	25.4	10.1
	IAHP B	75.5	32.5	58.2	76.2/131	-	27.6	5.4
	IMS	70.8	35.4	58.2	-	81.2	24.8	-
	IMS/GRD	50.1	57.5	58.2	-	101.9	24.8	-
	HP/RS	104.2	3.4	58.2	47.8/82	-	24.9	5.6
EE w/o HRV	BASE	74.1	-	47.5	35.7/73	35.7	25.1	7.6
	DHWHP	63.7	10.4	47.5	46.1/97	46.1	26.0	-
	IAHP A	56.4	17.7	47.5	53.4/112	53.4	25.4	12.3
	IAHP B	46.6	27.5	47.5	-	63.2	27.7	5.8
	IMS	40.9	33.2	47.5	-	68.9	25.1	-
	IMS/GRD	32.2	41.9	47.5	-	77.6	26.0	-
	HP/RS	71.4	2.4	47.5	38.4/81	-	25.2	6.0
EE w/HRV	BASE	53.7	-	47.5	32.4/68	56.1	25.4	8.6
	DHWHP	42.6	11.1	47.5	43.5/92	67.2	26.5	-
SEE w/o HRV	BASE	64.5	-	51.4	29.9/58	29.5	25.4	4.8
	DHWHP	52.8	11.7	51.4	41.6/81	41.6	26.3	-
SEE w/HRV	BASE	42.7	-	51.4	27.3/53	51.7	25.7	5.6
	DHWHP	30.9	11.8	51.4	39.1/76	63.5	26.7	-
SEE/P	BASE	55.7	8.8	51.4	38.7/75	38.2	25.7	6.2
	IMS	29.1	35.4	51.4	-	65.3	25.6	-
	IMS/GRD	24.6	39.9	51.4		69.8	27.0	

*Includes HRV contribution, if applicable

NOTE: Total Heating Energy refers to the total of Space + DHW heating energy (based on 19 GJ Annual DHW load).

TABLE 5 - ENERGY PERFORMANCE OF DIFFERENT HEAT PUMP/THERMAL STORAGE SYSTEMS (GFR = 15%) - Ottawa

Except for Systems 2A&B: Indoor Heat Pump and System 5: Heat-Pump-Coupled Rockstore, which try to maximize seasonal gains from air conditioning for supplying to heating loads, the heat pump systems have been run with passive cooling in the non-heating season. All systems have the heat pump air conditioning function available year-round. No separate air conditioning energy is listed in Table 5 for the DHWHP, IMS and IMS/GRD systems since the air conditioning function is provided by virtue of the heat pump heating operation. The cooling setpoint is 26°C for all systems except for System 1: Indoor Air Heat Pump/DHW (see below). The average temperatures in Zone 1 for July are listed in Table 5 as a check that similar comfort conditions to the base cases are provided by the heat pump systems.

For the base cases it can be noted that as the building load goes down so does the air conditioning energy consumption; however, this is not true when an HRV is added. While the space load goes down over the non-HRV base case the A/C energy goes up. This is due to the way that the HRV was modelled. The HRV recovers sensible heat throughout the year warming the fresh air (at 75% effectiveness) in the winter and summer if the outdoor temperature is below indoors and cools the air whenever the indoor air is cooler than outdoors. Even in the summer the fresh air preheat happens for the majority of the time thus reducing the "cooling" capability of the ventilation air in the HRV. It should be kept in mind that from May to September there are 44 GJ of solar and internal gains that enter the EE house in Ottawa. The modelling further assumes that above the cooling setpoint (26°C), 3 ACH of passive cooling in the summer are provided. With 10 ACH passive cooling in the summer the difference in A/C energy is reduced from 1.0 GJ to 0.4 GJ; however, 3 ACH are considered a more reasonable rate of passive cooling. The modelling suggests that bypassing the HRV in summer when the outdoor temperature is below indoors may be a worthwhile strategy. Note that exhaust-only heat pumps (eg. integrated mechanical system) replace HRV's; however, non-ventilating heat pumps (eg. DHW heat pumps) can be modelled with and without HRV's. For exhaust-only heat pumps the ventilation strategy was modelled such that the central exhaust fan draws fresh air into the house by negative pressure. In the NBC and EE houses, this reduces the effective ventilation load slightly since infiltration and supply air are not separated as is the case with balanced ventilation. For the SEE house, the ventilation air is drawn in via the sunspace which contributes a total reduction in load for the base case in Ottawa of 8.8 GJ. As the ventilation air is drawn through the sunspace it picks up gains, ie. heat loss into sunspace from living zone and direct gains from solar. From Tables 4 and 5 it can be seen that 55% of this benefit (4.8 GJ) can be attributed to the house heat loss and 45% to solar gains.

The output data for GFR's of 10%, 15% and 20% are given in Tables 6A to 6E at the end of section 3.4.

3.4.1 System 1:
Indoor Air Heat Pump/DHW Supply (DHWHP)

This system takes excess gains from the south zone when the space temperature is 23°C or higher. For the SEE house the heat pump is located in the sunspace and operates when the sunspace is above 15°C. The extracted heat is delivered to a 250 L storage tank for supplying preheated water to the DHW tank. Passive cooling is assumed to operate above 26°C. Compared to the base case there are savings ranging from 8.6 GJ to 11.8 GJ ($140 to $190) for the houses modelled. Typically, when located in the south zone the heat pump is exposed to temperatures above 23°C resulting in a C.O.P. of about 3.5. This results in an output of 3 kW, which satisfies the maximum hourly demand of the DHW profile (see section 3.3.1). However, the DHWHP is not able to supply all of the DHW load due to lower than setpoint conditions in the preheat tank as a result of insufficient air conditioning operation (ie. cooling setpoint is satisfied most of the time). This reduces the effective C.O.P. to around 2.0. For the SEE house the air temperature during heat pump operation is lower resulting in an average C.O.P. of about 3.0, but less back-up is required to satisfy the DHW load. The effective C.O.P. is, therefore, higher at 2.6. As well, when the heat pump is located in a sunspace the energy benefit is largely unaffected by the building load. This would be expected since the attached sunspace acts as a thermal buffer. The SEE house in Ottawa has minimum and average temperatures for the sunspace of 3.6°C and 13.4°C respectively. The cooling effect of the DHWHP drops the temperature regime by about 3°C which results in a minimum temperature of around freezing. In colder climates this could drop slightly further and if a higher minimum temperature is required (e.g. 5°C for growing plants) then a heater may be required slightly deteriorating the energy benefit in winter. However, the sunspace generally can be enjoyed in winter during solar periods as is typical for an unconditioned sunspace. In summer the temperatures are lower due to the DHWHP cooling contribution, thereby improving comfort.

Table 6A gives the results for the three GFR values. The effective C.O.P. ranges from a low of 1.8 (NBC house, GFR = 10%) to 2.6 (SEE house). For more discussion on DHW credits see section 5.2.1. Since this system does not supply full air conditioning it cannot be credited with the $2200 for central air conditioning. However, its cooling capacity is similar to installing a window air conditioner in the base case and assuming it satisfies half as much air conditioning as the central air conditioner. This yields a payback of just over 3 years, similar in performance to an HRV installation. Therefore, the combined payback of the DHWHP and HRV is also 3 years (see section 4.2) and it could be argued that energy-conscious homeowners practicing conscientious summer venting may experience acceptable comfort as well as significant savings in water heating and ventilation energy costs.

3.4.2 System 2: Indoor Air Heat Pump/Space and DHW Supply (IAHP)
Versions A and B.

The results for both systems are listed in Table 5. Both versions use space air (and one, exhaust air) cooling strategies to store energy in the large preheat tank. The summer gains bring the preheat tank to the 55°C setpoint. These gains are depleted in the first two or three months of the heating season, but the modelling showed that the tank does not drop below 36°C. This tempers the water for DHW demand, significantly reducing the energy consumption but incurs significant space heating consumption to satisfy the temperature rise to 55°C for the space heating coil.

Version A provides full air conditioning and is able to displace a modest amount of heating demand. For the EE house, GFR = 15%, this air conditioning function uses 12.3 GJ to deliver 17.7 GJ of cooling for a net benefit (including air conditioning energy consumption) of 13 GJ. While this system can be credited with air conditioning capability, thus reducing the incremental cost to $1850, the cost effectiveness is not particularly noteworthy see section 4.2.

Version B does not have full air conditioning for comfort control (average July temp. = 27.7°C) but supplies significant gains from space cooling and ventilation heat recovery to supply net savings of 29 GJ for the EE house, GFR = 15%. Even if the base case is assumed to have a window air conditioner (at a cost of $1000), the payback performance does not drop below 6 years (see section 4.2).

No further simulations were performed for these systems.

3.4.3 System 3: Integrated Mechanical System/Ice Storage (IMS)

This system utilizes excess solar gains, ventilation heat recovery, and grey water heat recovery, whenever possible, to supply to DHW and space heating loads. No passive cooling in winter to maximize gains to IMS storage is provided. It is difficult to split out solar gains explicitly; however, total gains (including solar gains, ventilation and grey water heat recovery) are listed in Table 6B. The savings with respect to the base case in Ottawa range from 39.3 GJ for the NBC house to 31.9 GJ for the EE house. As the base case heating loads reduce, the total energy savings typically also reduce, albeit at a slower rate. However, the preheat benefit for the SEE house actually increases the energy savings beyond the larger load EE house. The preheat effect has a similar effect of depressing the temperature in the sunspace as with the DHWHP since the ventilation cooling is about equal to the DHWHP space cooling.

3.4.4 System 4: Integrated Mechanical System/Ground Storage (IMS/GRD)

This system is similar to System 3, except that the ground storage needs to be sized according to the heating requirements of the house. The model assumes heat extraction from and delivery to a saturated soil volume (44% water by volume). A small study was undertaken to establish design guidelines for sizing the storage volume and the heat exchange (UA) between the soil and the evaporator. Maximum storage potential is the difference between the fully melted and fully frozen state. All of the water is frozen typically in February. This indicates that the phase-change storage is depleted and any further extraction occurs as sensible heat, thus quickly dropping the storage temperature and deteriorating the heat pump performance. Using this criterion, the volume was made as small as possible without a significant increase in heating energy consumption. Similarly, the UA coefficient, which corresponds to heat exchanger pipe length within the ground storage, was kept as small as possible without any significant effect on energy performance. Further economy can be realized by sizing heat pump capacity at less than maximum heating demand. Using a heat pump satisfying 75% of the peak demand typically satisfies 90% to 95% of the annual total heating load. Note that with this integrated heating system, no separate sizing for domestic water heating is required. From the costing criteria described in Section 4.0, i.e. the total of incremental equipment cost and 5-year energy cost, the most cost-effective storage volumes and heat transfer coefficients were evaluated. The resulting total heating load is slightly higher than with larger volumes and UA values, but the capital cost savings exceed the the increase in energy costs.

The volume of the ground storage is proportional to the storage energy capacity:

$$\text{Volume } (m^3) = A \times \text{Storage (GJ)}$$

Note that at 0^oC there is no sensible temperature change in the strorage media, i.e. soil, water.

For a 44% water/soil mixture (F = 0.44) and heat of fusion of 334 GJ/m^3:

$$\begin{aligned}\text{Volume } (m^3) &= 1/F \times \text{Storage (GJ)}/0.334\ GJ/m^3 \\ &= \text{Storage (GJ)} \times 6.8\ m^3/GJ\end{aligned}$$

From the modelling study for the Ottawa climate it was found that the required storage energy capacity varied with the total heating load as follows:

$$\text{Storage (GJ)} = -1.9 + 0.0455 \times \text{Total Heating Load (GJ)}$$

This relationship results in positive values of storage capacity for total heating loads greater than 41 GJ (or space heating loads of 22 GJ), suggesting that below about 45 GJ, ground phase change storage is unecomical. In terms of sizing:

$$\text{Volume } (m^3) = -13 + 0.31 \times \text{Total Heating Load (GJ)} > 0.5$$

From simulations for Winnipeg and Vancouver it was found that this sizing relationship is not accurate enough and a climate modifier is required. In other words, sizing is dependent not only on the heating load but also on the distribution of the degree days during the heating season. The indicator chosen was the following ratio for the Oct.- Apr. heating season:

$$F = \frac{\text{Maximum Monthly Degree-Days}}{\text{Total Heating Degree Days}}$$

The different climates have the following F values:

Location	Oct.-Apr. CDD	Max. Mnthly CDD	F
Ottawa	4420	973	.22
Winnipeg	5400	1243	.23
Vancouver	2770	541	.195

The sizing equation can then be rewritten as follows:

Volume (m^3) = -13 + F x 1.41 x Total Heating Load (GJ)

Note that F was derived from space heating degree-days and is applied to Total Heating Load, which includes DHW. This was done for simplicity but is reasonably accurate since ventilation and grey water heat recovery more than satisfy DHW load and storage requirements are roughly proportional to the maximum space heating demand.

From the values for F it can be seen that a more even heating climate, like Vancouver, would have proportionately smaller storage than Winnipeg since the storage is better utilized. It may be noted that a secondary effect exists due to the ground temperatures in Vancouver being higher than in Winnipeg and , with storage temperature typically at 0°C, slightly more gains from deep ground are realized,.

The UA coupling factor (between the soil and the brine circulating via the evaporator) is related to the maximum instantaneous extraction rate:

$$UA\ (W/^{o}C) = \text{Extraction Rate (W)} / \Delta T_{brine/soil} (^{o}C)$$

From the simulations it was found that heat pumps should be sized for an entering evaporator fluid temperature of -5°C. The heat pump/storage model imposes the ΔT between the brine and the soil depending on the conditions encountered, and it was found that for the optimum UA the average T was about 7°C. The required UA can be calculated as follows:

$$UA\ (W/^{o}C) = \frac{\text{Extraction Rate (W)}}{\Delta T\ (^{o}C)}$$

$$= \frac{[(COP - 1)/COP] \times \text{Heating Capacity (W)}}{\Delta T\ (^{o}C)}$$

where Heating Capacity is sized for 75% of the peak space heating requirement and COP is the coefficient of performance, both assumed to be at a seasonal average entering evaporator fluid temperature of -5°C. For the heat pump performance assumed in this study this COP is 2.25.

$$UA\ (W/^{o}C) = \frac{[1.25/2.25] \times \text{Heating Capacity (W)}}{7^{o}C}$$

$$= 0.08 \times \text{Heating Capacity (W)}$$

Table 7 indicates the ground storage parameters chosen for the runs listed in Table 5 and 6C. The energy savings for the ground storage modelling for the three GFR values range from 39.8 GJ to 57.5 GJ. Total gains, ie. reduction of heating load due to solar gains, ventilation and grey water heat recovery, and ground heat, range from 64.4 GJ to 111.3 GJ.

3.4.5 System 5:
Heat-Pump-Coupled Rock Store (HP/RS)

This system uses increased air conditioning operation to store gains for offsetting space heating requirements. Even with very large storage volumes there is little energy available from storage for Jan., February and March. While the potential for savings is significant with a larger, 100 m^3 rockstore (net savings 50 GJ, EE house GFR 20%), the savings for similations using the more practical storage volume of 10 m^3 have no savings and thus no cost effectiveness. Note that the 10 m^3 storage occupies an area 2 m x 2.5 m in the basement while the 100 m^3 storage would take up more than half of a typical basement. Further simulations were not pursued for this system.

The space and DHW energy consumption and utilized gains supplying to net load for the three promising systems and different house types in Ottawa is shown in Figs. 14A-14C.

As well, energy flow charts for the EE house, GFR = 15%, for Ottawa are presented in Figs. 15A-E. The following explanations help to understand the flow charts:

Net Space Heating Load - space heating load after internal gains have been credited but no solar gains.

Potential Gains - total gains available either by design of the house or by design of the mechanical system.

Utilized Gains - Gains used to offset input energy.

Potential Solar Gains - all solar gains transmitted to house interior.

Potential Ventilation Gains - HRV; the assumed equipment is limited by 100% effectiveness of ventilation load of 58 L/s (assumed heating season average indoor temperature of 22°C)
- Exhaust-only heat recovery; these systems are assumed to be capable of cooling the exhaust air to 0°C, but for various reasons do not. The year round exhaust heat recovery capability is governed by an annual average temperature drop of 22°C at 58 L/s.

Potential Grey Water Gains - the systems recovering heat from grey water are capable of pulling the water temperature down to 0°C (ie. just before it freezes). Again this is not presently feasible for a number of reasons, such as heat exchange efficiencies. The annual grey water heat recovery capability is based on 30% hot water at 55°C and 70% cold water at 8°C mains supply. This gives an average of 22°C water at a total water flow of 833 L/day.

Potential Ground Gains - the ground is used more as a storage medium than an energy source but some useful gains will be realized. It is not possible to explicitly define this value from the ENERPASS output values available.

3.4.6 Modelling Variations

The modelling took into account aperture size, climate and preheat potential of attached sunspaces. The modelling was performed with the base case and the three more promising systems, DHWHP, IMS and IMS/GRD.

The results of the aperture variation for GFR's of 10%, 15% and 20% are given in Tables 6A-6C. The houses are assumed to be located in Ottawa. As the south glass area becomes larger more solar gains are utilized to offset the increased net heating load (see fig. 13). The result is that for the more energy-efficient houses the base heating load is the same or reduces slightly. The largest increase in base heating load occurs in the NBC house when the aperture is increased from 10% to 15%. This difference amounts 6 GJ, all other differences are less than 2 GJ. In other words, at best there are only small energy benefits to increasing glazing area, but conversely there is no significant penalty for the use of large glass areas, particularly if high performance windows are used.

Note that the 20% houses yield a small improvement due to the use of smaller frame areas effectively maintaining a higher amount of insulated wall than if the frame areas were larger. This also highlights the need for improving frame R-values which, with high performance glazing is the last remaining weak link in the thermal envelope.

The DHWHP System heating energy credits (Table 6) improve slightly as more solar gains cause more air conditioning to occur. The improvements are in the order of 0.5 GJ or 5% for a 5% increase glazing area. For the NBC house, GFR = 10%, there is a noticable reduction in the energy credit due to the significantly lower air conditioning requirement, particularly in the winter, over the other houses.

The IMS heating energy credit (Table 6B) is a function of the total heating load to a maximum of about 40 GJ. For GFR = 20%, however, the credit continues to increase even though the total heating load has reduced slightly. This implies that there is a small effect of the increased solar gains on the energy credit. However, most of the gains are ventilation and grey water heat recovery.

For the IMS/GRD system (Table 6C) the solar gains make up an even smaller proportion of the gains as the system can also take advantage of ground heat. While there is a perceptible improvement in the energy credit for houses with GFR = 20%, it is a fraction of that noted with IMS system.

The climate variations for systems 1, 3 and 4 are given Tables 8A-8C. For System 1: DHWHP, the DHW credits are roughly proportional to the DHW load which varies according to the main supply temperatures. Space cooling in Ottawa and Winnipeg accounts for 53% of the hot water load in the NBC house to 62% for the SEE house. Vancouver has less operation on cooling demand and supplies 48% to 60% of the load from solar gains.

For System 3: IMS (Table 8B) the maximum energy credit is about 40 GJ for large total heating loads (see discussion in section 5.2.2).

As the total heating load to be supplied drops the energy credit can drop significantly due to increase non-utilizability of gains, eg. for the SEE house in Vancouver. However, the amount of credit is driven by the total heating load and should be relatively insensitive to climate. System 4: IMS/GRD (Table 8C), rather than having an upper limit for the credit, uses gains to supply 95% of the heating. Thus the credit is proportional to the heating load and amounts to just over 60% of the heating load to be supplied.

The table below shows the amounts of ventilation air preheat obtained by drawing 58 L/s via the sunspace of the SEE house.

	10%	15%	20%
Ottawa	8.1	8.8	9.3
Winnipeg	-	11.0	-
Vancouver	-	5.0	-

The ventilation loads for Ottawa, Winnipeg and Vancouver are 29 GJ, 36 GJ and 18.5 GJ respectively. The preheating function accounts for about 30% of the ventilation load and this is in addition to any exhaust air heat recovery via heat pump operation. Typically exhaust air heat recovery, by extracting a constant amount of heat, would exceed the performance of HRV's in Vancouver and recover some what less heat in Winnipeg than an HRV (assumed at 75% effectiveness). Combined with preheat the ventilation load reduction can exceed 100% if expressed as an equivalent effectiveness.

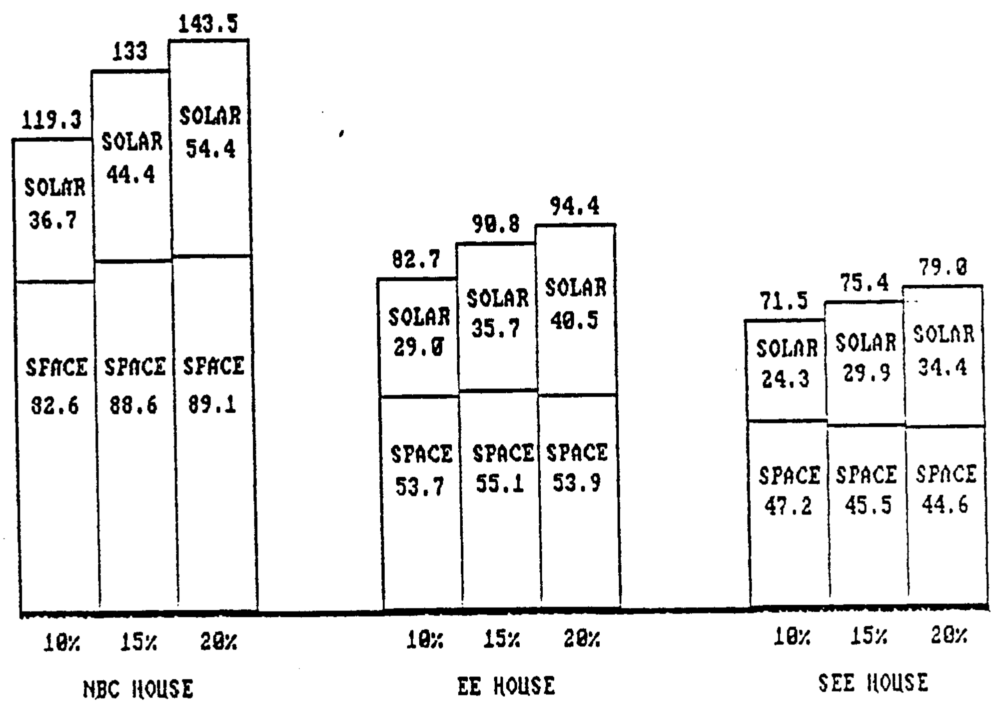


Figure 13 SOLAR GAIN UTILIZATION IN BASE CASE HOUSES WITH DIFFERENT APERTURES

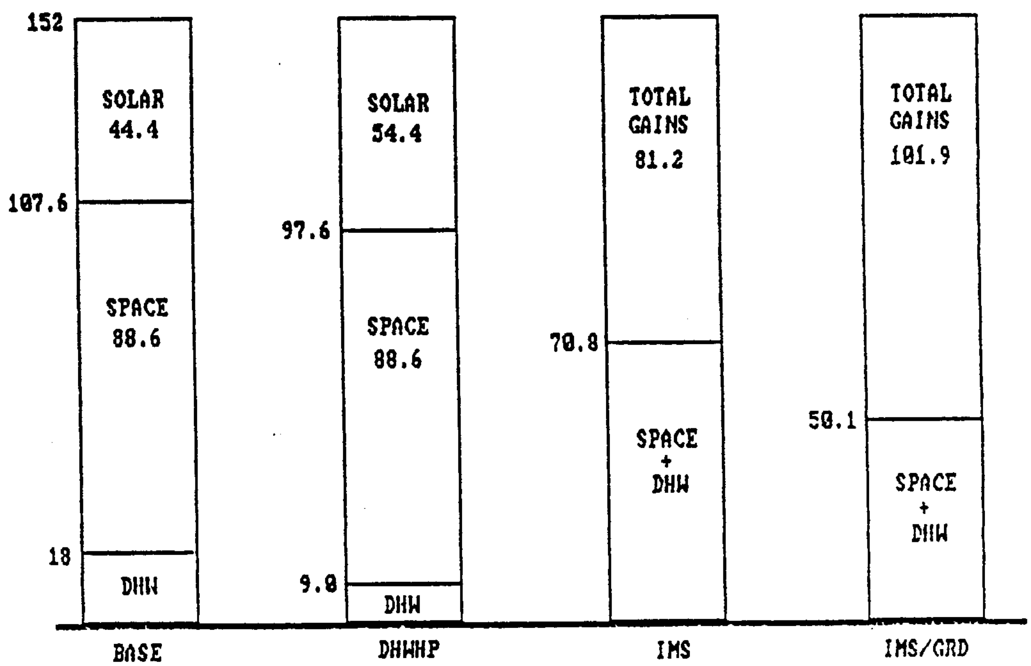


Figure 14A ENERGIES SUPPLYING TO NET LOAD FOR 3 SYSTEMS
NBC House, GFR = 15%, Ottawa

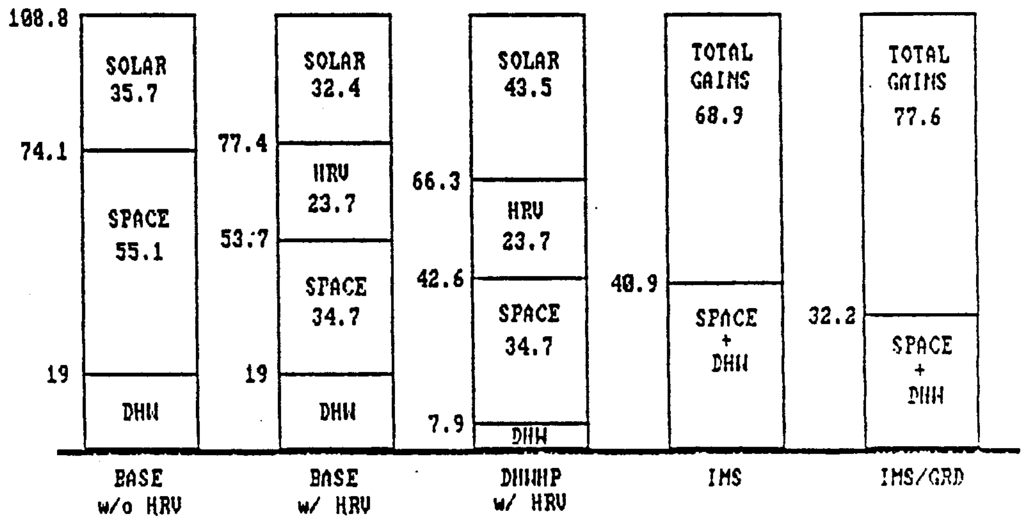


Figure 14B ENERGIES SUPPLYING TO NET LOAD FOR 3 SYSTEMS
EE House, GFR = 15%, Ottawa

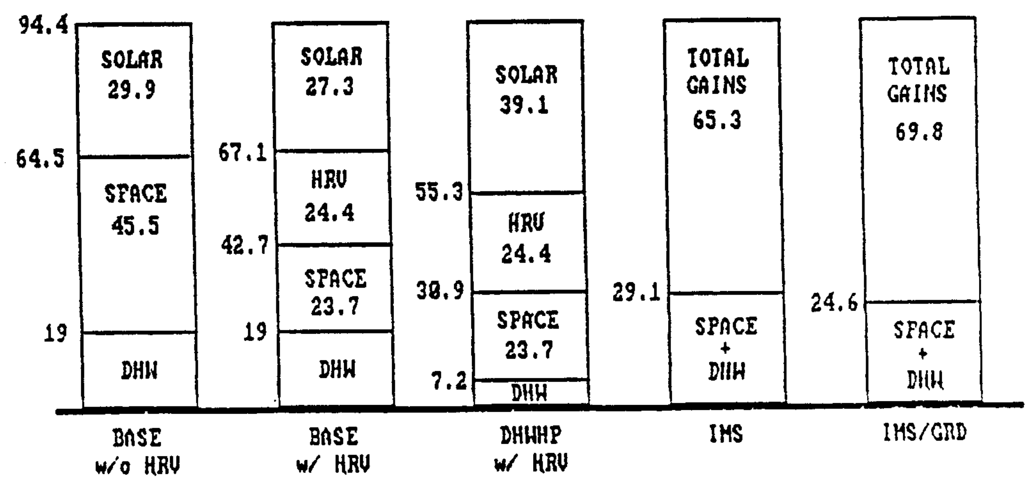


Figure 14C ENERGIES SUPPLYING TO NET LOAD FOR 3 SYSTEMS
SEE House, GFR = 15%, Ottawa

HOUSE	GFR 10%				15%				GFR 20%			
	TOTAL HEATING LOAD GJ	DHW CREDIT GJ	NET HEATING LOAD GJ	TOTAL GAINS GJ	TOTAL HEATING LOAD GJ	DHW CREDIT GJ	NET HEATING LOAD GJ	TOTAL GAINS GJ	TOTAL HEATING LOAD GJ	DHW CREDIT GJ	NET HEATING LOAD GJ	TOTAL GAINS GJ
NBC	101.6	8.6	93.0	45.3	107.6	10.0	97.6	54.4	108.1	10.4	97.7	64.8
EE w/o HRV	72.7	9.9	62.8	38.9	74.1	10.4	63.7	46.1	72.9	10.9	62.0	51.4
EE w/ HRV	53.1	10.6	42.5	35.6 (59.2)	53.7	11.1	42.6	43.5 (67.2)	51.7	11.5	40.2	49.5 (73.2)
SEE w/o HRV	66.2	11.6	54.6	39.8	64.5	11.7	52.8	41.6	63.6	11.8	51.8	46.2
SEE w/ HRV	43.7	11.6	32.1	34.2 (58.4)	42.7	11.8	30.9	39.1 (63.5)	42.2	11.8	30.4	43.3 (67.6)

* Numbers in brackets include HRV contribution

TABLE 6A - GLAZING APERTURE VARIATION
System 1 - DHWHP

HOUSE	GFR 10%				GFR 15%				GFR 20%			
	TOTAL HEATING LOAD GJ	ENERGY CREDIT GJ	NET HEATING LOAD GJ	TOTAL GAINS GJ	TOTAL HEATING LOAD GJ	ENERGY CREDIT GJ	NET HEATING LOAD GJ	TOTAL GAINS GJ	TOTAL HEATING LOAD GJ	ENERGY CREDIT GJ	NET HEATING LOAD GJ	TOTAL GAINS GJ
NBC	101.6	34.4	67.2	71.1	107.6	36.8	70.8	81.2	108.1	39.3	68.8	93.7
EE	72.7	31.9	40.8	60.9	74.1	33.0	40.9	68.9	72.9	34.5	38.4	75.0
SEE	66.2	36.3	29.9	60.6	64.5	35.4	29.1	65.3	63.6	35.7	27.9	70.1

TABLE 6B - GLAZING APERTURE VARIATION
System 3 - IMS

HOUSE	GFR 10%				GFR 15%				GFR 20%			
	TOTAL HEATING LOAD GJ	ENERGY CREDIT GJ	NET HEATING LOAD GJ	TOTAL GAINS GJ	TOTAL HEATING LOAD GJ	ENERGY CREDIT GJ	NET HEATING LOAD GJ	TOTAL GAINS GJ	TOTAL HEATING LOAD GJ	ENERGY CREDIT GJ	NET HEATING LOAD GJ	TOTAL GAINS GJ
NBC	101.6	54.7	46.9	91.4	107.6	57.5	50.1	101.9	108.1	56.9	51.2	111.3
EE	72.7	40.6	32.1	69.6	74.1	41.9	32.2	77.6	72.9	42.2	30.7	82.7
SEE	66.2	40.2	26.0	64.4	64.5	39.9	24.6	69.8	63.6	39.8	23.8	74.2

TABLE 6C - GLAZING APERTURE VARIATION

System 4 - IMS/GRD

HOUSE	DESIGN HEAT LOAD kW	HEAT PUMP CAPACITY kW	TOTAL* HEAT LOAD GJ	VOLUME m^3	UA $W/^oC$
<u>Ottawa</u>					
NBC	16	12	108	20	1000
EE	11.5	8.5	74	10	700
SEE	8	6	68	4	500
<u>Winnipeg</u>					
NBC	18	13.5	130	29	1100
EE	12.5	9.5	88	16	750
SEE	9.5	7	68	9	550
<u>Vancouver</u>					
NBC	10.5	8	83	10	650
EE	7	5.5	55	2	475
SEE	5.5	4	41	<0.5	350

*Includes 19 GJ annual DHW load

TABLE 7 - GROUND STORAGE SIZING FOR 3 HOUSES

HOUSE	OTTAWA 4400 CDD				WINNIPEG 5400 CDD				VANCOUVER 2800 CDD			
	TOTAL HEATING GJ	DHW CREDIT GJ	SOLAR TRANS. GJ	A/C* CREDIT GJ	TOTAL HEATING GJ	DHW CREDIT GJ	SOLAR TRANS. GJ	A/C* CREDIT GJ	TOTAL HEATING GJ	DHW CREDIT GJ	SOLAR TRANS. GJ	A/C* CREDIT GJ
NBC	92.6	10.0	58.2	4.6	119.0	11.0	95.7	4.1	74.4	8.7	39.1	3.2
EE w/o HRV	63.7	10.4	47.5	3.8	73.1	12.1	53.7	3.2	41.3	9.4	31.6	2.7
EE w/ HRV	42.6	11.1	47.5	4.3	47.2	12.8	53.7	3.6	27.2	10.2	31.8	3.0
SEE w/o HRV	52.8	11.7	51.4	2.4	66.1	12.7	57.4	2.2	35.5	10.8	34.4	1.3
SEE w/ HRV	30.9	11.8	51.4	2.8	39.3	12.7	57.4	2.5	21.6	10.8	34.4	1.5

* - assumed to be 1/2 of cooling energy consumption of base case

TABLE 8A - CLIMATE VARIATION, GFR = 15%
System 1 - DHWHP

HOUSE	OTTAWA 4400 CDD				WINNIPEG 5400 CDD				VANCOUVER 2800 CDD			
	TOTAL HEATING LOAD GJ	ENERGY CREDIT GJ	SOLAR TRANS. GJ	A/C CREDIT GJ	TOTAL HEATING LOAD GJ	ENERGY CREDIT GJ	SOLAR TRANS. GJ	A/C CREDIT GJ	TOTAL HEATING LOAD GJ	ENERGY CREDIT GJ	SOLAR TRANS. GJ	A/C CREDIT GJ
NBC	70.8	35.4	58.2	9.1	92.1	37.9	65.7	8.1	44.5	38.6	39.1	6.3
EE	40.9	33.2	47.5	7.6	54.6	33.7	53.7	6.4	26.7	27.9	31.1	5.3
SEE	29.1	35.4	51.4	4.8	39.6	39.2	57.4	4.4	19.3	27.0	31.8	2.6

TABLE 8B - CLIMATE VARIATION, GFR = 15%
System 3 - IMS

HOUSE	OTTAWA 4400 CDD				WINNIPEG 5400 CDD				VANCOUVER 2800 CDD			
	TOTAL HEATING LOAD GJ	ENERGY CREDIT GJ	SOLAR TRANS. GJ	A/C CREDIT GJ	TOTAL HEATING LOAD GJ	ENERGY CREDIT GJ	SOLAR TRANS. GJ	A/C CREDIT GJ	TOTAL HEATING LOAD GJ	ENERGY CREDIT GJ	SOLAR TRANS. GJ	A/C CREDIT GJ
NBC	50.1	57.5	58.2	9.1	62.7	67.3	65.7	8.1	34.8	48.3	39.1	6.3
EE	32.2	41.9	47.5	7.6	40.3	48.0	53.7	6.4	15.8	23.2	31.1	5.3
SEE	24.6	39.9	51.4	4.8	30.7	48.1	57.4	4.4	17.5	28.8	31.8	2.6

TABLE 8C - CLIMATE VARIATION, GFR = 15%
System 4 - IMS/GRD

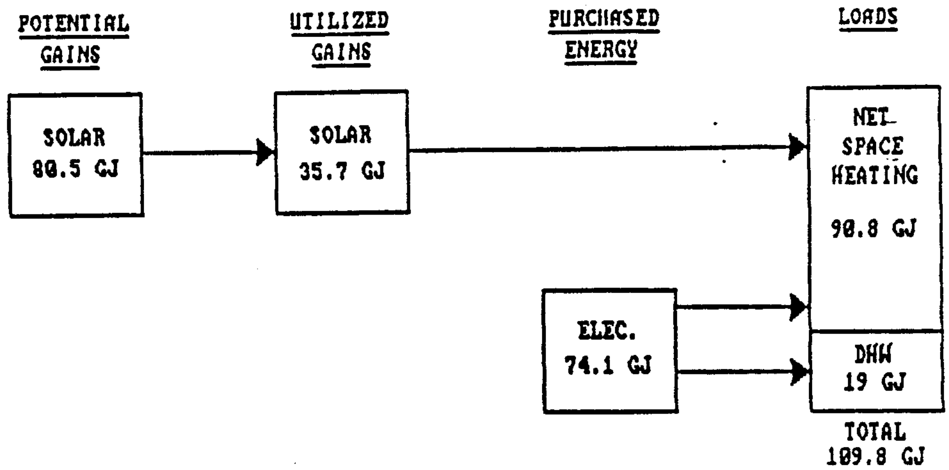


FIGURE 15A ENERGY FLOWS - BASE CASE MECHANICAL SYSTEM

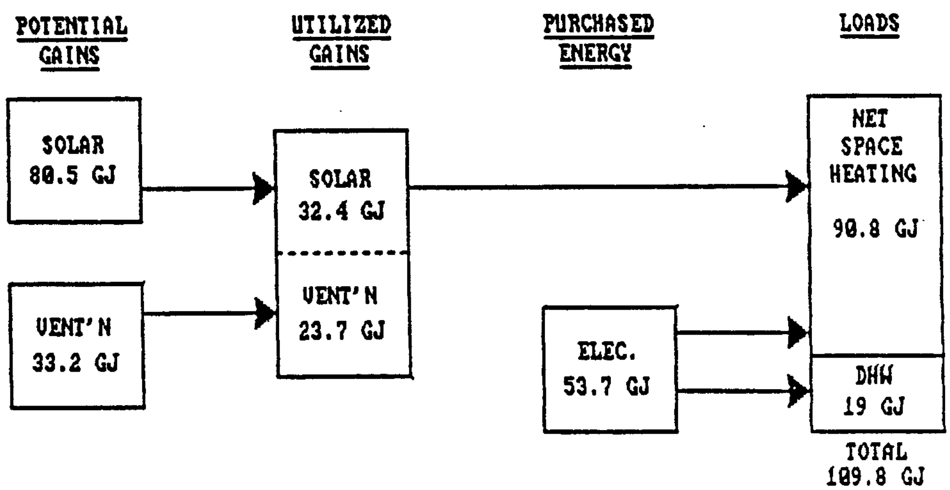


FIGURE 15B BASE CASE MECHANICAL SYSTEM WITH HRV

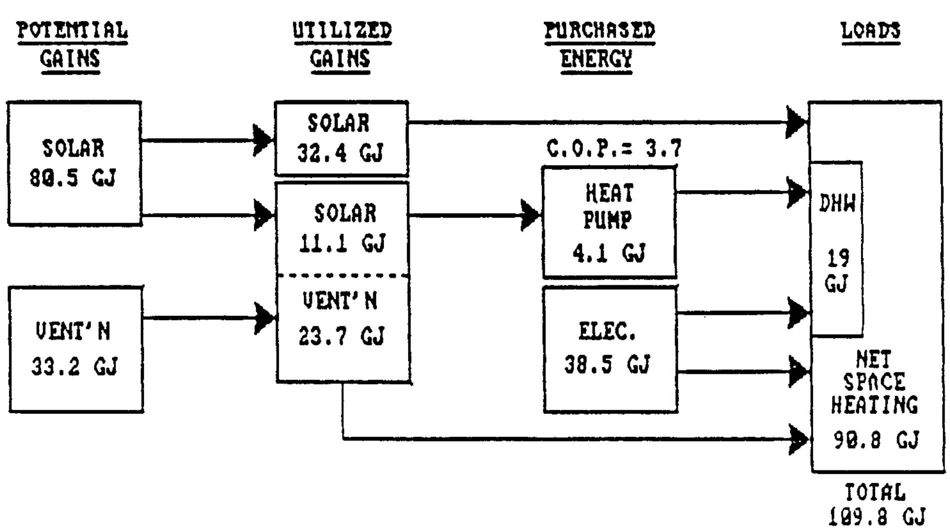


FIGURE 15C DHW HEAT PUMP WITH PREHEAT TANK AND HRV

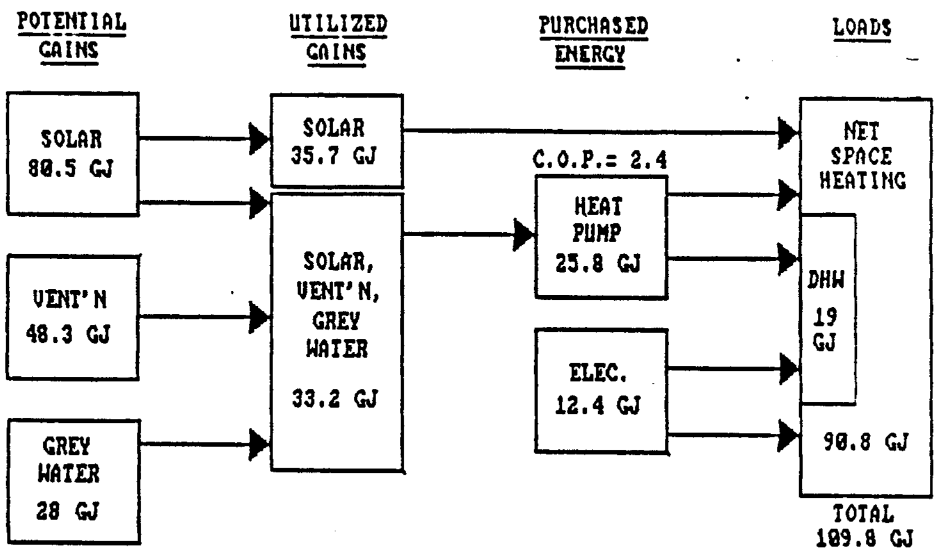


FIGURE 15D INTEGRATED MECHANICAL SYSTEM/ICE PHASE CHANGE STORAGE

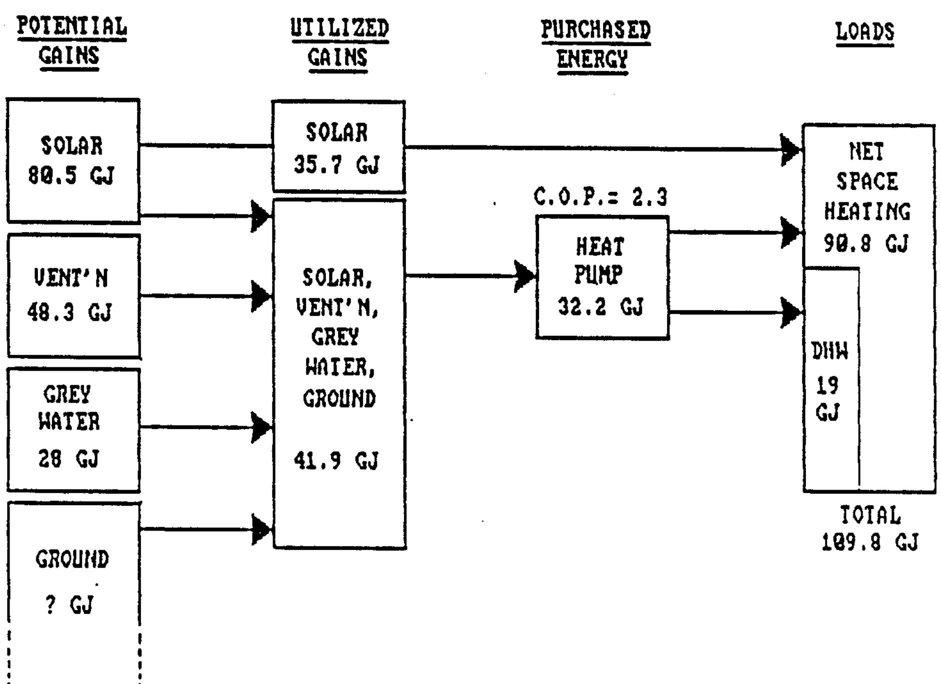


FIGURE 15E INTEGRATED MECHANICAL SYSTEM/GROUND PHASE CHANGE STORAGE

4.0 CAPITAL COSTS AND COST EFFECTIVENESS

4.1 Capital Costs of Systems

The capital costs of the various systems modelled are described below. Note that all costs are installed equipment costs. These costs are representative of typical prices and labour rates derived from various surveys of mechanical contractors and our files. The base case system includes an electric forced-air furnace at $650 (plus $150 labour electrical hook-up), an electrical DHW tank at $375 (plus $100 plumbing and electricals).

4.1.1 System 1: Indoor Air Heat Pump/DHW Supply

This system requires the addition of a 3 kW water heating heat pump and a 250 L pressurized preheat tank. The heat pump costs $1250 (plus $150 labour) and the insulated preheat tank costs $300 (plus $50 labour). Note that this system does not provide full summer cooling, but the cost of a 2 kW cooling capacity air conditioner can be credited.

Add:	Heat pump	$1400
	Preheat tank	350
	Net Incremental Cost	$1750
Delete:	Window style air conditioner	($1000)
NET INCREMENTAL COST		$ 750

4.1.2 System 2: Indoor Air Heat Pump/Space and DHW Supply, versions A and B

These systems replace the furnace, hot water tank and a central ventilation system. Version B is similar to an exhaust air heat pump presently available for about $3000, but requires upgrading to provide full space heating. In a mature but low volume market an installed cost of $4500 (includes $200 labour) seems reasonable for this kind of heat pump. Version A would be a larger heat pump but less complicated without the heat recovery coil - assume same cost as Version B. The heat recovery option may run around $500 including fan-coil, housing, etc. A 10,000 L non-pressurized tank, insulated for storing warm water, would cost about $2000 to assemble on-site. Version A can be assumed to replace central air conditioning ($2200) while Version B has partial air conditioning, assumed similar to a window unit ($1000).

		A	B
Add:	Indoor Air Heat Pump	$4500	4500
	Storage Tank	2000	2000
Delete:	Electric Furnace	($ 800)	($ 800)
	Electric DHW Tank	(450)	(450)
	Central Ventilation System	(1200)*	(1200)*
	HRV	(2200)*	(2200)*
	A/C (A:central, B:window)	(2200)	(1000)
NET INCREMENTAL COSTS;			
	no A/C, no HRV	$4050	4050
	no A/C, w/ HRV	3050	3050
	w/ A/C, no HRV	1850	3050
	w/ A/C, w/ HRV	850	2050

*one or the other, depending on base case

4.1.3 System 3: Integrated Mechanical System/Ice Storage

This system replaces the furnace, DHW tank, central ventilation or HRV, and air conditioning. In terms of sizing this system supplies full heating demand for the SEE house but the EE and NBC houses would need additional backup supplied to the DHW tank and a larger space fan coil output. This additional cost is assumed to increment at a rate similar to the forced air furnace and does not enter into the cost calculation. In rough figures, the $6000 installed cost includes the ice tank and the grey water heat exchanger at $750 each. The heat recovery fan coil would be about $500.

Add:	IMS (incl. ice storage tank)		$6000
Delete:	Electric Furnace	($ 800)	
	Electric DHW Tank	(450)	
	Central Ventilation System	(1200)*	
	HRV	(2200)*	
	A/C	(2200)	
NET INCREMENTAL COSTS;			
Base Case with:	no A/C, no HRV	$3550	
	no A/C, w/ HRV	2550	
	w/ A/C, no HRV	1350	
	w/ A/C, w/ HRV	350	

*One or other, depending upon base case

4.1.4 System 4: Integrated Mechanical System/Ground Storage

This system is similar to System 3, but the ice tank is replaced by an in-ground thermal storage with heat exchange tubes (assume 6 W/^{o}C per metre of length). The cost is estimated as follows:

Heat Exchanger Tubing	$ 3.50/m of pipe (includes manifold)
Insulation ($3/RSI-m^2)	$ 10/m^2 of area or $6.67/m^3 storage

Liner	\$ 50 + 1.25/m^3 of vol.
Excavation and Backfill	\$400
Installation Labour	\$ 50

Therefore, Total Cost = (\$7.92 x Volume) + (\$3.50 x Length) + \$500

For the three storage sizes used in the simulations, the costs for Ottawa climate are as follows:

House	Volume	UA	Pipe Length	Cost
SEE	4 m^3	500 W/^oC	83.3 m	\$ 825
EE	10 m^3	700 W/^oC	116.7 m	\$1000
NBC	20 m^3	1000 W/^oC	166.7 m	\$1250

The cost of System 3 is \$6000 which includes the ice tank at a cost of \$750. Therefore, the ground storage has an incremental cost ranging from \$75 for the smallest storage size to \$500 for the 20 m^3 storage.

The heat pump component of System 3 (IMS) has a nominal capacity of 5 kW. For the IMS/GRD system the heat pump needs to be sized according to the capacities listed in Table 7. The incremental costs for these heat pump sizes is as follows:

HOUSE	DESIGN HEAT LOAD	HEAT PUMP CAPACITY	INCR. COST
SEE	8 kW	6 kW	\$100
EE	11.5 kW	8.5 kW	\$350
NBC	16 kW	12 kW	\$700

As with System 3, the incremental cost for backup capacity is similar to that for the forced air furnace sizing and does not enter into the incremental cost calculation. Including the storage costs, the system costs are as follows:

			NBC	EE	SEE
Add:	IMS w/Ground Storage		\$7200	\$6600	\$6175
Delete:	Electric Furnace	(\$ 800)			
	Electric Water Heater	(450)			
	Central Exhaust System	(1200)*			
	HRV	(2200)*			
	A/C	(2200)			

NET INCREMENTAL COSTS:		NBC	EE	SEE
Base Case with:	No A/C, no HRV	\$4750	\$4150	\$3725
	No A/C w/ HRV	-	3150	2725
	W/ A/C, no HRV	2550	1850	1525
	W/A/C, w/ HRV	-	1950	525

* One or other, depending upon base case

4.1.5 System 5: Heat Pump Coupled Rockstore

This storage medium consists of a structural box containing about 10 m^3 of washed river gravel and has a damper system allowing for reversible airflow, similar to one installed in the Toller-Tener house in Ottawa. The cost of such an installation is about $2500. Air-source heat pumps range from $4000 to $4500, depending on the output requirement of the heat pump:

		NBC	EE/SEE
Add:			
	Heat Pump Coupled Rockstore	$7000	$6500
Delete:	Electric Furnace	($ 800)	
	Air Conditioner	(2200)	
NET INCREMENTAL COST:		NBC	EE/SEE
		$4000	$3500

* One or other, depending upon base case

4.2 Cost Effectiveness

The cost effectiveness of the systems under study were evaluated using the traditional simple payback method, i.e. how many years are required to pay off the incremental capital cost with energy cost savings, and a five-year total cost method which calculates the total of any incremental capital cost and five years of energy costs. In order to keep the calculation simple, no interest or inflation rates are used. The second method tends to give a better overview of cost implications compared to the payback method which can be confusing, eg. when capital cost is less than the base case system. For the purposes of cost effectiveness calculations, it is assumed that air conditioning would be a desirable feature when comparing systems. Thus, except for Systems 1 and 3B, incremental costs assume to displace the cost of central air conditioning ($2200). Since Systems 1 and 3B have less than full cooling capacity, these systems can be compared to a base case with a 2 kW window air conditioner priced at $1000. As a rough energy estimate, this air conditioning unit displaces 1/2 of the annual air conditioning energy in the respective base case.

Table 8 summarizes the incremental capital cost for each of the systems when compared to the base case electric furnace system without heat recovery (see section 4.1). The tables all include an HRV (seasonal effectiveness = 75%) for comparison. In Table 9 the energy savings from Table 5 have been converted to cost savings at a rate of $16 per GJ for electric energy supply. Dividing the incremental capital cost for each system by the energy cost savings gives the payback periods in years (Table 10).

Table 11 gives the five-year energy costs (includes air conditioning energy consumption, as applicable) for the various systems and houses in Ottawa. Electricity was priced at $16/GJ or $0.06/kWh while high efficiency gas is assumed to cost $10/GJ of output. The base case and DHWHP systems gas combustion equipment includes space and (back-up) DHW heating. The other systems (ie. IAHP, IMS, IMS/GRD) are assumed to be all electric even if the comparative base case is gas-fired. All air conditioners are run on electricity.

Table 12 gives the total of the five year energy cost plus the equipment incremental cost. For the gas option a base case mechanical system consisting of combustion equipment for space and water heating for tight envelope construction (i.e. sealed combustion furnace with induced draft water heater or a combination space/hot water boiler). An installed incremental cost of $2500 for gas-fired equipment is used in Table 12 for the base case and DHWHP systems to generate the total five year costs. The DHWHP also has combustion equipment as the heating supply, but the other systems are totally electric. Gas cost is assumed to be $10/GJ of delivered energy (i.e. after 90% efficiency corrections) while air conditioning and heat pump functions remain electric at $16/GJ.

In Table 9 it can be seen that the heat pump coupled rockstore (HP/RS) has low energy cost saving but high incremental cost and is not considered feasible for further analysis. Similarly, the DHWHP without a heat recovery ventilator (HRV) has poor cost effectiveness (See Tables 10 and 11). As well, the two versions of the indoor air heat pumps (IAHP) exhibit marginal cost effectiveness. Thus the DHWHP in combination with HRV, the integrated mechanical system (IMS) with isolated water/ice phase-change storage and the integrated mechanical system with ground phase-change storage (IMS/GRD) are the most likely to yield a good return on the increased initial cost over a conventional system.

	HRV	DHW	DHW w/HRV	IAHP A/B	IMS	IMS/GRD	HP/RS
NBC	-	750	-	1850/3050	1350	2550	4000
EE	1000	750	1750	1850/3050	1350	1950	3500
SEE	1000	750	1750	1850/3050	1350	1525	3500

TABLE 8 - INCREMENTAL CAPITAL COSTS OF VARIOUS SYSTEMS (Cdn $) - Electric Base Case

	HRV	DHW	DHW w/HRV	IAHP A/B	IMS	IMS/GRD	HP/RS
NBC	-	235	-	180/500	710	1065	110
EE	310	225	565	210/410	650	790	45
SEE	335	225	575	- / -	645	715	-

TABLE 9 - ENERGY COST SAVINGS FOR VARIOUS SYSTEMS (GFR = 15%) (Electricity = $16/GJ or $0.06/kWh)

	HRV	DHWHP	DHWHP w/HRV	IAHP A/B	IMS	IMS/GRD
NBC	-	3.2	-	10.3/6.1	1.9	2.4
EE	3.2	3.3	3.1	8.9/7.4	2.1	2.5
SEE	3.0	3.3	3.0	- / -	2.1	2.1

TABLE 10 - PAYBACK PERIODS OF INCREMENTAL COSTS FOR VARIOUS SYSTEMS (in years) - Elec. Base Case

		BASE*	HRV	DHWHP	DHWHP w/HRV	IAHP A/B	IMS	IMS/GRD
NBC	ELEC	$9335	-	$7810	-	$8430/6470	$5665	$4010
	GAS(90%)	6110	-	4995	-	8430/6470	5665	4010
EE	ELEC	6535	4985	5095	3410	5495/4190	3270	2575
	GAS(90%)	4315	3375	3305	2260	5495/4190	3270	2575
SEE	ELEC	5545	3865	4225	2470	- / -	2330	1970
	GAS(90%)	3610	2585	2810	1715	- / -	2330	1970

*** - with central air conditioner**

TABLE 11 - 5-YEAR ENERGY COSTS FOR ELECTRIC AND HIGH EFFICIENCY GAS FURNACES

		BASE*	HRV	DHWHP	DHWHP w/HRV	IAHP A/B	IMS	IMS/GRD
NBC	ELEC	9335(C)	-	-	-	10280/10520	7015	6560
		8975(W)	-	8560	-	- /9520	-	-
	GAS(90%)	8610(C)	-	-	-	10280/ -	7015	6560
		8240(W)	-	8245	-	- /9520	-	-
EE	ELEC	6535(C)	5985	-	-	7345/ -	4620	4525
		6230(W)	5640	5845	5160	- /7240	-	-
	GAS(90%)	6815(C)	6875	-	-	7345/ -	4620	4525
		6510(W)	6529	6555	6510	- /7240	-	-
SEE	ELEC	5545(C)	4865	-	-	- / -	3680	3495
		5350(W)	4640	4975	4220	- / -	-	-
	GAS(90%)	6110(C)	6083	-	-	- / -	3680	3495
		5915(W)	5859	6060	5965	- / -	-	-

* - (C); central air conditioner ($2200) included in base case
(W); window air conditioner ($1000) included in base case

TABLE 12 - TOTAL COSTS FOR VARIOUS SYSTEMS
(Incremental Capital Costs + 5 year Energy Costs)

5.0 DESIGN OF HEAT PUMP/THERMAL STORAGE COMBINATIONS

5.1 Sizing of Components

For the more promising the following component sizing can be suggested.

5.1.1 System 1: Indoor Air Heat Pump/DHW Supply (DHWHP)

While hot water heat pumps can supply any size hot water load, the technology addressed in this study is basically designed to replace the heating output of a standard domestic hot water tank, typically 1.5 to 4.5 kW. An electric 275 L(60 Imp. Gal) tank would have 3 kW of heating. Heat pump manufacturers (e.g. E-TECH, DEC Thermastor, Fedders) have models available in this size for residential installations.

The preheat tank used in the simulation has a volume of 250 L or about the same size as the standard hot water tank. Larger tanks were modelled but were found to exhibit negligible energy savings. Essentially this system doubles the amount of water storage with the second tank acting as a thermal buffer between the times when cooling is available and DHW demands occur.

5.1.2 System 3: Integrated Mechanical System/Ice Phase-Change Storage (IMS)

A system providing all the integrated functions described has been developed by our firm and is presently undergoing field testing. The heat pump was developed as a one-package heating/cooling/heat recovery system for energy-efficient houses with total (space + DHW) heating loads of less than 70 GJ, which have a peak demand of 6 kW. This type of house allows compatability between heat pump output and heat extraction for capturing gains. Without reducing the energy savings, larger heating loads can be supplied with additional back-up heat or by increasing the available gains, e.g. by the use of ground storage (see System 4).

The ice storage was optimized at 450 L and is configured as a 0.75 diameter tank 1.2 m high. The heat exchanger consists of 40 m of copper tubes resulting in a total extraction rate of 250 W/^{o}C.

5.1.4 System 4:
Integrated Mechanical System/Ground Phase-Change Storage (IMS/GRD)

This system is similar to System 3 but has an in-ground ice storage, thus being able to take advantage of significantly more gains by extracting heat from surrounding soil, as well as from ventilation air, grey water and space air. A packaged system does not exist at this time, therefore the heat pump is either a custom-engineered system using a water source heat pump or an scaled-up version of System 3 to satisfy the heating load required.

The ground storage configurations used in this study (refer to Fig. 3) have a constant vertical dimension of 1.5 m and the top of the storage is located 1 m below the ground surface. Therefore, the area of the storage in plan changes proportionately with the volume. The top of the storage is highly insulated and the insulation extends 1.5 m beyond the storage perimeter. The heat exchanger is assumed to be a multi-tube copper array with a heat transfer coefficient of 6 W/m of tube length. Alternate configurations, such as pool solar collectors, could be utilized as in-ground heat exchangers. The volumes and UA values would be calculated as given in Section 3.4.4 to yield results like those in Table 7.

5.2 Energy Credit Calculations

Simple energy analysis tools, whether manual methods or month-by-month computer programs (e.g. HOT2000), do not have sufficient flexibility to account for the mechanical systems described in this study. In the case of the R2000 Housing Program, which uses a total energy budget, new technologies can only be credited with energy savings if a modelling procedure compatible with HOT2000 can be carried out. In this section, procedures for establishing the energy benefits of the more promising heat pump/thermal storage combinations are proposed.

5.2.1 System 1: Indoor Air Heat Pump/DHW Supply

Previous work (Ref. 3) looked at using building thermal storage in combination with air recirculation to provide hot water via a DHW heat pump. In this strategy cooling occurred whenever there was a DHW demand. This increased the heat load in the winter but, with the resulting increase in solar gain utilization and reduced summer DHW energy consumption, this strategy was found to be economically attractive. Comfort was maintained by the significant building mass and modest passive solar apertures. The present study looks at isolated thermal storage and larger passive solar apertures. The strategy chosen, therefore, is to use the DHWHP as a cooling system only when the indoor temperature is above 23°C , thus avoiding excessive cooling of the living space. The simulations showed that this strategy has relatively little effect on the space heating load and reduces the energy consumption strictly on the DHW heat pump C.O.P. as a result of the average annual space temperature. The two factors that can affect the indoor temperature are the house energy-efficiency level (ie. space load) and solar glazing aperture. For GFR = 15%, as the space heating load drops from 90 GJ to 35 GJ the net C.O.P. increases from 2.1 to 2.4. As the glazing aperture reduces the net C.O.P. goes down accordingly, suggesting a decrease in the average indoor temperature or available excess solar gains. The C.O.P. performance follows an exponential curve of the form:

$$\text{Net COP} = COP_{min} + ([COP_{ave} - COP_{min}]\ e^{-SHL/TSG})$$

where COP_{min} is the lower limit C.O.P. (i.e. at an infinite Space Heating Load or no excess gains, so that heat pump operation occurs only when outdoor temperature is above 23°C). The amount this value is above C.O.P. = 1 is about 7.5% of the cooling C.O.P. (i.e. C.O.P. - 1). In our case the increment is 7.5% x (3.75 - 1) or 0.2. Therefore, COP_{min} limit = 1 + 0.2 = 1.2

COP_{ave} is the maximum C.O.P. at the annual average conditions that the heat pump experiences - in this case, assumed to operate at an average indoor temperature of 25°C - corrected for standby losses in the DHW tank (i.e. minimum back-up, MB) which are supplied by resistance heating (about 20 W or 0.6 GJ/yr for a well-insulated tank):

$$COP_{ave} = \frac{\text{DHW LOAD}}{\text{MB} + \dfrac{\text{DHW LOAD} - \text{MB}}{COP_{25^oC}}}$$

For our system, $COP_{25^oC} = 3.75$:

$$COP_{ave} = \frac{19 \text{ GJ}}{0.6 + \dfrac{19 - 0.6}{3.75}} = 3.5$$

SHL is space heating load of a conventional heating system.

TSG is the <u>annual</u> transmitted solar gains to space.

Thus for the heat pump performance used in this study:

$$\text{Net COP} = COP_{min} + ([COP_{ave} - COP_{min}]e^{-SHL/TSG})$$

$$= 1.2 + 2.3e^{-SHL/TSG}$$

The simulations show little change in space heating load when compared to the base case (which has air conditioning in addition to passive cooling). Thus as an estimate of energy credits for the DHW heat pump/preheat tank system, the DHW energy consumption can be reduced by:

$$\text{Energy Credit} = \frac{\text{Net COP} - 1}{\text{Net COP}} \times \text{DHW Load}$$

This system algorithm can be compared to a system which operates whenever there is DHW demand and uses building construction as storage instead preheat tank thus cooling the living space. In winter the space heating load is increased, which is partially offset by the increase use made of the transmitted solar gains. Note that all DHW heating loads, including standby losses, are satisfied by heat pump operation. An algorithm was previously prepared for the R2000 Program (Ref. 7). The procedure was as follows:

1. Run HOT2000 with conventional space and DHW systems but reduce internal gains by cooling imposed on a daily basis:

   Internal gain reduction = [(COP - 1)/COP] x daily DHW load

2. Once the results from the HOT2000 run are obtained the DHW energy consumption is reduced by the same amount:

   DHW credit = [(COP - 1)/COP] x DHW load

For the EE House (w/o HRV, GFR = 15%) the two approaches compare as follows:

	Preheat tank	DHW Demand
DHW Credit (GJ)	10.4	14.0
Space Increase (GJ)	-	5.8
Net Credit (GJ)	10.4	8.2

When the heat pump is located in a sunspace and operating on cooling demand, the conditions the heat pump experiences are similar, independent of the heating load of the rest of the house. As well, there is only a slight improvement in performance with larger solar gains, apparently because free cooling maintains similar temperatures. Typically, the heat pump located in the sunspace utilizes 11.7 GJ of cooling at a C.O.P. of 3.3, which corresponds to an annual average temperature of 16°C in the sunspace during heat pump operation. The heat pump supplies, therefore, 16.8 GJ of DHW load. The DHW credit is simply 11.7 GJ. The amount of backup required would vary depending on the value of COP_{16^oC} for the system chosen.

5.2.2. System 3:
Integrated Mechanical System/Ice Phase-Change Storage (IMS)

This system takes its gains from grey water, ventilation air as well as excess solar gains and, as such, is not as highly dependent on transmitted solar gains. From the simulation results it is evident that the heat pump operates at a consistent C.O.P. of about 2.2; however, the amount of energy it supplies toward total heating varies. There is an intrinsic upper limit which the heat pump is capable of supplying, i.e. the maximum annual gains of about 40 GJ. The form of the equation for calculating the heat pump contribution is:

$$\text{HP Heating (GJ)} = \frac{COP}{COP - 1} \times 40\ \text{GJ}(1 - e^{-THL/A})$$

$$= 72.5(1 - e^{-THL/50})$$

where THL is the total heating load (Space + DHW) - in GJ

A is an empirical coefficient equal to 50

The Energy Credit is further calculated as follows:

$$\text{Energy Credit} = \frac{COP - 1}{COP} \times \text{HP Heating}$$

$$= \frac{1.2}{2.2} \times \text{HP Heating}$$

For the SEE house THL is the load the heat pump needs to supply, ie. after preheating is credited:

For example, for GFR = 15%,

Total Load = 64.5 GJ
Preheat Credit = 8.8 GJ
HP System Load = 55.7 GJ

Therefore, HP Heating = 0.9 x 55.7 GJ = 48.7 GJ

and, HP Energy Credit = 1.2/2.2 x 48.7 = 26.6 GJ
Total Energy Credit = 27.3 GJ + 8.8 GJ = 36.1 GJ

5.2.3 System 4:
Integrated Mechanical System/Ground Storage (IMS/GRD)

This system is sized to supply 95% or more of the total heating load and thus incurs a small amount of backup. The heat pump operates at an average entering glycol temperature of $-4^{o}C$, or a C.O.P. of 2.4. The backup reduces this performance to a C.O.P. of 2.2. The energy credit calculation is again not highly influenced by the amount of solar gain:

$$\text{Energy Credit} = \frac{COP - 1}{COP} \times THL$$

$$= \frac{1.2}{2.2} \times THL$$

where THL is the total heating load (space + DHW) in GJ

COP is the net C.O.P. for the heat pump at the average annual entering glycol temperature (eg. $-5^{o}C$). The net C.O.P. is about 95% of $COP_{-5^{o}C}$ from manufacturer's data.

6.0 REVIEW OF ISSUES RELATED TO STANDARDS

Basic heat pump performance ratings are straight forward and simply state how much energy is required to supply a certain function, eg. space cooling or heating. For air conditioning the performance is usually given at a design condition such as 35°C outdoor temperature and a design indoor temperature of 26°C. The performance value is either cooling output at a net C.O.P., which includes all energy consumed to deliver the cooling or an Energy Efficiency Ratio (EER):

$$EER = \frac{\text{Cooling Output}}{\text{Elec. Input}}$$

For heating heat pumps performance is rated at different outdoor conditions for outdoor air source heat pumps or different water temperatures for water or ground source heat pumps. In other words, it is easy to test heat pumps to extract heat (evaporator side) and supply heat (condenser side) at different external conditions.

In the case of integrated mechanical systems, the basic function of the heat pump is the same, responding simply to the temperatures it sees in the storage tank and/or the hot water tank. However, due to the gains being collected the storage tank temperatures becomes a dynamic function independent of outdoor temperature or heating/cooling demand.

A suggested methodology for rating integrated mechanical systems with thermal storage is as follows.

1. Determine the heat pump performance (output, C.O.P.) for the range of operating temperatures expected, particularly by the evaporator.

2. Characterize the components that gather gains or deliver heat indirectly with respect to the heat pump operation. In the IMS-type systems this refers to the heat exchange performance of the cooling coil, ventilation heat recovery coil, grey water heat exchanger and ice tank or ground heat exchanger. In the air conditioning heat pumps, with storage on the warm side, this would refer to the characteristics of heat delivery from the preheat tank via a standard DHW tank to the heating coil.

3. Once these parameters are known there are two choices as to how to proceed.

   A) The information discussed so far becomes the published information for the heat pump/thermal storage system and is used in a rating procedure. A certified analysis tool, like ENERPASS, with the capability to accept this information, would be used to derive the energy performance, back-up sizing, etc. for the climate and house design requirements.
   B) The second choice is to develop equipment-specific energy benefit algorithms, perhaps similar to those proposed in this study. Basically, the procedure in A above is carried out by the certification agency and published for the variables that affect the energy benefits, ie. climate, heat pump output. This is the approach the R2000 program has taken with the HABITAIR unit from Fiberglas Canada. After testing the unit under a variety of conditions, a straight energy credit is applied to the annual heat loads using conventional mechanicals.

7.0 CONCLUSIONS

This study investigated 5 generic systems of thermal storage/heat pump combinations for residential houses. This included diurnal and seasonal-scale storage; however, there is a practical limit for indoor storage volumes in terms of how much room it can occupy. Thus the typical large storage in this study was sized at 10 m^3 or 10,000 L. Larger storages can yield better energy performance and true seasonal storage behaviour, as in the case of system 5, but due the impracticality mentioned and increased cost Systems 3A and B (Indoor Air Heat Pump) and System 5 (Heat Pump Coupled Rockstore) with larger storage volumes are not considered feasible.

Of the systems studied, the most cost-effective heat pump/thermal storage configurations are fully integrated mechanical systems with isolated ice storage (IMS, for houses with total heating loads - space + DHW - less than 60 GJ/yr) and with phase-change ground storage (IMS/GRD). The cost effectiveness is slightly better than HRV's on a payback basis (less than 3 years) but the cost savings realized after 5 years (including incremental capital costs) are more than 2 times as large. These systems extract heat from ventilation air, grey water, and ground, as well as utilizing excess solar gains. As such, their performance is not highly affected by varying the solar aperture. On the other hand, they are able to control overheating with glazing-to-floor ratios up to 20%, thereby allowing the designer significant architectural freedom. These two systems are able to draw from various gains to displace heating loads. If, for example, grey water heat recovery was deemed to be undesirable, the solar gains would contribute more significantly but the total savings would be lower.

The payback of a DHW heat pump extracting heat from space to offset water heating (System 1) combined with a heat recovery ventilator has a payback of just over 3 years. This is a low-cost package yielding significant ventilation and water heating energy savings ($500 per year) while providing partial air conditioning. This combination may be adequate for many houses, particularly if located on exposed sites and conscientious venting is practiced is summer. It may also be possible to design a larger air conditioning unit that heats hot water when there is a demand, otherwise acts as a central aiconditioner. This equipment would need to be developed from a technical perspective; however, as long as the improvements can be achieved for an incremental cost of $1200, then the cost effectiveness will be the same (about a 3-year payback).

System 3B, a ventilation heat recovery heat pump with partial air conditioning capability supplying to a 10 000 L tank, can realize $500 annual saving; however, it cannot be credited with air conditioning capability and, as a result, the incremental cost over a conventional system is high. There seems to be promise in developing an improved air conditioning heat pump with warm water storage that also has a ventilation heat recovery function. If the cost of such a system is not significantly higher a payback of about 4 years would be achieved.

The total 5-year cost savings (incl. incremental costs) for the three more promising configurations are as follows (electricity 16/GJ, gas $ 10/GJ):

System 1:	DHWHP w/HRV	$1100 (elec. base case only)
System 2:	IMS	$1600-2400
System 3:	IMS/GRD	$2000-2600

The energy benefits of systems can readily be characterized in terms of manufacturers' specifications under typical operating conditions. The energy benefits are subtracted from annual total (space + DHW) heating consumption established from thermal modelling with a conventional mechanical system.

Preheating of ventilation air by drawing it via the one-storey attached sunspace used in this study contributes an equivalent heat recovery effectiveness of 30% or typically 9 GJ or $145. This is additional to any exhaust air heat recovery that may be realized by an exhaust-only ventilating heat pump.

The most interesting result of this study is the phase-change ground storage modelling of a volume of saturated soil or man-made aquifer in contact with surrounding soil. Computer simulations yielded optimized annual storage capacities amounting to only 8% of the annual total (space + DHW) heating loads. This requires significantly less yard area for locating the storage, and heat exchange may be achieved simply by burying 1 or 2 large solar pool collectors or a ground heat exchanger coil similar to the helical direct expansion coil developed at NRC.

The computer modelling confirmed that improved window technology and increased solar gains allow architectural freedom to design large south facing window areas without an energy penalty. Several heat pump/thermal storage combinations are able to control summer and winter overheating while making use of increased solar gains when combined with other gains, such as ventilation heat recovery, grey water heat recovery and ground heat extraction yield impressive cost feasibility. Fields trials of these systems,in particular the artificial aquifer ground storage, should be pursued. Concurrently, work should be pursed on the development of a standard procedure for rating these types of systems and determining their energy credits.

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APPENDIX A:

HEAT PUMP/THERMAL STORAGE DESIGN AND INSTALLATION GUIDELINES

HEAT PUMP AND THERMAL STORAGE DESIGN AND INSTALLATION GUIDELINES

With increased use of high R-value glazing, large south-facing windows are becoming more popular because the energy penalty for such large apertures is reduced and discomfort due loss of radiation from the body is less of a problem. Consequently, large solar gains can be realized but if not utilized they can overheat the living space and are often returned back to outdoors by venting. In other words, as building envelopes become more energy efficient, waste heat generated by household activities and house design can at times exceed the heating demand of the house while at other times of the day significant purchased energy is needed. The solution to utilizing the available waste heat is two-fold:

1. By house or mechanical system design allow the potential waste heat to be captured. This includes the use of large passive solar apertures, cooling of house air with year round air conditioning to capture excess solar gains, extracting heat from exhaust air via a heat recovery ventilator or exhaust air heat pump, and possibly recovering heat from grey water (from sinks and showers).

2. Provide thermal storage to bridge the non-coincidence of collected gains vs. space and domestic water heating demand.

This note outlines three heat pump/thermal storage systems that take advantage of excess solar gains by operating a winter and summer air conditioning function and storing the extracted heat in a thermal storage medium for eventual supply to space and domestic water heating.

Three promising systems are described in terms of heat pump and thermal storage sizing, installation guidelines, energy performance and estimated capital cost. The major tasks of the designer are to be able to specify heat pump capacity and performance and decide on storage size, type and charging/discharging strategy. The following guidelines give information on how to choose a heat pump given manufacturer's data. The definition of coefficient of performance (C.O.P.) may vary between manufacturers or type of equipment. The definition used in this note is:

$$\text{C.O.P.} = \frac{\text{TOTAL OUTPUT (kW)}}{\text{TOTAL INPUT (kW)}}$$

where TOTAL INPUT = Compressor Input + Evaporator Fan/Pump + Condenser Pump (if existing)

Note that capacity and C.O.P. change depending on the source temperature.

Thermal storage description and sizing is also included. Storage media used are water (sensible and ice/water phase change)and a soil/water mixture (including phase change). Heat transfer media include brine (i.e. a water/glycol anti-freeze solution) or water via copper heat exchangers, and air circulation.

Energy performance calculations were developed for houses of different energy-efficiency levels and glazing areas. Once annual total (space and domestic water) heating loads and annual solar gains reaching the house interior (i.e. after it passes through the window and before any is dumped back to outside) are obtained from available design tools modelling conventional heating systems, energy credits which account for the heat pump/thermal storage contribution are easily calculated. If an unheated one-story sunspace or solarium is located on the south facade of the house, increased energy savings can be realized either by locating the cooling equipment in the sunspace or drawing outdoor ventilation air through the sunspace, thus preheating it with solar gains.

Using these systems can yield net savings up to 15000 kWh or $900 per year and total savings in five years (includes incremental system cost) up to $2500. In addition, these systems provide improved comfort control, thus complementing the new architectural styles made possible by high performance windows.

1. INDOOR AIR HEAT PUMP/DHW SUPPLY

The heat pump is located in the major solar gain space of the house. Heat extracted from indoor air by the cooling side of the heat pump when the indoor temperature rises above a setpoint (e.g. 23°C) at a few degrees lower than the cooling setpoint (e.g. 26°C). The heat produced by the heat pump is stored in a preheat tank which supplies the stendard DHW tank.

SIZING/INSTALLATION

A number of companies offer heat pumps for typical residential hot water loads. The capacity is usually similar to an electric element of a standard hot water tank, i.e. 3 kW. Assume a space air temperature as typical operating conditions.
For the preheat tank - use the same size as the DHW tank.

Locate the heat pump such that the cool outlet air does not cause discomfort, possibly in a closet with discharge at the ceiling. Plumbing would be run to a utility room, typically in the basement, where the preheat and DHW tank are located.

ENERGY PERFORMANCE

The energy savings are based on the effective coefficient of performance (C.O.P.) of the heat pump:

$$\text{Energy Savings (kWh)} = \frac{(COP - 1)}{COP} \times \text{DHW Load (kWh)}$$

where COP is the effective Coefficient of Performance taking into account the average C.O.P. of the heat pump and back-up resistance energy

The calculation of the effective C.O.P. varies with space heating load and location of heat pump. For an installation located in a direct gain space, the effective C.O.P. is derived as follows:

$$COP = COP_{min} + [COP_{max} - COP_{min}] \times e^{-SHL/TSG}$$

where COP_{min} is the minimum C.O.P. that can be experienced by the heat pump:

$$COP_{min} = 1 + 0.075(COP_T - 1)$$

COP_T is the manufacturer's specified C.O.P. at the average annual temperature for the space in which the heat pump is located (e.g. T = 25°C).

COP_{max}	is the maximum possible C.O.P. for the system. This value is a reduced C.O.P. accounting for DHW tank stand-by losses (about 20 W or 175 kWh/yr for an energy conserving tank - RSI 2.0 insulation):

$$COP_{MAX} = \frac{\text{DHW Load(kWh)}}{175\text{kWh} + \frac{\text{DHW Load} - 175\text{kWh}}{COP_T}}$$

SHL	is the conventional annual (Oct.- Apr.) space heating load.
TSG	is the annual transmitted solar gains.

Example:

DHW Load = 5000 kWh/yr

COP_{25^oC} = 3.75 (from manufacturer's data)

$$COP_{MAX} = \frac{5000\text{ kWh}}{175\text{kWh} + \frac{(5000 - 175)\text{kWh}}{3.75}} = 3.4$$

$$COP_{min} = 1 + 0.075(3.75 - 1) = 1.2$$

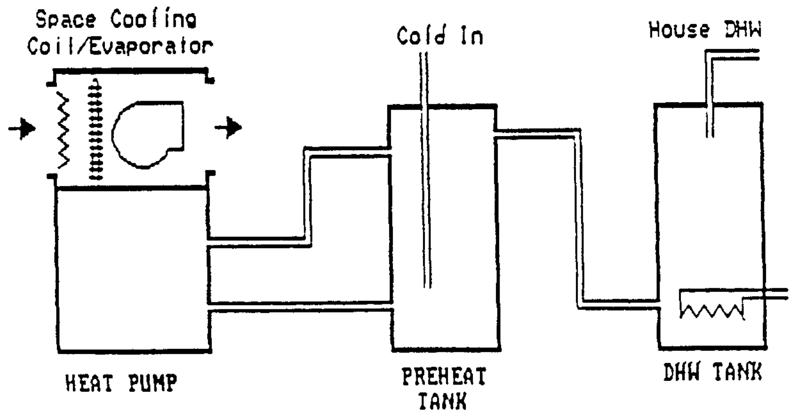
$$COP = 1.2 + (3.4 - 1.2) \times e^{-SHL/TSG} = 1.2 + 2.2\ e^{-SHL/TSG}$$

For an installation in an attached sunspace the energy savings are 61.5% x DHW load (eg. 3075 kWh) at an effective COP_T at T = 16°C.

COST

The additional equipment consists of the heat pump, preheat tank and additional plumbing materials and labour:

Estimated Installed Costs:	Heat pump	$1400
	Preheat tank	350
		$1750



INDOOR AIR HEAT PUMP/DHW SUPPLY

2. INTEGRATED MECHANICAL SYSTEM/ICE STORAGE

A fully-integrated mechanical system supplies space and DHW heating, space cooling and ventilation while extracting gains from exhaust air, grey water and via space cooling and stores them in a 450 L water tank. The evaporator, in turn, extracts energy from the storage tank via a heat exchanger and tries to freeze the water in the storage tank. The ice tank is designed as a diurnal phase change storage and has storage capacity of 40 kWh when compared to the maximum ice condition. The extracted energy combined with compressor input energy is transferred to the condenser through which water from the hot water tank is circulated and heated. Space heating is supplied from the hot water tank via a hydronic coil for forced-air heating or directly via hydronic radiators. If the fresh air is drawn into the house passively in the exhaust-only configuration, the system can also take advantage of fresh air preheat via a massive sunspace.

SIZING/INSTALLATION

The heat pump is a one-package heating/cooling/heat recovery system designed to capture waste heat from typical residential activities. The heat pump output, at a nominal 6 kW, is sufficient to extract gains delivered to the ice tank. Without reducing the energy savings, houses with larger heating loads can be supplied with additional back-up heat or by increasing the available gains, e.g. by the use of ground storage (see System 4).

The ice storage was optimized at 450 L and is configured as a 0.75 diameter tank 1.2 m high. The heat exchanger consists of 40 m of copper tubes resulting in a total extraction rate of 250 W/$^{\circ}$C.

ENERGY PERFORMANCE

The heat pump operates at an average ice tank temperature of 0°C, which defines the C.O.P. However, the amount of energy it supplies toward total heating varies. This system takes its gains from grey water, ventilation air and excess solar gains and, as such, there is an intrinsic upper limit which the heat pump is capable of supplying, i.e. the maximum gains of 11100 kWh. The relationship for calculating the heat pump contribution as a function of the conventional heating load is:

$$\text{HP Heating (kWh)} = 11100 \times \frac{COP}{COP - 1} \times (1 - e^{-THL/14000})$$

where THL is the total heating load (Space + DHW) - in kWh

The Energy Credit is further calculated as follows:

$$\text{Energy Credit} = \frac{COP - 1}{COP} \times \text{HP Heating}$$

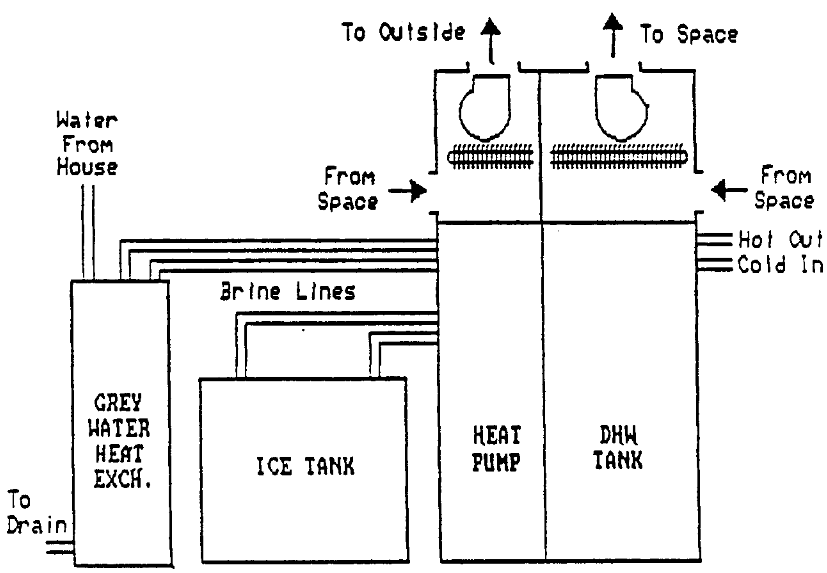
For a house with a ventilation preheat strategy , e.g. via a sunspace, the THL is the load the heat pump needs to supply, ie. after preheating is credited:

For example, for a COP = 2.2,

Total Load =	17920 kWh
Preheat Credit =	2440 kWh
HP System Load =	15480 kWh

Therefore, HP Heating = $20350\ (1 - e^{-15480/14000}) = 13600$ kWh

and, HP Energy Credit = 1.2/2.2 x 13600 = 7420 kWh
Total Energy Credit = 7420 kWh + 2440 kWh = 9860 kWh
Heating Energy Consumption = 17920 - 9860 = 8060 kWh



INTEGRATED MECHANICAL SYSTEM/ICE PHASE-CHANGE STORAGE

3. INTEGRATED MECHANICAL SYSTEM/GROUND PHASE-CHANGE STORAGE

The basic difference between this system and System 3 above is that the evaporator side storage is replaced by a seasonal ground storage. The moisture in the ground may freeze and store additional energy when the ground temperature drops to 0°C. Similiar in concept to the ice tank, the ground storage is a large "tank" of soil/water mixture (44% water by volume) or a man-made aquifer. Its walls are in contact with the ground, except for the top which is assumed to be highly insulated. Cold water from the ground storage tank (usually ranging from 0°C to 10°C) is circulated to pick up gains from exhaust air, space cooling and grey water. The brine (i.e. water/ethylene glycol solution), which circulates via the heat exchanger and the evaporator, extracts energy from the ground storage. As the "tank" temperature drops or rises, energy is either extracted or added to the surrounding soil. The energy extracted by the evaporator combined with the compressor input energy is used to supply DHW and space heating via a hot water tank as in System 3.

SIZING/INSTALLATION

For purposes of sizing the storage volume and the heat exchange (UA) between the soil and the evaporator, maximum use of the ground store is made when all of the water is frozen, typically in February. This indicates that all of the phase-change storage is depleted and any further extraction occurs as sensible heat, thus quickly dropping the storage temperature and deteriorating the heat pump performance. Using this criteria the volume and the UA coefficient, which corresponds to heat exchanger pipe length within the ground storage, were made as small as possible without a significant increase in heating energy consumption. Further economy can be realized by sizing heat pump capacity at less than maximum heating demand. Using a heat pump satisfying 75% of the design space heating demand typically satisfies more than 95% of the annual total heating load.

From simulations for Winnipeg and Vancouver it was found that sizing is dependent not only on the heating load but also on the distribution of the degree days during the heating season. A climate modifier is required and has the following form:

$$F = \frac{\text{Maximum Monthly Degree-Days}}{\text{Total Heating Degree Days}}$$

The different climates have the following F values:

Location	Oct.-Apr. CDD	Max. Mnthly CDD	F
Ottawa	4420	973	.22
Winnipeg	5400	1243	.23
Vancouver	2770	541	.195

For a 44% water/soil mixture by volume the following relationship can be used for sizing:

$$\text{Volume } (m^3) = -13 + F \times 0.005 \times \text{Total Heating Load (kWh)} > 0.5$$

Note that for total heating loads below about 12500 kWh (incl. 5275 kWh DHW load), ground phase-change storage is likely unecomical and System 3 may be more appropriate.

The UA coupling factor (between the soil and the brine circulating via the evaporator) is related to the maximum instantaneous extraction rate. The heat pump imposes a soil/brine ΔT of about $7^{\circ}C$. The required UA can be calculated as follows:

$$UA\ (W/^{\circ}C) = \frac{\text{Extraction Rate (W)}}{\Delta T\ (^{\circ}C)}$$

$$= \frac{[(COP - 1)/COP] \times \text{Heating Capacity (W)}}{7^{\circ}C}$$

where Heating Capacity is sized for 75% of the peak space heating requirement and COP is the coefficient of performance, both assumed to be at a seasonal average entering evaporator fluid temperature of $-5^{\circ}C$.

If the heat pump COP at $-5^{\circ}C$ brine temperature is 2.25:

$$UA\ (W/^{\circ}C) = \frac{[1.25/2.25] \times \text{Heating Capacity (W)}}{7^{\circ}C}$$

$$= 0.08 \times \text{Heating Capacity (W)}$$

An assumed ground storage configuration consists of a liner filled with saturated soil. It has a vertical dimension of 1.5 m and the top of the storage is located 1 m below the ground surface. The top of the storage is highly insulated and the insulation extends 1.5 m beyond the storage perimeter. The heat exchanger is assumed to be a multi-tube copper array with a heat transfer coefficient of 6 W/m of tube length. Alternate configurations, such as pool solar collectors, could be utilized as in-ground heat exchangers.

ENERGY PERFORMANCE

This system is sized to supply 95% or more of the total heating load and thus incurs a small amount of backup. Once the total heating load for a conventional heating system has been obtained, the energy credit calculation is as follows:

$$\text{Energy Credit} = \frac{COP - 1}{COP} \times THL$$

where THL is the total heating load (space + DHW) in kWh

COP is the net C.O.P. for the heat pump at the average annual entering glycol temperature (eg. $-5^{\circ}C$). The net C.O.P. is about 95% of $COP_{-5^{\circ}C}$ from manufacturer's data.

COSTS

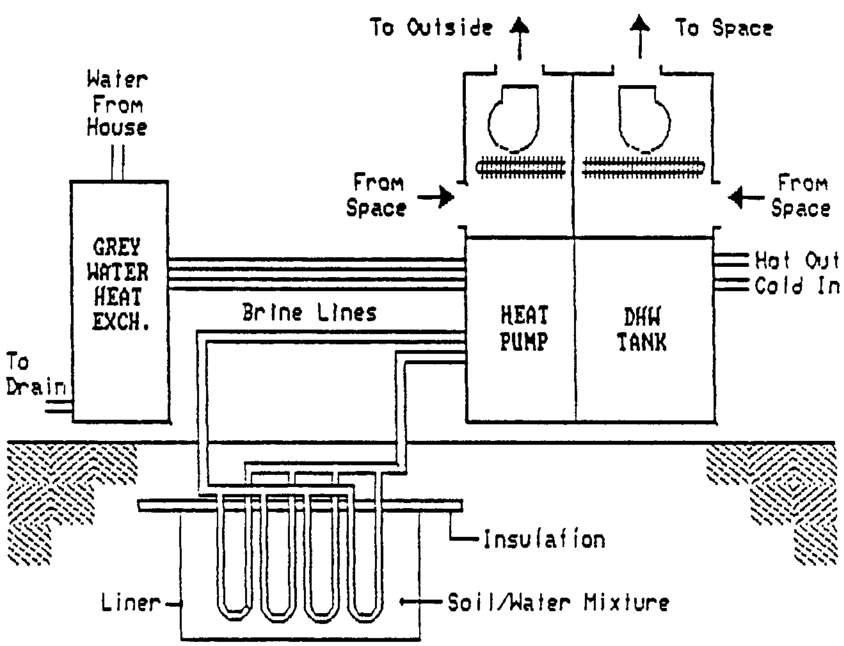
The cost of the ground storage is estimated as follows:

Heat Exchanger Tubing	\$ 3.50/m of pipe (includes manifold)
Insulation (\$3/RSI-m^2)	\$ 10/m^2 of area or \$6.67/m^3 of storage
Liner	\$ 50 + 1.25/m^3 of vol.
Excavation and Backfill	\$400
Installation Labour	\$ 50

Therefore, Total Cost = (\$7.92 x Volume) + (\$3.50 x Length) + \$500

The storage costs would typically range from \$750 to \$1500 depending on volume and pipe length.

The heat pump would typically cost \$5000 to \$6000. This brings the average total cost (heat pump + storage) to \$8500 compared to a conventional system cost of \$5700.



INTEGRATED MECHANICAL SYSTEM/GROUND STORAGE
WITH PHASE-CHANGE

APPENDIX B

GROUND STORAGE MODELLING WITH ENERPASS

Ground Storage Modelling with ENERPASS

The model consists of a volume of water/soil mixture (44% water by volume), which fixes the thermal capacitance of the storage, and a thermal coupling factor (UA) via a heat exchanger buried in the storage. The ground storage has no other thermal link to the house and losses to the surface are minimized by insulating the top of the storage, which is assumed to be 0.5 m below grade (see schematic). In ENERPASS this is achieved by using a separate basement zone, setting all inter-zone coupling factors (conduction, air exchange) to 0 and insulating the top 0.5 m of the "basement" walls with a high R-value. The rest of the "basement" is in direct contact with the ground and, Q_B, the thermal losses and gains from the surrounding earth, which is assumed not affected by the storage behavior, is calculated according to the standard Mitalas basement model.

The thermal capacitance of the storage is different above and below freezing. If p is the volumetric fraction of water in the storage, the thermal capacitance, in $kJ/m^3{}^oC$, is:

For storage temperature $\geq 0^oC$,

$$CAP = p \times 4190 \ kJ/m^3{}^oC + (1 - p) \times \rho(Cp)_g$$

For storage temperature $< 0^oC$,

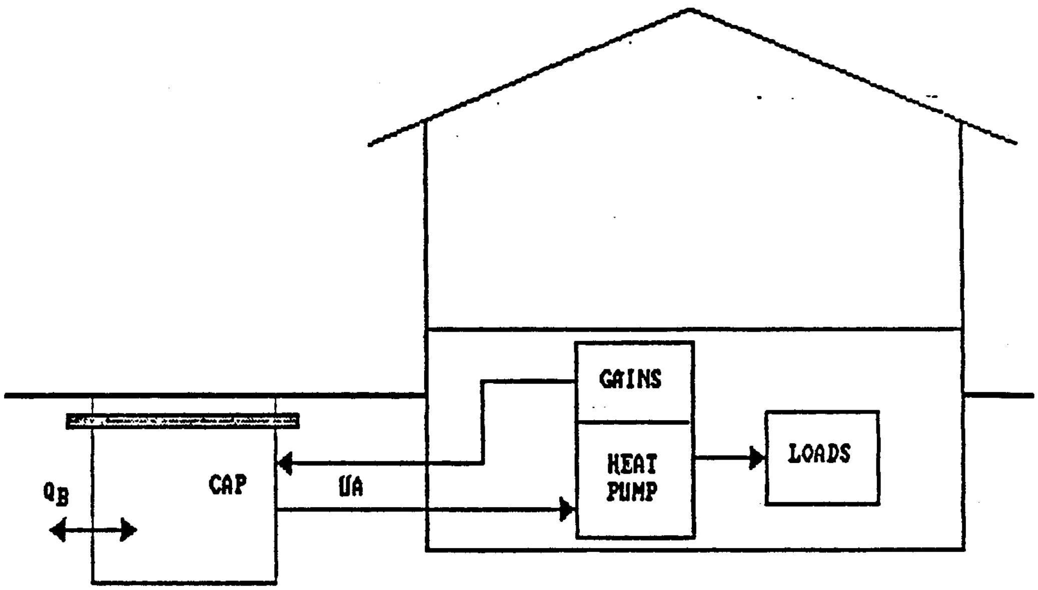
$$CAP = p \times 1990 \ kJ/m^3{}^oC + (1 - p) \times \rho(Cp)_g$$

where $\rho(Cp)_g$ is 1824 $kJ/m^3{}^oC$ for rock
1285 $kJ/m^3{}^oC$ for clay
1212 $kJ/m^3{}^oC$ for sand

The value for sand was used in this study. Upon freezing the phase change from water to ice liberates p x 334 MJ/m^3.

The UA heat exchange factor models uniformly distributed piping in the storage extracting heat for delivery to the evaporator of the heat pump. The UA coupling factor includes effects of the actual heat exchanger surface, pipe material and heat exchange fluid flow rate. When the heat exchanger chills the surrounding water to below freezing, ice forms in increasing diameters on the pipe surface. The UA changes due to the thermal conductivity of the ice thickness and the circumference. Approximately, the improvement in heat transfer due to the increase in circumference (ie. surface area) is offset by the reduction of heat transfer by the poorer conductivity as the ice continues to grow, thus allowing for a constant value to be used.

In actual operation the heat exchanger would both freeze and melt at the pipe surface. This becomes a very complex dynamic problem of tracking ice histories. Instead, in this model the heat exchanger is used only to extract heat while the gains are delivered via a water loop to the free water in the storage. It is assumed that the ground storage is controlled such that it is never fully frozen. If the storage becomes fully frozen it is of no interest as the thermal capacity for further storage is low and the temperature drops quickly as the heat extraction is all sensible energy.



GROUND STORAGE SCHEMATIC

APPENDIX C:

REVERSIBLE AIRFLOW ROCK STORE FACTSHEET

HYBRID PASSIVE SOLAR/ROCK STORAGE SYSTEM FACTSHEET

Description:

Buildings with large passive solar direct gain require sufficient thermal storage capacity and coupling to hold the temperature rise to within comfortable limits. The quantity, distribution, and location of mass in the building construction may impose a "heavy" constraint on the building architecture. In addition, temperature stratification will often require mechanical ventilation to circulate heated air to areas not receiving direct sunlight. In such cases, a storage bin of rocks through which room air is passed can be employed to effectively control temperature swing and increase the utilization of solar gain.

This fact sheet provides information on the design of a rock storage system for passive solar houses developed by ALLEN-DRERUP-WHITE LTD. Diagram 1 is a schematic of the system configuration. The furnace fan, furnace and reversing damper are controlled by a two stage heating, one stage cooling thermostat generally located near a return air intake on an upper level area open to the solar gain area. On first stage cooling, the furnace fan is operated and the damper is in cooling position allowing warm air from the high return to be circulated to the top of the rock bin, cooled through the rocks, and returned to distribution. On first stage heating, the damper is positioned in heating mode and air is circulated through the rocks from bottom to top and distributed. Second stage heating causes the furnace to come on. The reversable flow strategy enables optimum heat storage and recovery.

Advantages:

Although the system is expensive (approximately $2,500), it is a most effective means of achieving a high degree of comfort control and will yield and improved passive solar contribution to space heating. Other techniques of achieving higher mass are also expensive and less effective. Fewer constraints are imposed on the building architecture, permitting larger south facing window areas, more open flow between floors, and no specific mass elements. Since the heat is stored in an insulated container, temperature may be more easily regulated such as with night setback thermostats. The system may also be used to store heat from a wood stove thereby making this auxiliary heating option much more convenient.

Components:

The three components of the system which are additional to a conventional forced air heating system are the rock storage bin, the reversing damper, and the heat/cool thermostat. Diagram 2 illustrates a specific construction method for a top and bottom manifolded rock storage bin. The preferred rock used is screened and washed pebbles varying between 3/4" and 1-1/2" in diameter. Crushed rock and a larger diameter may be used depending on availability. The reversing damper can be fabricated by a local sheet metal shop as per diagram 3. Sizes may require changing depending on flow capacity. The damper motor should be heavy duty. The thermostat is a standard heating/cooling type and must be wired in accordance with the furnace and damper motor requirements.

Sizing Rock Storage:

The volume of rock storage may be determined using the following calculation procedure for temperature swing due to solar gain. By selecting the outdoor temperature and time of year considered critical, the calculation will predict temperature rise and storage volume may be modified to achieve an acceptable limit.

1) Calculate HLF, the building heat loss factor (w/°C).

2) Calculate mCp, the thermal capitance of inteior mass including direct gain area, rock storage and areas with adequate forced convection coupling (w·h/°C).
NOTE: include only 50mm of thick masonry.

3) Calculate Q_s, the average rate of solar gain over the gain period accounting for transmissivity of glazing and absorptivity of interior (w).

4) Select ΔT_i, the initial interior to exterior temperature difference (°C).

5) Select t, the effective period of solar gain (h).

6) Calculate $t/t^* = \frac{t \cdot HLF}{mCp}$

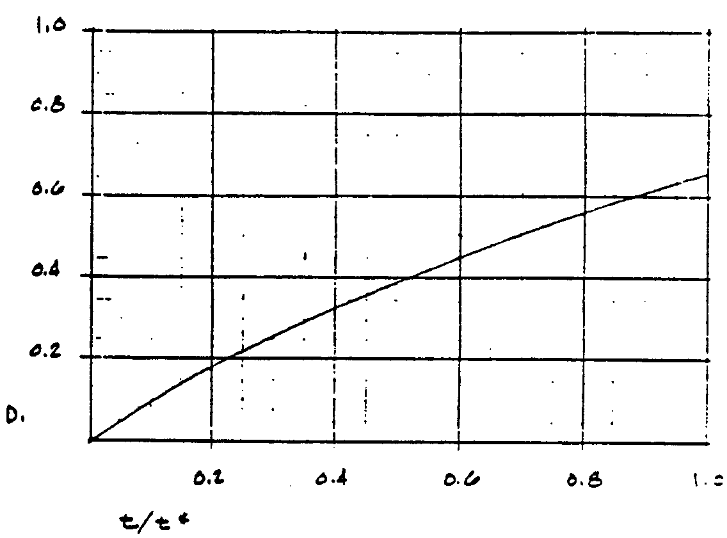
7) Determine D from chart and calculate the temperature swing,

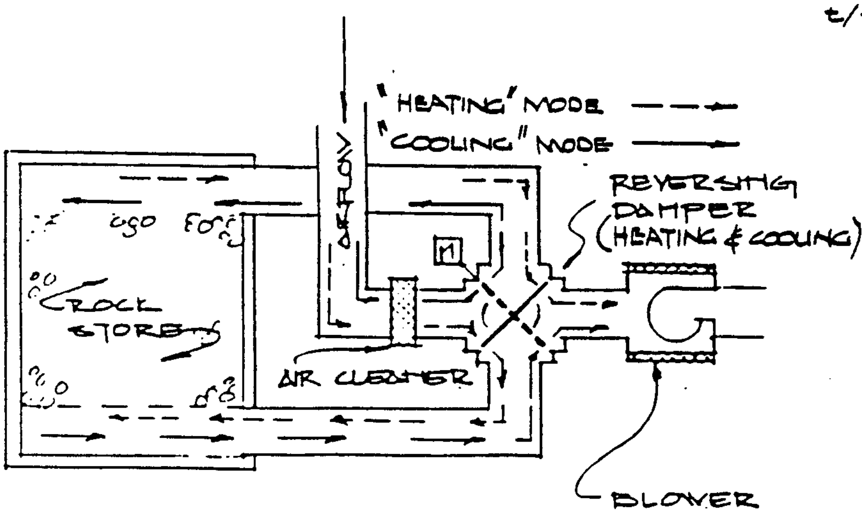
$$\Delta T_s = D\left(\frac{Q_s}{HLF} - \Delta T_i\right)$$

If $t/t^* < .10$

$$\Delta T_s = \frac{t}{mCp}\left(Q_s - \Delta T_i \cdot HLF\right)$$

DURATION FACTOR CHART.





DIAGRAM, 1.

Sample Calculation

1) HLF = 172 w/°C
2) mCp = 9,000 w·h/°C without rocks
= 13,000 w·h/°C with rocks
3) Q_s = 13,000 w
4) ΔT_i = 10°C
5) t - 5 hrs.
6) t/t* = .098 without rocks
= .066 with rocks
7) ΔT_s = 5.75°C without rocks
= 4.3°C with rocks