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MODELLING INTEGRATED MECHANICAL SYSTEMS IN RESIDENTIAL HOUSING

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ABSTRACT

We propose an abstract model for estimating the annual energy consumption of integrated mechanical systems (IMS). The model was designed specifically for compatibility with the Canadian Standards Agency's (CSA) new IMS rating standard, and is easily calibrated using the data collected in this test protocol. Unlike previous, high-resolution IMS models, this model provides a generic representation of IMS technology, allowing application to a wide range of system configurations. This paper reviews relevant parts of the CSA test standard, and describes the model and its implementation in the ESP-r simulation environment.

INTRODUCTION

Most Canadian houses have three mechanical service requirements: space-heating, domestic hot water (DHW) heating, and ventilation. Traditionally, these services were provided separately by built-for-purpose products such as furnaces, water-heaters, and heat-recovery ventilators (HRVs).

Today, manufacturers recognize the advantages in merging these systems. By combining space- and water-heating, and ventilation functions into a single, *integrated mechanical system* (IMS), manufacturers can deliver products that perform better, take up less space and are easier to install. And when designed to perform these three functions in concert, an IMS system can achieve higher efficiency, thereby saving money and reducing greenhouse-gas (GHG) emissions. (Home Builder Magazine, 2003)

In 1999, the Canadian federal government joined with with HVAC equipment manufacturers, utilities, and industry associations to foster the development of IMS technology. Called ēKOCOMFORT, this consortium undertook the research, product development, and laboratory- field-testing necessary to make IMS products a reality.

Recognizing the need for a standard metric for comparing IMS products emerging on the marketplace, the Canadian Standards Association (CSA) led development of a test standard for residential IMS systems. Called P.10, this standard describes the apparatus, instrumentation and test regimes necessary to characterize the performance of IMS products, and establishes the minimum level of performance required for certification as a P.10-compliant product. The P.10 standard also prescribes a standard performance report (called Annex C) for reporting results. (CSA, 2007)

While P.10-compliant IMS products are available on the market today, there is only limited support for this technology in building simulation software. To further encourage uptake of IMS devices and support their inclusion in energy efficiency incentive programs, a model capable of simulating an IMS unit's response to building loads is needed. In particular, this model must be compatible with the Annex C performance report prescribed in the P.10 standard.

We propose such a model. This paper provides an overview of IMS technology and reviews relevant sections of the P.10 test standard. It also describes the model and its implementation in the ESP-r building simulation program.

IMS TECHNOLOGY

An IMS product is any device that consumes gas or propane and provides space heating, domestic hot water, and ventilation with heat recovery. This definition is intentionally broad; the ēKOCOMFORT consortium wished to allow manufacturers as much flexibility as possible to design a highly efficient systems, and CSA (2007, Annex D) specifically notes a wide range of IMS configurations are expected in the future.

Thus, IMS systems take many different forms. An IMS unit may combine all three functions into a single "box" similar in size to a modern, high-efficiency furnace. Or the IMS system may comprise more familiar air handler, water heater, and HRV products, provided a single agency assumes responsibility for specifying all three products and prescribes a configuration that maximizes system efficiency.

Figure 1 depicts one plausible configuration of an IMS device. In this system, an air handler circulates air through a heating coil, and then distributes it to the space. Two additional fans move fresh and exhaust air through a heat-recovery core to provide ventilation. A low-mass boiler or tankless water heater provides hot water to the heating coil for space-heating use, and to a heat exchanger where heat is transferred to potable water for DHW use. Finally, a tempering valve protects occupants from scalding.

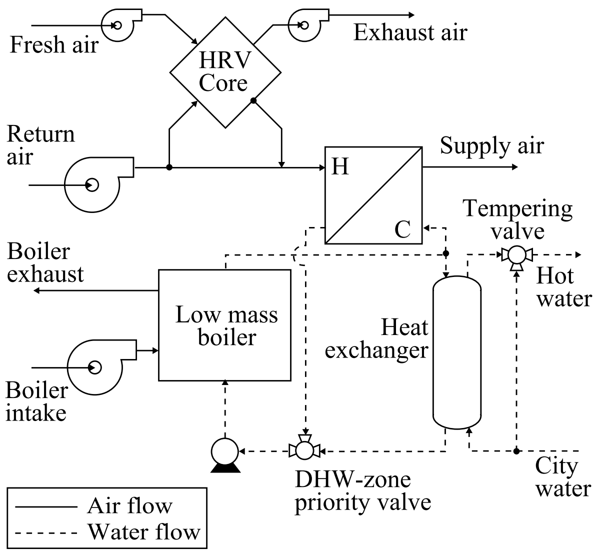

Figure 1: Example IMS configuration

IMS products typically provide continuous ventilation during the spring, fall and winter, necessitating operation of the circulation and HRV fans. In systems with hot water storage, hot water is drawn to meet space- and water-heating loads, and the burner cycles on and off to maintain the temperature in the water tank. Systems based on boilers or tankless water heaters activate the burner in direct response to heating loads.

P.10 STANDARD

CSA (2007) developed standard P.10 to provide a consistent test protocol and rating methodology for IMS products. To allow a wide variety of system designs, P.10 places no restrictions on the configuration of an IMS unit. Compliance with the standard is determined entirely by the unit's performance.

The standard prescribes three separate test regimes to examine an IMS product's performance in each function: i) *space heating performance* tests ii) *water-heating performance* tests, and iii) *ventilation and heat recovery performance* tests. In addition to these, P.10 also prescribes an *integtated function test* which exercises all three functions simultaneously.

Space heating performance tests

The P.10 space heating performance tests characterize an IMS unit's full- and part-load efficiency during space-heating operation. In these tests, the IMS unit's fuel and electrical consumption and heat delivery are determined at 100%, 40% and 15% of full heating capacity. The P.10 standard defines a net-efficiency ($\eta_{SH\text{-}net}$) at each of these part-load ratios as follows:

$$\eta_{SH\text{-}net} = \frac{\dot{Q}_{SH} - P_{total}}{\dot{m}_{fuel} \cdot HHV_{fuel}} \tag{1}$$

where $\dot{Q}_{SH}$ is the average rate of heat delivery to the house, and P_{total} is the IMS unit's average rate of electricity consumption. Symbol $\dot{m}_{fuel}$ describes the IMS unit's average rate of fuel consumption, while HHV_{fuel} is the higher heating value of the fuel.

The P.10 standard also defines a composite space heating efficiency (η_{CSH}) which aggregates the performance at the three part-load ratios:

$$\eta_{CSH} = 0.1\eta_{SH\text{-}net,100\%} + 0.6\eta_{SH\text{-}net,40\%} + 0.3\eta_{SH\text{-}net,15\%} \tag{2}$$

By using these weighting factors, CSA recognized that residential heating systems spend a small portion of each year operating at full load, and operate at some fraction of their capacity for the remainder of the heating season.

Water heating performance tests

The P.10 water heating performance tests characterize an IMS unit's efficiency using a procedure similar to the CSA (2004) standard for gas-fired water heaters (P.3-04). The IMS unit is subjected to a simulated-usage-test (SUT), in which six 50 l hot water draws are placed on the unit over a six-hour period, followed by an 18-hour period in which no loads are placed on the unit. The P.10 standard describes the IMS unit's performance over this period using the *water heater performance factor* (*WHPF*):

$$WHPF = \frac{\dot{Q}_{DHW}}{\dot{m}_{fuel,SUT} \cdot HHV_{fuel} + P_{DHW}} \tag{3}$$

where $\dot{Q}_{DHW}$ is the average rate of heat delivered to the hot water, $\dot{m}_{fuel,SUT}$ is the average rate of fuel consumption in the IMS unit during the SUT, and P_{DHW} is the average rate of power consumption. Since the temperature and volume of the domestic hot water draws used in the SUT are prescribed by the P.10 standard, $\dot{Q}_{DHW}$ is a constant, known value.[1]

Similarities between the CSA P.3 and P.10 standards allow meaningful comparison between the water-heater-performance-factor metric and the more familiar energy-factor (EF) metric, which is commonly used to describe conventional gas-fired water heaters.

[1] Actually, the P.10 standard contains several intermediate calculations that i) equivalence computed *WHPF* values for systems with- and without hot water storage; and ii) account for periods in the test when the hot-water make-up and delivery temperatures can not be maintained at the values prescribed in the standard. Though simplified here for brevity, Equation 3 still faithfully describes the functional form of the P.10 *WHPF* definition.

Ventilation performance tests

The ventilation and heat recovery performance tests characterize the efficiency of the IMS unit's HRV component when providing fresh air. These tests are syntactically similar to the CSA (2000) standard for testing HRV equipment (C439).

The ventilation performance tests impose constant return air temperature (22 °C) and outdoor air temperature (0 °C) conditions on the IMS unit. The performance of the IMS unit in these tests is aggregated in the sensible energy recovery efficiency (η_{SR}):

$$\eta_{SR} = \frac{\dot{m}_{FA} c_p \left(T_{SA} - T_{FA}\right) - \dot{Q}_{SF} - \dot{Q}_{SH} - \dot{Q}_{C} - \dot{Q}_{Def}}{\dot{m}_{\max} C_p \left(T_{RA} - T_{FA}\right) + \dot{Q}_{EF} + \dot{Q}_{EH}} \tag{4}$$

where $\dot{m}_{FA}$ is the mass flow rate of the fresh air, and $\dot{m}_{\max}$ is the larger of the fresh and exhaust air flow rates. Symbol c_p describes the specific heat of air. T_{SA} describes the supply air temperature, T_{FA} the fresh (outdoor) air temperature, and T_{RA} the return (indoor) air temperature.

Equation 4 contains a number of other terms accounting for other energy sources and heat losses in the HRV core. Symbols $\dot{Q}_{SF}$ and $\dot{Q}_{EF}$ describe the rates of energy transfer to the supply- and exhaust-air streams from the HRV fans. Similarly, $\dot{Q}_{SH}$ and $\dot{Q}_{EH}$ describe the rate of energy transfer into the supply- and exhaust-air streams from electric heaters in the HRV core (if any such heaters are installed in the IMS). Parameter $\dot{Q}_{C}$ describes the heat transfer through the IMS unit's case, and $\dot{Q}_{Def}$ describes the energy supplied to the HRV unit during its defrost cycle.

The P.10 standard describes computation of these additional energy terms ($\dot{Q}_{SF}$, $\dot{Q}_{EF}$, $\dot{Q}_{SH}$, $\dot{Q}_{EH}$, $\dot{Q}_{C}$, and $\dot{Q}_{Def}$) in detail. These terms account for heat transfer to the fresh-air stream from sources other than the exhaust air stream. By including these terms, the sensible heat recovery efficiency describes the the sensible heat transfer to the fresh air stream as a fraction of the total amount of sensible heat available in the exhaust stream.

Overall performance

To permit easy comparison of IMS products, the P.10 standard provides two metrics that aggregate performance in all three functions. The *overall thermal performance factor* (*OTPF*) describes the ratio between the sum of the IMS unit's thermal output and the thermal energy input (that is, the energy content of the fuel) needed to produce these outputs:

$$OTPF = \frac{2000\dot{Q}_{SH,\max} + 8150}{\frac{2000\dot{Q}_{SH,\max}}{\eta_{CSH}} + \frac{4400}{WHPF} + \frac{3750(1-\eta_{SRE})}{\eta_{CSH}}} \tag{5}$$

where $\dot{Q}_{SH,\max}$ is the IMS unit's full-load space-heating output.

The IMS unit's annual electricity consumption (E_{annual}) is estimated as follows:

$$E_{annual} = \left(200E_{SH,100\%} + 3000E_{SH,40\%} + E_{SH,15\%}\right) + 365E_{DHW,SUT} + 1560E_{cont} \tag{6}$$

where $E_{SH,100\%}$ is the IMS unit's average hourly electricity consumption when operating at full load, $E_{SH,100\%}$ is the consumption at 40% capacity, and $E_{SH,15\%}$ is the consumption at 15% capacity. Symbol $E_{DHW,SUT}$ describes the average hourly consumption observed over the 24 hour water-heating SUT test, and $E_{standby}$ describes the average hourly electricity consumption of the unit in standby mode (that is, when it does not provide any heating or ventilation).

The numerical constants in Equations 5 and 6 reflect assumptions about the fractions of the year an IMS unit spends in various modes. CSA (2007, Annex D) describes these assumptions in detail.

An IMS unit achieving a minimum overall thermal-performance factor of 0.78 qualifies for certification as a P.10-compliant product. Units achieving overall thermal-performance factors of at least 0.85 qualify for a *premium performance* designation.

Integrated function test

In addition to the separate function tests, the P.10 test protocol includes an 24-hour integrated test that exercises all three functions simultaneously. CSA (2007) intends this test to verify that an IMS unit can meet coincident ventilation-, space- and water-heating loads, and that the unit's controls perform as designed.

This test operates the IMS unit to meet a constant space-heating load equivalent to 71% of its space-heating capacity. In addition, the test imposes the same 300 l domestic hot water load used in the water-heating SUT. Finally, the unit must supply continuous ventilation using fresh air supplied at −25 °C.

Annex C Report

The P.10 standard prescribes an information sheet for reporting the rated performance of IMS units. Called *Annex C*, this sheet provides:

- The overall-thermal performance factor
- The water-heating performance factor
- The net space-heating efficiency at 15%, 40% and 100% capacity
- The sensible heat recovery efficiency
- The circulation fan power at full and part-load, the controls standby electrical consumption, and the power used by ventilation fans

Annex C also reports the gas and electricity consumption observed during the integrated function test.

MODEL DEVELOPMENT

Parent et al. (2001) and Beausoleil-Morrison and Haddad (2003) previously described a dynamic, lumped-parameter models for IMS technology. Using the principles proposed by Bourdouxhe et al. (1998), Parent et al. applied the log-mean-temperature-difference method to characterize heat transfer in the IMS unit's space heating coil and heat recovery ventilator. Their model specifically considers IMS units with integrated water storage, and uses a first-order differential equation to describe dynamic characteristics of the storage tank.

Beausoleil-Morrison and Haddad (2003) refined the storage tank component of this model, and developed a modular implementation that can be coupled to various other models for air-handlers and heating coils. Their work did not include integration of mechanical ventilation systems with heat-recovery.

These lumped-parameter IMS models characterize mass flow and heat transfer between components within the IMS system, such as the circulation fan, heating coil, and combustion flue. This level of resolution allows researchers to determine the effect of varying one parameter (for example, the flow rate between the space-heating coil and the water storage tank) on system performance.

But the lumped-parameter models are not well-suited for estimating the annual fuel consumption and GHG-emissions from P.10-compliant products. They assume the IMS system incorporates a gas-fired, water-storage tank, but CSA (2007) recognizes that P.10-compliant products may use other heat sources, such as tankless water heaters. And the pipe, duct, and control configurations assumed by the lumped-parameter models are not sufficiently generic to represent the different arrangements used by manufacturers.

Finally, the lumped-parameter models require several inputs (for example, the excess air ratio and the flue heat transfer coefficient) that are not determined within the scope of P.10 testing, and are thus not published on the Annex C report. Without these parameters, the lumped-parameter models cannot be readily calibrated to represent P.10-compliant products.

Modelling philosophy

While the lumped-parameter models discretize IMS devices with sufficient detail to discern heat transfer between subcomponents within the unit, we propose abstract, *idealized* representation of IMS technology. Rather than describing the system with mechanistic relationships that apply to only one configuration, the idealized model relies heavily on the performance metrics collected during P.10 testing, allowing application to any P.10-compliant product.

Without detailed knowledge of the system's configuration and control schemes, the idealized IMS model cannot characterize the short-term interactions of the IMS unit's subsystems in response to changing loads. For instance, one IMS system might cycle a burner on and off to meet reduced thermal loads, while another system could use a variable-firing-rate burner. The idealized model does not distinguish between these control strategies; instead, it assumes the performance metrics collected in testing account for short-term, part-load behaviour. And this approach is consistent with the P.10 specification—the standard requires part-load space heating efficiencies be computed by running the IMS unit under constant load for at least an hour. The idealized model's predictions should agree well with measurements at an hourly time-step, but we expect the agreement will deteriorate at shorter timescales.

The net- and composite-space-heating efficiency ($\eta_{SH\text{–}net}$ and η_{CSH}), the water-heater performance-factor (*WHPF*) and the overall thermal performance factor (*OTPF*) metrics defined in the P.10 standard blend fuel and electricity consumption measurements into a single description of energy use. While these metrics are valuable when comparing the performance of IMS products, they are less helpful in building simulation. A house's fuel costs and GHG emissions are affected by the local electric generation mix and rate structure. To meaningfully compute costs and emissions, the model must disaggregate energy use by fuel source. It must determine not only how much electricity, natural gas, oil and wood a house uses, but also the time-of-year and the time-of-day that this energy is used.

For these reasons, the idealized IMS model uses these metrics to derive familiar expressions of efficiency that relate the unit's thermal output to the heating value of the consumed fuel. In addition, the model makes two key assumptions about the design and operation of IMS technology:

- *Electronic ignition:* The model assumes electronic-ignition is used to activate the combustion module, and the IMS unit only consumes fuel when responding to space- and water-heating demands.

- *Pseudo-steady-state operation:* The model neglects the IMS unit's thermal mass. The P.10 test specification explicitly considers thermal mass when computing the performance ratings, and the model assumes the ratings adequately account for these effects.

Space-heating operation

The rate of heat delivery to the building ($\dot{Q}_{SH}$) is the sum of the rate of heat transfer at the heating coil ($\dot{Q}_{coil}$) and the work performed by the circulation fan (P_{fan}):

$$\dot{Q}_{SH} = \dot{Q}_{coil} + P_{fan} \tag{7}$$

While Annex C does not report the circulation fan power during space heating operation, it does report total power consumption (P_{SH}), as well as the power consumption of the ventilation fans (P_{vent}) and the controls in standby mode ($P_{controls}$). Using these values, the fan power can be estimated:

$$P_{fan} = P_{SH} - P_{controls} - P_{vent} \tag{8}$$

The heat transfer at the heating coil is a function of the fuel flow rate and the *effective space heating efficiency* (η_{SH}):

$$\dot{Q}_{coil} = \eta_{SH}(\dot{m}_{fuel,SH} \cdot HHV_{fuel}) \tag{9}$$

where $\dot{m}_{fuel,SH}$ is the rate of fuel consumption for space heating purposes.

The effective space efficiency parameter required by the model can be computed from the net space-heating efficiency reported in the Annex C report:

$$\eta_{SH} = \eta_{net} - \left(\frac{P_{controls} + P_{vent}}{\dot{m}_{fuel,SH} \cdot HHV_{fuel}} \right) \tag{10}$$

Water-heating operation

The water-heater performance factor is analogous to the IMS unit's efficiency; it correlates the total useful energy delivered by the unit (that is, heated water) to the total energy consumed (that is, fuel and electricity). But for use in building simulation, the energy consumption must be disaggregated by end-use.

Inefficiency associated with water heating can be attributed to two sources: i) incomplete heat transfer from the fuel to the water, and ii) standby losses. To accurately estimate water-heating performance in response to varying demand patterns, the model must separately characterize these two energy losses. This way, the model can estimate energy consumption when the water-demand pattern differs from the profile used in the CSA SUT.

The rate of heat transfer to the hot water ($\dot{Q}_{DHW}$) is:

$$\dot{Q}_{DHW} = \dot{m}_{DHW} c_p \left(T_{DHW} - T_{supply} \right) \tag{11}$$

where $\dot{m}_{DHW}$ is the mass flow rate of the domestic hot water, c_p is its specific heat capacity, T_{DHW} is the hot-water delivery temperature, and T_{supply} is the temperature of the municipal water supply.

The actual amount of fuel consumed by the IMS unit for water-heating purposes ($\dot{m}_{fuel,DHW}$) is:

$$\dot{m}_{fuel,DHW} = \frac{\dot{Q}_{DHW} + \dot{Q}_{standby}}{\eta_{DHW} \cdot HHV_{fuel}} \tag{12}$$

where $\dot{Q}_{standby}$ is the hourly-averaged rate of standby energy loss in the IMS unit from the IMS unit, and η_{DHW} describes the effective efficiency at which the IMS unit heats water.

To directly compare the water heater performance factor and the effective thermal efficiency, we must first compute the average heat transfer to the hot water during the 24-hour SUT, as well as the average energy input over the same period:

$$\dot{Q}_{DHW,avg} = \frac{M_{draw} c_p \left(T_{DHW} - T_{supply} \right)}{\Delta t_{SUT}} \tag{13}$$

where $\dot{Q}_{DHW,avg}$ is the average rate of heat transfer to the domestic hot water over the SUT, M_{draw} is the total mass of water drawn from the tank, and c_p is the specific heat of water. Symbol Δt_{SUT} is the duration of the SUT.

The average rate of heat transfer ($\dot{Q}_{DHW,avg}$) is a constant value. It is a function of the supply and delivery temperatures and the mass of water drawn—all of which are prescribed by the P.10 specification.

Now the DHW effective thermal efficiency and water-heater performance factor can be compared:

$$\eta_{DHW} = \frac{\dot{Q}_{DHW,avg} + \dot{Q}_{standby}}{\dot{m}_{fuel,avg} \cdot HHV_{fuel}} \tag{14}$$

$$WHPF = \frac{\dot{Q}_{DHW,avg}}{\dot{m}_{fuel,avg} \cdot HHV_{fuel} + P_{DHW,avg}} \tag{15}$$

Equating these equations provides the following relationship between the effective thermal efficiency and the water-heater performance factor:

$$\eta_{DHW} = \left[\frac{\dot{Q}_{DHW,avg} + \dot{Q}_{standby}}{\dot{Q}_{DHW,avg} - (WHPF \cdot P_{DHW,avg})} \right] WHPF \tag{16}$$

Two terms on the right-hand-side of Equation 16 are known: $\dot{Q}_{DHW,avg}$ is a constant value, and *WHPF* is provided from the P.10 test report. The remaining terms ($\dot{Q}_{standby}$ and $P_{DHW,avg}$) must be obtained elsewhere.

The P.10 standard requires test agencies to estimate the hourly average heat loss, but this parameter is not included in the Annex C report. Without this data, the model estimates the ratio between flue- and standby-losses using the same methodology adopted in the HOT2000 program (Natural Resouces Canada, 2007) to disaggregate heating efficiency and standby-losses from energy factor (EF) ratings for conventional water heaters. Adopting the HOT2000 methodology not only provides a pragmatic method for estimating standby-losses from water-heater performance factor ratings—it also ensures that conventional water-heaters and IMS technology are modelled consistently, and that simulation results for these systems can be compared.

To estimate the heat loss from the IMS unit, we first define the the total amount of energy used by the IMS unit that is not transferred to the hot water:

$$\dot{E}_{loss} = (P_{WH,avg} + \dot{m}_{fuel,avg} \cdot HHV_{fuel}) - \dot{Q}_{DHW,avg} \tag{17}$$

Table 1: Ratio between flue- and standby-losses. (Natural Resouces Canada, 2007)

Tank type	Fuel	Loss ratio (α)
Condensing tank	Natural gas	0.0
	Oil	0.0
Conventional tank	Natural gas	0.314
	Oil	0.468
Induced draft fan tank	Natural gas	0.490
Tank-less gas heater	Natural gas	1.0
	Oil	1.0

where $\dot{E}_{loss}$ is the total amount of energy use that is not delivered to the building. The unrecovered energy includes both flue- and skin-losses, as well as the electricity consumed by the IMS unit. Thus:

$$\dot{E}_{loss} = \dot{Q}_{standby} + \dot{Q}_{\textit{flue-loss}} + P_{DHW,avg} \tag{18}$$

where $\dot{Q}_{\textit{flue-loss}}$ is the rate at which heat is exhausted through the flue.

The ratio of flue-loss to skin-loss depends on the design and operation of the IMS unit. A dimensionless parameter, α, describes this ratio:

$$\dot{Q}_{\textit{flue-loss}} = \alpha(\dot{E}_{loss} - P_{DHW,avg}) \tag{19}$$

$$\dot{Q}_{standby} = (1-\alpha)(\dot{E}_{loss} - P_{DHW,avg}) \tag{20}$$

Values for α have been collated for different water-heating systems by Natural Resouces Canada (2007) for use in the HOT2000 program. These values are presented in Table 1.

Combining equations 15, 17, 18, and 20 provides the following expression for the standby-losses:

$$\dot{Q}_{standby} = (1-\alpha)\left[\frac{\dot{Q}_{DHW,avg}}{WHPF} - \dot{Q}_{DHW,avg} - P_{DHW,avg}\right] \tag{21}$$

The power consumption during water-heating operation ($P_{DHW,avg}$) remains unknown, and is an input to the model. But we do not expect IMS units to consume a significant amount of electricity in water-heater operation, and assuming a value of zero (that is, $P_{DHW,avg} \approx 0$) should not introduce significant error. Alternatively, users can provide a value for the power consumption if a suitable estimate is available.

Heat recovery efficiency

When quantifying the performance of the ventilation and heat recovery equipment, the model must characterize:

- the incremental thermal load on the building associated with ventilation air
- the change in humidity in the conditioned space
- the electricity consumed by the supply and exhaust fans

Exact treatment of the heat-recovery equipment requires consideration of potentially unbalanced ventilation and condensation in the supply and exhaust streams. But the idealized IMS model does not provide the necessary resolution to consider these effects and makes the following assumptions:

- *Balanced ventilation:* The model assumes the supply and exhaust volumetric air flow rates, measured at the fresh-air outlet and exhaust-air inlet, are equal. In reality, unbalanced ventilation causes additional air exchange elsewhere in the building envelope, and reduces the IMS unit's efficiency because the infiltration or ex-filtration air does not undergo heat recovery in the unit's heat exchanger.
- *Sensible heat transfer in supply air stream:* The model assumes the vapour entrained in the supply air stream does not condense when the fresh air stream is cooled in the heat recovery component. In reality, condensate may form in hot and humid weather. But in Canada, these conditions prevail for small fractions of the year, and should not significantly affect the building's moisture balance.

The mass flow rate of dry air supplied to the building ($\dot{m}_{\textit{FA-dry}}$) and the rate of dry air exhausted from the building ($\dot{m}_{\textit{EA-dry}}$) are:

$$\dot{m}_{\textit{FA-dry}} = \dot{V}_{vent}\left(\frac{R_{air}T_{FA}}{p_{FA}}\right) \tag{22}$$

$$\dot{m}_{\textit{EA-dry}} = \dot{V}_{vent}\left(\frac{R_{air}T_{RA}}{p_{RA}}\right) \tag{23}$$

where $\dot{V}_{vent}$ is the volumetric flow rate of ventilation air and R_{air} is the ideal gas constant for air. Symbols T_{RA} and T_{FA} describe the indoor and outdoor dry-bulb temperature, while symbols p_{RA} and p_{FA} describe the indoor and outdoor air partial pressure.

The mass flow rates of the vapour entrained in the fresh and exhaust air streams are:

$$\dot{m}_{\textit{FA-vap}} = \omega_{FA}\dot{m}_{\textit{FA-dry}} \tag{24}$$

$$\dot{m}_{\textit{EA-vap}} = \omega_{RA}\dot{m}_{\textit{EA-dry}} \tag{25}$$

where $\dot{m}_{\textit{FA-vap}}$ describes the mass flow rate of vapour supplied, and $\dot{m}_{\textit{EA-vap}}$ the mass flow rate of vapour exhausted. Symbols ω_{FA} and ω_{RA} describe the humidity ratios of the ambient and zone air .

Finally, the total change in the water vapour encapsulated in the building ($\Delta m_{vap,vent}$) is:

$$\Delta m_{vap,vent} = (\dot{m}_{\textit{FA-vap}} - \dot{m}_{\textit{EA-vap}})\Delta t \tag{26}$$

Heat transfer

The heat transfer between the supply and exhaust air streams in the heat recovery core is quantified by the unit's *heat recovery efficiency* (η_{HR}):

$$\eta_{HR} = \frac{\dot{Q}_{core}}{\dot{C}_{p,\max}\left(T_{RA} - T_{FA}\right) + P_{FA\text{-}fan}} \quad (27)$$

where $\dot{Q}_{core}$ is the rate of heat transfer from the exhaust stream to the supply stream in the heat-exchange core, and $\dot{C}_{p,\max}$ is the larger of the supply and exhaust stream heat capacity flow rates (that is, the product of the mass flow rate and specific heat capacity). Symbol $P_{FA\text{-}fan}$ is the electrical power consumed by the fresh air fan.

The heat recovery effectiveness varies according to the zone and ambient temperatures. Cool ambient temperatures cause condensation in the exhaust air stream, enhancing heat exchange. But if the temperature falls further, the condensate may freeze and necessitate a defrost cycle, which degrades heat recovery effectiveness.

The onset of condensation can be reasonably estimated by comparing the exhaust air outlet temperature to the saturation temperature for the given humidity ratio. But the onset of freezing is harder to determine—the outlet temperature may not indicate if ice is accumulating in the heat exchanger. Moreover, IMS units may preemptively activate their defrost cycles if the outdoor temperature drops below a predetermined threshold.

The model does not attempt to characterize the IMS unit's behaviour in condensing and defrost operation. Rather, it assumes the unit's performance is quantified by one of three heat-recovery efficiency values describing these three modes of operation. That is:

$$\eta_{HR} = \begin{cases} \eta_{DRE} & \text{if } T_{FA\text{-}in} \leq T_{def} \\ \eta_{CRE} & \text{if } T_{FA\text{-}in} > T_{def} \text{ and } T_{EA\text{-}out} \leq T_{sat} \\ \eta_{SRE} & \text{if } T_{FA\text{-}in} > T_{def} \text{ and } T_{EA\text{-}out} > T_{sat} \end{cases} \quad (28)$$

where η_{SRE} describes the unit's efficiency in sensible operation when the exhaust outlet temperature ($T_{EA\text{-}out}$) is above the saturation temperature (T_{sat}), and η_{CRE} describes the unit's average efficiency when the outlet temperature is below the saturation temperature. Symbol η_{DRE} describes the unit's average efficiency when the outdoor temperature ($T_{FA\text{-}in}$) drops falls below the predetermined threshold (T_{def}) at which point the unit activates its defrost cycle.

While the model accounts for improved performance in condensing operation and degraded performance during a defrost cycle, the P.10 standard only prescribes testing in the sensible-heat-transfer regime, and Annex C only describes the unit's sensible heat recovery efficiency (η_{SRE}). Unless additional data describing performance in condensing and defrost operation are available, the model assumes the sensible heat recovery efficiency prevails in all operating conditions (that is, $\eta_{HR} = \eta_{SRE}$).

Finally, the total incremental thermal load attributed to ventilating the building ($\dot{Q}_{load\text{-}vent}$) is:

$$\dot{Q}_{load\text{-}vent} = \left(1 - \eta_{HR}\right)\left(\dot{C}_{p,\max}\left(T_{EA\text{-}in} - T_{FA\text{-}in}\right) + P_{FA\text{-}fan}\right) \quad (29)$$

IMPLEMENTATION

Purdy and Haddad (2002) previously proposed a method for coupling an idealized furnace model to the high-resolution ESP-r simulation engine; we used the same approach for the IMS model. In this implementation, ESP-r computes the heat injection necessary to maintain the building at the desired set-point temperature, and the idealized model estimates the IMS unit's performance in response to this thermal load. ESP-r invokes the idealized model on each time-step and, if the model determines the IMS unit cannot meet the building requirements, adjusts the building energy balance equations on the subsequent time step.

The ventilation component of the IMS model is also coupled to the ESP-r simulation environment. The model computes the change in the masses of dry air and vapour encapsulated in the building, and ESP-r adjusts the building's mass balance state-equations accordingly. This coupling permits study of the effect of ventilation on latent cooling loads during the summer months.

PRELIMINARY RESULTS

One Canadian-manufactured IMS product has completed the P.10-prescribed test protocol, and qualified for the P.10-certified premium-performance designation. To gauge idealized model's ability to represent this particular product, the model was calibrated using the inputs reported on the Annex C information sheet. These values are presented in Table 2.

The model was subjected to the same boundary conditions prescribed in the P.10 integrated function test specification. Preliminary analysis of these results suggests the model accurately predicts the units's energy consumption; the predicted gas consumption differed from measurements by 1.4%, while the electrical consumption estimates differed by 4.1%.

The accuracy of idealized IMS model can be further quantified by comparing its predictions to the high-resolution data collected during P.10 testing. These activities are beyond the scope of the current project, but are planned in the future.

CONCLUSIONS

The Canadian Standards Association recently published the P.10 test standard, which defines the test protocols necessary to rate residential integrated mechanical systems. This standard will likely become the most pervasive source of publically-available performance data for IMS systems.

Table 2: Model inputs derived from Annex C report

Parameter		Units	Value
Nominal burner input		W	43950
Space heating:	Capacity	W	27900
	Net efficiency at 15% part-load-ratio	%	88
	Net efficiency at 40% part-load-ratio	%	90
	Net efficiency at 100% part-load-ratio	%	84
	Electricity consumption at 15% part-load-ratio	W	180
	Electricity consumption at 40% part-load-ratio	W	320
	Electricity consumption at 100% part-load-ratio	W	475
Water heating:	Water-heater performance factor	–	0.81
Ventilation:	Fresh air flow rate	l/s	34
	Exhaust air flow rate	l/s	34
	Fresh fan power	W	27
	Exhaust fan power	W	27
	Sensible recovery efficiency	–	0.60
Standby:	Controls power	W	13
	Circulation fan power	W	103

To support estimation of the annual energy requirements of IMS technology, we proposed an idealized representation of IMS systems. The idealized model eschews high-resolution discretization of the IMS unit in favour of a generic approach that can represent any configuration of system recognized by the CSA. Moreover, we specifically designed the model for compatibility with the P.10 Annex C report. Thus, the model is readily calibrated provided the Annex C report is available.

We calibrated the model using an Annex C report for a P.10-compliant IMS product. Preliminary analysis suggests the model accurately predicts the energy consumption of this unit; the estimated gas consumption differed from measurements by 1.4%, while the electrical consumption estimates differed by 4.1%.

Finally, we plan to undertake a comprehensive validation study of the IMS model using data collected during P.10-compliance testing of an IMS unit. These activities will be the subject of a future paper.

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