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USING RATINGS DATA TO DERIVE SIMULATION-MODEL INPUTS FOR STORAGE-TANK WATER HEATERS

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ABSTRACT

Loss coefficients and fuel-conversion efficiency are inputs to practical simulation models of conventional domestic hot-water storage tanks. Formulae based on whole-tank energy balances are presented to derive these inputs from published ratings data. It is important to distinguish condition-dependent recovery efficiency from condition-independent fuel-conversion efficiency. Uncertainty in test results produces 20%-50% uncertainty in derived values of the loss coefficient. Results satisfy the basic energy balance pertaining to the test. The model results reproduce the ratings when the standard test is simulated using inputs derived from ratings. Previous work is reviewed and shown to agree with results here, with certain caveats stemming from recovery efficiency issues.

1. INTRODUCTION

Domestic hot-water (DHW) energy use has become relatively more important as the use of energy for space heating and cooling has decreased because of envelope improvements. Energy performance, costs, and other attributes of alternative DHW technologies must be consistently determined and compared to market requirements. The U.S. DOE Buildings and Solar Programs are actively engaged in such market-based assessments for DHW and other technologies (1), motivating this work. Electric- and gas-driven storage tanks are by far the most common DHW technologies. (“Gas” can be taken here to include other fossil fuels.) In this paper, we present a data-based procedure for determining several key simulation-model inputs for conventional DHW storage tanks.

There are three levels of models for predicting storage-tank performance. On one extreme, there are two- and three-D computational fluid-dynamics models, solving the Navier-Stokes equations (in appropriately simplified form) at ~ 10^4-10^6 coupled spatial nodes. These models are unsuitable for annual simulations because of long run times. On the other extreme are static whole-tank models based upon time integration of the whole-tank energy balance, as used below in deriving input formulae. These models cannot address important dynamic questions, such as runout or load control. In between these extremes are one-dimensional, finite-difference, dynamic models based upon solving coupled-mass and energy-balance equations (2). These simulation models are appropriate for annual performance, can address dynamic questions, and are the models we refer to in this paper.

Inputs for storage-tank models are problematic. Tank insulation material and thickness are not reported in rating data (3,4). Although inputs might appear straightforward for the conduction problem through insulated tank surfaces, insulation properties may not be well-known, insulation homogeneity is generally unknown (voids or gaps may be present), and tank surfaces are not entirely insulated. In addition, tanks usually have copper pipes attached, which function as “thermal shorts” that are difficult to model. For gas tanks, a central flue creates a complex natural convection loop, and the corresponding coefficient(s) are difficult to estimate. Similar comments apply *a fortiori* to combustion efficiency. For confidence and credibility, these inputs should be based upon measurement.

U.S. water heaters are tested as in (3), providing measured performance metrics under a specific set of conditions. Test conditions are intended to be representative of average-use conditions. The ratings are published in (4) for most U.S. tank models, allowing simple, credible product intercomparison. There is concern about the accuracy of the test, estimated at ± .025 on the energy factor in (5). Performance under other conditions must be inferred externally via a model. In this paper, we derive algorithms to infer several key inputs for storage-tank models using published rating data.

First, we describe simulation models for storage water heaters, focusing on key inputs and their meanings. Next, we describe the standard test, focusing on test outputs and their meaning. Formulae for deriving inputs are then given. Validation of this process requires consistency. The model must satisfy the tank energy balance, and the model must reproduce the test results when simulating the test with inputs derived from that test. Previous studies suggesting similar algorithms are reviewed and shown in agreement with this work, with several important caveats. Finally, we discuss conclusions and future work.

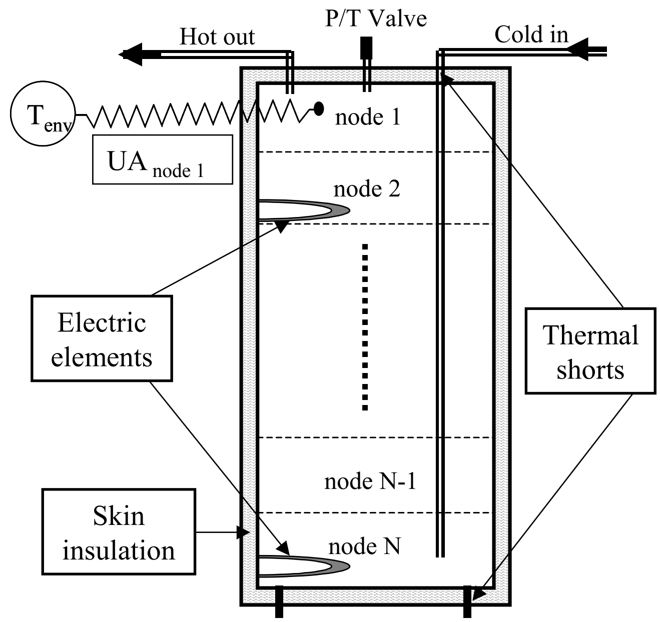

Fig. 1: Schematic electric tank, showing uniform skin insulation, electric heating elements, nodes, and some thermal shorts (pipes, P/T relief valve, and metal feet).

2. SIMULATION MODEL

Dynamic one-dimensional models for storage tanks have long been available in popular simulation tools, as in (2). These models are based upon mass and energy conservation applied to isothermal spatial zones called "nodes." A vertical electrical tank is shown schematically in Figure 1. It is divided into N vertically stacked nodes that are assumed isothermal. Any radial or azimuthal variations in temperature or flow are neglected, and the nodes model buoyancy stratification as results from heat inputs and from cold inlet water.

Key inputs governing long-term energy performance include the tank conductances and fuel-conversion efficiency. Conductances in (2) include a uniform U_{skin} and additional UA_n for each node. U_{skin} is intended to include insulation and film resistances, as in elementary texts. However, thermal shorts are of similar magnitude as skin conduction. Although most often set to zero, UA_n allows explicit specification of thermal shorts and their location. UA_n values are uncertain, because of variation in piping geometry and boundary conditions, as well as in heat-transfer basics (e.g., fluid convection of several types may exist in pipes, and is seldom modeled). If UA_n are set to zero, the U_{skin} must implicitly include thermal shorts. For gas tanks, an additional conductance input in (2) is the flue loss coefficient UA_{flue}, involving in principle a complex natural convection loop up the stack and back through the house to the burner. The loop operates continuously, contributing significantly to tank losses.

Conversion efficiency matters only for gas. $\eta_{c,elec} \equiv 1$, because essentially all the electrical power is deposited inside the tank. There are no mechanisms for any substantial losses. For gas, $\eta_{c,gas}$ is the fraction of the input-gas energy content instantaneously converted to thermal energy in the tank through complex combustion and heat-transfer processes. As evident from Equation (Eq.) 1 below, η_c could be measured by measuring the fuel input power and the tank temperature derivative when $\Delta T_{t\text{-}env} = 0$. η_c is typically ~0.8, with remaining energy going mostly up the stack as heat and incomplete combustion products.

Other important inputs include the number of nodes (N) and the jet-induced mixing of draw inlet water in the bottom of the tank. These parameters determine the potential for stratification and are key to answering dynamic questions such as runout and sizing. The first-hour rating (3, 4) could be used to determine a value for N. However, the result would apply only to the test-flow rate. Future work will address determination of these parameters.

Algorithms in (2) are based upon mass and energy balances on each node. The analogous instantaneous whole-tank energy balance can be expressed as:

$$CdT_t/dt = \eta_c P_{aux} - UA\Delta T_{t\text{-}env} - mc_p\Delta T_{out\text{-}in.} \qquad (1)$$

3. DOMESTIC HOT WATER TEST STANDARD

Test procedures for most DHW systems are described in (3), including tests for storage tanks. In the 24-hour performance test, 64.3 gallons of water is drawn out at $T_{set} = 135 \pm 5\ ^oF$ in six equal draws spaced one hour apart starting at t=0. Test results are published in (4), including values for the energy factor (EF), the recovery efficiency (RE), the tank volume V_t, and the auxiliary fuel input power P_{aux}. It is important to clearly understand the definitions of EF, and RE. Omitting certain corrections not germane here, (3) states that:

$$EF = Q_{out,d}/Q_{aux,d}, \quad (2)$$

where:

$Q_{out,d} = M_d c_p \Delta T_{out\text{-}in} = 41{,}092$ Btu/day

$Q_{aux,d}$ = total auxiliary energy input over the day

$$RE_{gas} = Q_{out,dr}/Q_{aux,dr} \quad (3)$$

where:

$Q_{out,dr} = (M_{dr} c_p \Delta T_{out\text{-}in})$ = energy withdrawn by the first draw of the 24-hour test;

$Q_{aux,dr}$ = energy input recovering from that draw.

RE_{elec} is not measured; it is set to 0.98 for all electric tanks (3). Measurement error in EF and RE of ~.025 (5) obviates reliable measurement, and $RE_{elec} \cong 0.98$ is a reasonable low-end value. *RE should not be confused with η_c.* η_c is a system constant, whereas RE depends on conditions under which it is measured (see Eq. 6). η_c is always greater than RE, by 0.002 - 0.03 for attainable values of UA and P_{aux}. The denominator in Eq. 3 ($Q_{aux,dr}$) makes up for both draw energy $Q_{out,dr}$ and tank losses during the recovery cycle, whereas the numerator $Q_{out,dr}$ does not include tank losses. This difference distinguishes RE from η_c.

We can relate these data to simulation-model inputs through the whole-tank energy balance. Integrating Eq. 1 over a day (d) and assuming no net change in tank temperature, we have the test-period energy balance:

$$\eta_c Q_{aux,d} = (M_d c_p \Delta T_{out\text{-}in} + UA \Delta T_{t\text{-}env} \Delta t_d) \quad (4)$$

Combining Eqs. 4 and 2 yields:

$$EF = Q_{out,d}/[(UA \Delta T_{t\text{-}env} \Delta t_d + Q_{out,d})/\eta_c) \quad (5)$$

Similarly, integrating Eq. 1 over a cycle of fuel input (a "burn" for gas), inserting Eq. 3, recognizing that the time duration Δt_{dr} of the fuel cycle is $\sim Q_{aux,dr}/P_{aux}$, and manipulating, yields:

$$\eta_c = RE + UA(\Delta T_{t\text{-}env})/P_{aux}. \quad (6)$$

RE depends on $\Delta T_{t\text{-}env}$, which is set at 67.5 ^oF in (3 and 4). Figure 2 gives variation in value of RE as a function of the temperature difference $\Delta T_{t\text{-}env}$ during the recovery, for the tank labeled "Standard Gas" in Table 1. RE varies from the value measured during the DOE test as $\Delta T_{t\text{-}env}$ changes, which is clear from Eq. 6. Error will be induced if RE is assumed constant under conditions differing from the DOE test.

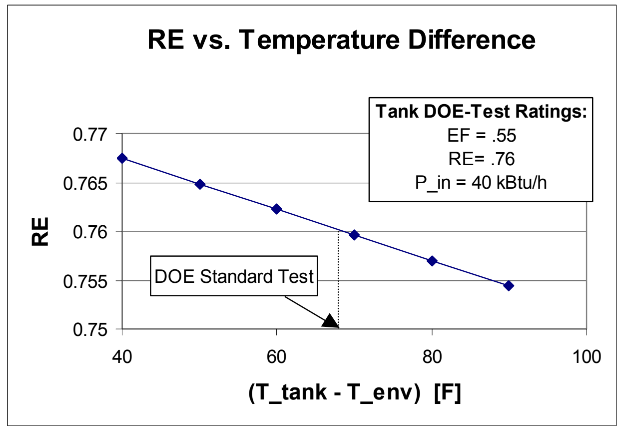

Fig. 2: Recovery Efficiency (RE) versus $\Delta T_{t\text{-}env}$, as predicted by Eq. 6. $\Delta T_{t\text{-}env}$ is the tank-to-environment temperature difference during the recovery cycle.

4. INPUT FORMULAE AND EXAMPLES

Different formulae are developed for UA_{elec} and UA_{gas}, because RE and η_c are treated differently for the two tank types. For gas tanks, $\eta_{c,gas}$ has to be inferred from RE_{gas} using Eq. 6, whilst for electric tanks $\eta_{c,elec} \equiv 1$ and $RE_{elec,(4)}$ is not useful. For an electric tank, Eq. 5 with $\eta_c \equiv 1$ yields:

$$UA_{elec} = Q_{out,d}(1/EF-1)/(\Delta T_{t\text{-}env} \Delta t_d). \quad (7)$$

For a gas tank, solving Eqs. 5 and 6 simultaneously gives:

$$UA_{gas} = (RE/EF-1)/[\Delta T_{t\text{-}env}(\Delta t_d/Q_{out,d} - 1/(P_{aux}EF))]. \quad (8)$$

Note that UA_{gas} is to be interpreted as the *total* loss coefficient including all skin losses, thermal shorts, and losses up the flue. Thus, with this process, $UA_{flue} \equiv 0$ in (2). Knowing UA, $\eta_{c,gas}$ is calculated from Eq. 6 with RE taken from (4).

Thermal shorts are implicit in UA, and the nature of the problem determines whether the shorts should be modeled and located explicitly via the inputs UA_n. For dominantly isothermal tanks, such as gas tanks or electric tanks with both elements at the same set point, thermal shorts can be subsumed into U_{skin}, effectively re-distributing the thermal shorts over all nodes. In this case:

$$U_{skin} = UA/A_{t(2)},\ UA_n = 0 \qquad (9)$$

where $A_{t(2)}$ is the tank surface area calculated as in (2) from shape assumption (e.g., a cylinder), V_t and H_t. The examples in Table 1 below use this option.

Alternatively, when performance depends in a significant way on the location of shorts (such as in two-element electric tanks with different T_{set} or in stratified solar tanks), thermal shorts are estimated externally first; using Eq. 9 will under-predict losses in this case. With some $UA_n \neq 0$:

$$U_{skin} = (UA - \Sigma_n UA_n)/A_{t(2)}. \qquad (10)$$

The resulting U_{skin} in this case embodies only skin insulation and errors in UA_n estimations; for gas tanks, it also includes the central flue.

The tank volume (V_t) reported in (4) is the manufacturer's quote for volume; it is not measured. Perhaps because of certain accuracy specifications, it appears that the true V_t is almost always 10% lower than specified for electric, and similarly 5% lower for gas tanks (6). We suggest making this correction to V_t data taken from (4) or from manufacturer specification. Tank volume affects the calculated surface area ($A_{t(2)}$).

The gas pilot is ignored in (2), with pilot energy included implicitly in the burner energy. This *ersatz* is tantamount to assuming $\eta_{c,pilot} \cong \eta_c$. The assumption would be exact if heat-transfer coefficients from air to tank walls were constants, independent of temperature and stack-flow rate. It is believed this *ersatz* introduces negligible error in energy consumption, as long as the standby loss rate always exceeds the pilot input. This is because the assumption is probably reasonably correct and pilot energy is relatively small. However, standby loss-recovery cycles will happen dramatically more often in the simulation than they will in reality. If the timing of such cycles were of importance, the tank models in (2) would have to be modified to include pilot lights. This would be relatively easy, but apparently has proven unnecessary to date.

Inferred values of UA and η_c are given in Table 1 for four tanks taken from (4), spanning available EF values for gas and electric. The inferred UA values range from a high of 10.5 Btu/h-F for $EF_{gas,min}$ =.56 to 1.34 Btu/h-F for $EF_{elec,max}$ = .95. Percentage uncertainty in derived UA values are also shown in the table, based upon uncertainties of ± .025 in δEF and δRE (5). UA uncertainty is ~20%, increasing to ~50% for the premium electric tank. Reduction of this error is necessary to make the ratings data useful for deriving reliable inputs, especially to show how the EF values might be biased upward (5). This error should be reduced by a factor of five, which would bring maximum UA error at $EF_{elec,max}$ to ~10%. The political inertia around rating changes will present substantial challenge.

Consistency requires that the energy balance of Eq. 4 be satisfied with the derived values of UA and η_c. Errors in the energy balance (normalized by dividing by $Q_{out,d}$) are less than 0.1% for all four tanks in Table 1, using the derived values shown in Table 1 for UA and η_c. Consistency also demands that the model, when simulating the test, reproduce the EF and RE used to derive the model inputs. Ratings for four tanks were taken from (4), and model inputs derived using Eqs. 6 through 8. The 24-hour test was mimicked in detail, and EF were then calculated from the simulation results, including corrections specified in (3) for average tank temperature and outlet temperature that do not maintain average values at T_{set}. Results are shown as $EF_{(2)}$ in Table 1. It can be seen that $EF_{(2)}$ values agree with the assumed EF within ±.003.

5. COMPARISON TO PAST WORK

A "standby loss coefficient" that incorporates fuel-conversion efficiency is defined in (7), denoted here as L_{st}:

$$L_{st} = Q_{aux,st\text{-}loss}/\Delta T_{t\text{-}env,st} \qquad (11)$$

Since $Q_{aux,st\text{-}loss} = Q_{st\text{-}loss}/\eta_c$, L_{st} is related to an overall loss coefficient $UA_{(7)}$ as

$$UA_{(7)} = \eta_c L_{st} \qquad (12)$$

where the subscript "(7)" denotes the UA inferred through L_{st}. It is shown in (7) that:

$$L_{st} = (1/EF - 1/RE)/[\Delta T_{t\text{-}env}(\Delta t_d/Q_{out,d} - 1/(P_{aux}RE))]. \qquad (13)$$

Because the notation "UA" is used for L_{st} in (7), one might erroneously identify L_{st} with UA. Although Eqs. 12 and13 are formally correct, care is needed when using them with electric tanks. RE_{elec} is not measured, is generally *inconsistent* with EF_{elec}, and should not be used in Eq. 13.

These analytical conclusions are corroborated by numerical results shown in Table 1. For gas tanks, it can be seen that $UA_{(7),gas} \cong UA_{gas}$. However, for the electric tanks, $UA_{(7),elec}$ values are in error by as much as 34% for the tank with EF = 0.95. The energy balance is not satisfied to 0.3% and 2% for the standard and premium electric tanks, respectively. If Eq. 6 (with assumption $\eta_c \equiv 1$) is used to calculate a new value for RE_{elec}, then use of Eq. 10 and 11 also agrees identically with our results for electric. Recalculated RE_{elec} are also given in Table 1 in parentheses next to $RE_{elec,(4)}$. It can be seen that the $UA_{(7)}$ error is largest for the tank with the greatest divergence between recalculated and mandated RE_{elec} values.

Ref. (8) numerically estimates long-term DHW energy use from ratings data; a value is given for total losses from the tank over period Δt, $Q_{loss,\Delta t,(8)}$. $UA_{(8)}$ can be inferred as:

$$UA_{(8)} = Q_{loss,\Delta t,(8)}/(\Delta T_{t\text{-}env}\Delta t). \tag{14}$$

For electrical tanks, it is assumed that $\eta_{c,elec} \equiv 1$ and RE_{elec} is not used, as in this paper. Table 1 shows that numerical results from (8) agree with our results, with one caveat. It is noted in (9) that for gas tanks the input in (8), labeled "Recovery Efficiency," should be taken as η_c, not RE. Guidance in (9) is that $\eta_c = RE + (.01\text{-}.02)$. For $UA_{(8)}$ in Table 1, we input the previously calculated $\eta_{c,gas}$. In the context of (8) alone, iteration would be required to derive η_c from outputs and RE, using Eq. 6. If RE is taken as the input in question, $UA_{(8)}$ differs by approximately 8% from our results for the two gas tanks.

6. CONCLUSIONS AND FUTURE WORK

Algorithms based on whole-tank energy balances are given to infer values for the loss coefficient UA and the conversion-efficiency η_c from EF, RE, and P_{aux} values published in (4). Algorithms differ for electric or gas tanks, because RE and η_c must be handled differently for gas and electric. For gas tanks, $UA_{flue} = 0$ and the pilot, if present, is implicitly included in main burner energy. The method is consistent in that the energy balance is satisfied and that the model with test-based inputs reproduces the starting-test data. Results are consistent with previous work, with certain caveats related to the difference between RE and η_c.

The error in UA using rating data is 20%-50%, using $\pm$ .025 uncertainty (5) on EF, RE, as reported in (4). If so, the EF rating for electric tanks (with proposed $EF_{min} = 0.9$ and $EF_{max} = .95$ on practical grounds) is practically meaningless. This is unacceptable. An uncertainty of $\pm$ .005 in EF is probably attainable (5), and would make the maximum UA uncertainty on the order of 10%. However, barriers to any rating changes are severe.

Future work is needed to provide a procedure for deriving a well-determined value of the node number and to characterize mixing upon draw as a function of flow rate. We plan to simultaneously identify node number and flow-rate-dependent draw-inlet mixing fractions. We will do this by fitting the outlet temperature profile over draws, and purging out all energy with the fuel input off, at multiple flow rates and spanning typical usage (6). Because of the central flue, results may vary somewhat for gas or electric, and both tank types will be tested. It is believed that the results should apply to any tank having similar tank and dip-tube geometry.

7. NOMENCLATURE

Symbols

A	Surface area of tank
C	Tank thermal capacitance
c	Specific heat
EF	Energy factor from test
H	Tank height
L	Pseudo loss coefficient
M	Mass of water drawn
m	Mass flow rate
N	Total number of nodes
P	Power into tank
Q	Quantity of energy
RE	Recovery efficiency from test
T	Temperature
t	Time
U	U value
UA	Total loss coefficient of the storage tank
V	Tank volume
δ	Variation in, derivative
Δ	Difference of
η	Efficiency of fuel conversion
Σ	Summation over nodes

Subscripts

aux	Auxiliary (electricity, gas, etc.)
c	Conversion of fuel to thermal energy in water
d	One day, 24 hours
dr	Draw
elec	Electricity auxiliary
env	Environment surrounding tank
flue	Gas tank central flue and stack
gas	Natural gas auxiliary (or any fossil fuel)
in	Input to the tank (fuel or mains)
loss	Losses from tank to environment
max	Maximum value of
min	Minimum value of
out	Out of the tank (load or temp)
out-in	Out of tank – into tank (temperature or energy)
p	Constant pressure specific heat
pilot	Pilot light in gas tank
set	Set point for thermostat
skin	Skin of tank (films+insulation)
st	Standby
t	Tank
t-env	Tank – environment (temperature)
(n)	Value calculated as in or taken from Ref. n.

8. ACKNOWLEDGMENTS

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TABLE 1. RATING DATA AND DERIVED PARAMETERS FOR FOUR RATED TANKS

Parameter	Units	Standard Gas	Premium Gas	Standard Elec.	Premium Elec.
EF[1]	-	0.55	0.61	0.86	0.95
RE[1]	-	0.76	0.76	0.98 (.981)[2]	0.98 (.995)[2]
P_{aux}[1]	kBtu/hr	40	34	15.4	18.8
UA[3]	Btu/hr-F	10.50	6.799	4.129	1.335
$\delta UA/UA$[4]	-	21%	27%	21%	53%
η_c[5]	-	0.778	0.773	1.000	1.000
$EF_{(2)}$[6]	-	0.553	0.612	0.861	0.948
L_{st}[7]	Btu/hr-F	13.50	8.79	4.074	0.901
$UA_{(7)}$[8]	Btu/hr-F	10.50	6.799	4.074	0.901
$UA_{(8)}$[9]	Btu/hr-F	10.49	6.799	4.127	1.336

[1] Data in the dark gray area at the top of the table are taken from (4). RE_{elec} data in parentheses are calculated as per note 8.
[2] Values for RE in parentheses are calculated from Eq. 6, using UA from this table and rated P_{aux}.
[3] Using Eqs. 7 and 8 for electric and gas, respectively.
[4] Assumes an uncertainty of ± .025 on EF, and RE_{gas}, with quadrature addition for gas (δEF, δRE) contributions.
[5] Using Eq. 6, for gas; $\eta_c \equiv 1$ for electric tanks.
[6] EF computed from results of the simulation model (2) when simulating the standard test.
[7] L_{st} is calculated from Eq. 13, using RE values from (4).
[8] $UA_{(7)}=\eta_c L_{st}$, as in Eq. 12.
[9] $UA_{(8)}$ from annual runs with (8) at DOE test conditions, using η_c as the input for "Recovery Efficiency," as in (9).