THE MERCURY EJECTOR REFRIGERATION SYSTEM AS A MEANS OF COOLING AND HEATING RESIDENCES

By

Eugene H. Wood

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Chapter I

INTRODUCTION

"The cooling of residences has formerly been given little consideration except in the case of the more expensive ones. While many years will be required and much educational work before general house cooling can be successful, nevertheless the movement is well under way and will be hastened as better and cheaper equipment becomes available. In the small home a consciousness has arisen toward electrification, and in this the all year round air condition will play no small part. There is as much logic in cooling to the zone of comfort as in heating to it, especially when both functions can be performed with much of the same equipment. The added cost will be charged to convenience, which has a real, though elusive value"[1].

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1. Fan Engineering, Buffalo Forge Company, p. 274.

Chapter II

PURPOSE OF THESIS

In practically all refrigeration machines the heat which is absorbed by the condenser water is wasted. The reason for this is that the heat cannot be sufficiently concentrated to raise the temperature of the condenser water high enough to be utilized for heating or other purposes for which it might be desired. Because of the fact that mercury condenses at a very high temperature, the temperature of condensing water may be raised correspondingly high when used as a cooling medium in a mercury condenser. The mercury ejector type of refrigerator is capable of producing any desired amount of refrigeration and at the same time water used for the condenser can be raised to a very high temperature. With this in mind, an examination will be made in the following pages of the possibilities of using the mercury ejector refrigerator for the double purpose of heating and cooling. For a specific application the size of unit will be determined to fulfill the refrigeration and heating requirements for the typical home in the eastern part of Tennessee. The size of unit refers to the amount of mercury required to be evaporated per hour.

Chapter III

THEORY INVOLVED

During the past few years a great majority of the theaters, auditoriums, and other places of public assemblage have been equipped with refrigeration apparatus for air conditioning. Air conditioning is beginning to play an important part in modern industry also. The field is large and has become important both from the standpoint of its benefits to the public and to the promoters of public welfare. Business has found it a paying proposition, and the cost of installation and operation of this equipment is comparatively small compared with the returns.

The case is different in the average home. A refrigerated home at the present time would be considered a very expensive luxury; in fact, it is practically impossible for the person with a limited income to equip his home with the present refrigeration equipment. It is the problem of the engineer to improve the efficiency of the existing machine or discover new and cheaper methods of producing refrigeration. It is rumored that the General Electric Company has developed a refrigeration system of air conditioning which is applicable to the average home, and that it will be put on the market as soon as the present depression subsides. No doubt, with such a large potential industrial enterprise existing no effort will be spared until the problem is solved.

Before going into the description of the mercury

ejector cycle, it might be well to recall briefly the familiar gas compression cycle. The refrigerant comes to the compressor in the form of a gas. It is compressed, and in the process of compression is heated to some higher temperature. The heated gas is then discharged to a condenser which is water cooled. Here much of the heat is taken from the gas; it is lowered in temperature below the boiling point and collects in the bottom of the condenser as a liquid, but still under high pressure. From the condenser it passes down through a throttling valve into the coils of the evaporator. The ammonia, after entering the evaporator vaporizes by absorbing its latent heat of evaporation from the surroundings. The gas is then ready to pass on to the compressor for a repetition of the cycle.

The mercury ejector refrigerator produces its cooling by forced evaporation of water. The mercury pump, which is a vapor aspirator, sucks water vapor from the surface of the liquid water in the evaporator and compresses it sufficiently for it to condense in the condenser. The liquid water then runs back to the evaporator and the process is repeated.

A boiler is used to vaporize the mercury, the resulting vapor passing through a nozzle into a compression funnel where it is condensed. The liquid mercury flows back to the boiler by gravity. On its way back to the boiler the mercury purges the system. This is explained in detail below.

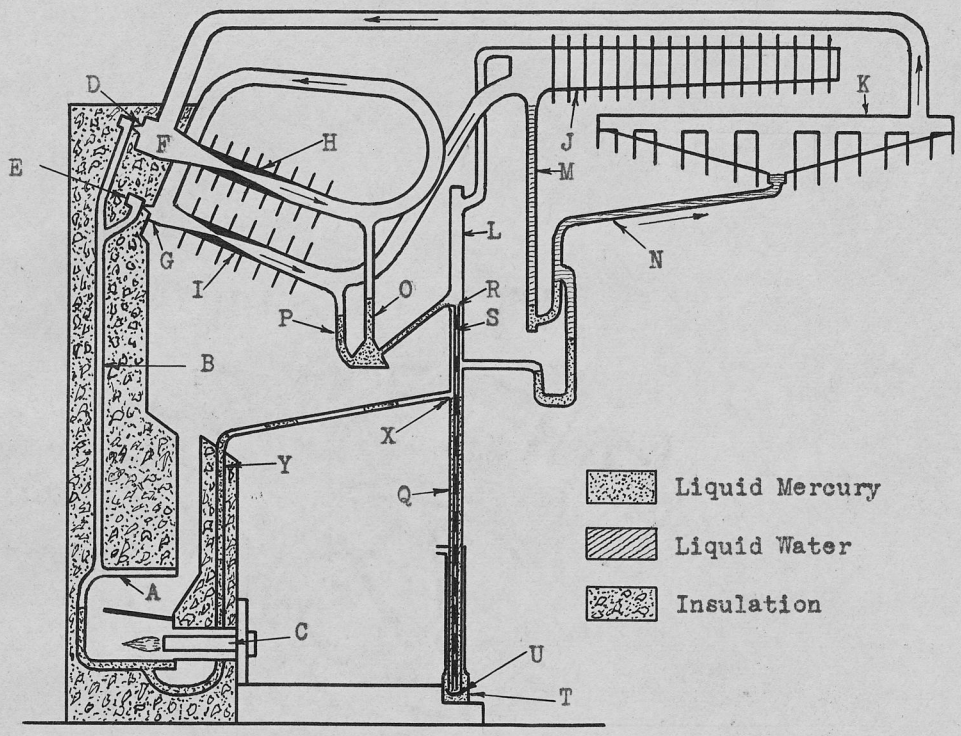

DIAGRAM OF MERCURY EJECTOR REFRIGERATOR

Fig. 1

A ..... Boiler
B ..... Mercury Vapor Riser
C ..... Gas Burner
D, E .. Nozzles
F, G .. Mixing Chambers
H, I .. Compression Funnels
J ..... Water Vapor Condenser
K ..... Evaporator
L ..... Purger
M ..... Water Condenser Drain
N ..... Water Inlet into Evaporator
O, P .. Funnel Drains
Q ..... Tube Containing Purger Capillary
R ..... Collar Supporting Top of Capillary
S ..... Purger Capillary
T ..... Outer Purger Container
U ..... Purger Shield
X ..... Spillover Point to Boiler
Y ..... Boiler Feed Tube

Referring to figure 1 a detailed explanation of operation will be made. When heat is applied to boiler A mercury vapor rises through tube B and passes through nozzles D and E. As shown in the drawing, these nozzles act in parallel. In passing through them the vapor drops in pressure and acquires a very high velocity. When the vapor reaches the compression funnels the energy due to its velocity is changed to kinetic energy and the pressure increases. Because of the increase in pressure the mercury vapor is condensed on the walls of the funnel, these walls being cooled either by air or water. In the mixing chambers F and G water vapor is entrained in the mercury stream and passes with it through the compression funnels. As the mercury condenses the water vapor passes on to the water vapor condenser. Here the water vapor is condensed, the liquid running out at the bottom and standing in the tube M in a liquid column of sufficient height to depress the mercury which fills the bottom of this tube sufficiently to permit the water to pass into the tube N and then into the evaporator.

Mercury condensed in the two funnels H and I runs down tubes O and P and then up to tube Q. The collar R at the top of the capillary tube S causes the mercury to flow down the capillary tube instead of between Q and S. Mercury flowing down through the capillary tube entrains non-condensing gases and carries them down between slugs of mercury until finally, due to the weight of mercury

resting on it, the air is compressed slightly above atmospheric pressure. The air emerges from the bottom of the capillary tube into container T. It then bubbles up through the surrounding mercury and escapes through the opening in T. The mercury flows out between shield U and tube Q. It then flows upward between Q and S to the point X. From there it flows through tube Y back to the boiler[1].

Efficiency of Cycle Computed

The efficiency of the cycle, to be used in later calculations, will now be computed. The method followed was obtained from an article by Mr. Whitely in the Sep. issue of the 1932 Refrigeration Engineering Magazine. He assumed a temperature of 20° F. in the evaporator. An evaporator temperature of 40° would be more desirable for our purpose, so that a recalculation of the efficiency is necessary.

The cycle divides itself into two parts: (1) the production of work from heat, and (2) the creation of refrigeration from the work so produced. The maximum work theoretically derivable from a given amount of heat H, received at absolute temperature T_1, and rejected at a temperature T_2 is $(T_1 - T_2) / T_1$. In the present case, assuming a mercury boiler pressure of 15# / sq. in. abs. $T_1 = 677 + 460 = 1137°$ absolute; the jet temperature at

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1. Whitely, "Mercury Ejector Refrigerator", Refrig. Eng. Magazine, Sep. 1932.

the mouth of the first stage nozzle is that corresponding to a mercury vapor pressure of 0.1217# / sq.in., or about 330°. The maximum amount of work that can be obtained from this cycle cannot exceed (677 - 330) / 1137 = 30.5% of the heat put in.

Mr. Whitely works out further fundamental limitations. For instance the difference between the Rankine and the Carnot cycle reduces the above figure to 28.5%. Another factor is the loss of energy in collision between mercury and water vapor in entrainment. The law of conservation of momentum makes possible the accurate computation of this fractional energy loss for uniform mixing of any given proportion of mercury and water vapor; it is a factor equal to the ratio of the mass of water vapor to the total mass of both substances. This ratio is small in any practical case, due to the much greater mass of the mercury molecule. For an equimolecular proportion, which corresponds roughly to this efficiency, it is about 3%, which reduces the limiting efficiency from 28.5% to 25.5%.

A necessary recompression of much of the mercury vapor along with the water vapor gives rise for a further correction in efficiency. The walls of the compression funnel are cool enough to condense any mercury reaching them, and tests show that over 90% of the mercury is condensed and drained there. Therefore an easy assumption that should approximate the truth is that half as much

energy is used in mercury recompression as would compress it all to the final pressure. Assuming this condenser pressure to be 75 mm. of mercury, the mercury compressed to this pressure (still in the state of wet vapor) would be at 450°F. Since the assumption is rough, the difference between Rankine and Carnot cycle can be ignored, and energy quantities can be taken as proportional to corresponding temperature intervals. Thus the corrected limiting efficiency will be

$$\left[677 - (450 + 330)/2\right] / \left[677 - 330\right] \left[25.5\%\right]$$
$$= 21\%$$

On the side of the cycle in which the work is utilized to produce refrigeration, it may be noted that operating between 40 and 115°F. the limiting ratio of refrigeration obtainable to work expended is equal to

$$(T_1) / (T_2 - T_1) = (40 + 460) / (115 - 40)$$
$$= 500 / 75 = 6.67$$

As the water vapor is necessarily superheated during compression due to its mixing with the mercury vapor, a further correction can be applied, provided the average temperature of compression can be estimated. This can only be done roughly, since the vapor lies between a stream of hot mercury at 330 to 450° and the funnel walls at 160 to 230°; its low initial temperature is offset by the heat put into it by the collision energy noted above. On the whole, 320° seems to be a reasonable estimate for the average; it is slightly higher than the mean of the four

boundary temperatures just noted. This alters the above ratio of 6.67 by the factor

$$[460 + (40 + 115 / 2)] / (320 + 460)$$

$$= 537.5 / 780 = 0.69$$

making the corrected limiting ratio of refrigeration effect to available work equal (0.69) (6.67) = 4.6 Combining this with the other limiting ratio of available work to heat input, 21%, gives a limiting ratio of refrigeration to heat input equal to

$$(0.21) (4.6) = 0.965$$

Mr. Whitely states further that the present actual performance of this type of unit (B.T.U. of useful refrigeration divided by B.T.U. going into mercury) amounts to about 32% of the above theoretical figures. The value 0.965 will then be reduced to (0.965) (0.32) = 0.31

Let us now investigate the possibilities of using this mercury ejector system of refrigeration as a means of alternately heating and cooling the average residence, and determine the size of such a unit that will accomplish this purpose.

Chapter IV

CALCULATION OF HEATING AND REFRIGERATION REQUIREMENTS FOR AVERAGE HOME

Construction of House and Other Necessary Data

The walls are of frame construction, and the outside is a brick veneer on sheathing on 2" X 4" studding. The inside construction is of metal lathe and three-fourths inch plaster. The ceiling consists of 1" flooring, joists, metal lathe, and three-fourths inch plaster. The roof consists of slate on sheathing. The pitch of the roof is 1 to 2. The floors are of 2" flooring on joists. The windows are double hung and weather stripped. The height of the rooms is ten feet.

The left side of the house is subjected to an average wind velocity of 15 mi. per hour. The remaining sides are subjected to an average wind velocity of 5 mi. per hour. The left side contains a wall area of 308 sq. ft. and a window area of 125 sq. ft. The feet of window cracks is 155.5. The remaining sides contain a wall area of 832 sq. ft. and a window area of 255 sq. ft. The feet of window crack is 413. The total ceiling area is 1440 sq. ft. The floor area is also 1440 sq. ft.

The heating requirements are based on an outside temperature of 0°F. and an inside temperature of 70°F. Air due to infiltration is assumed sufficient for ventilation. Other data will be given as needed.



Fig. 2

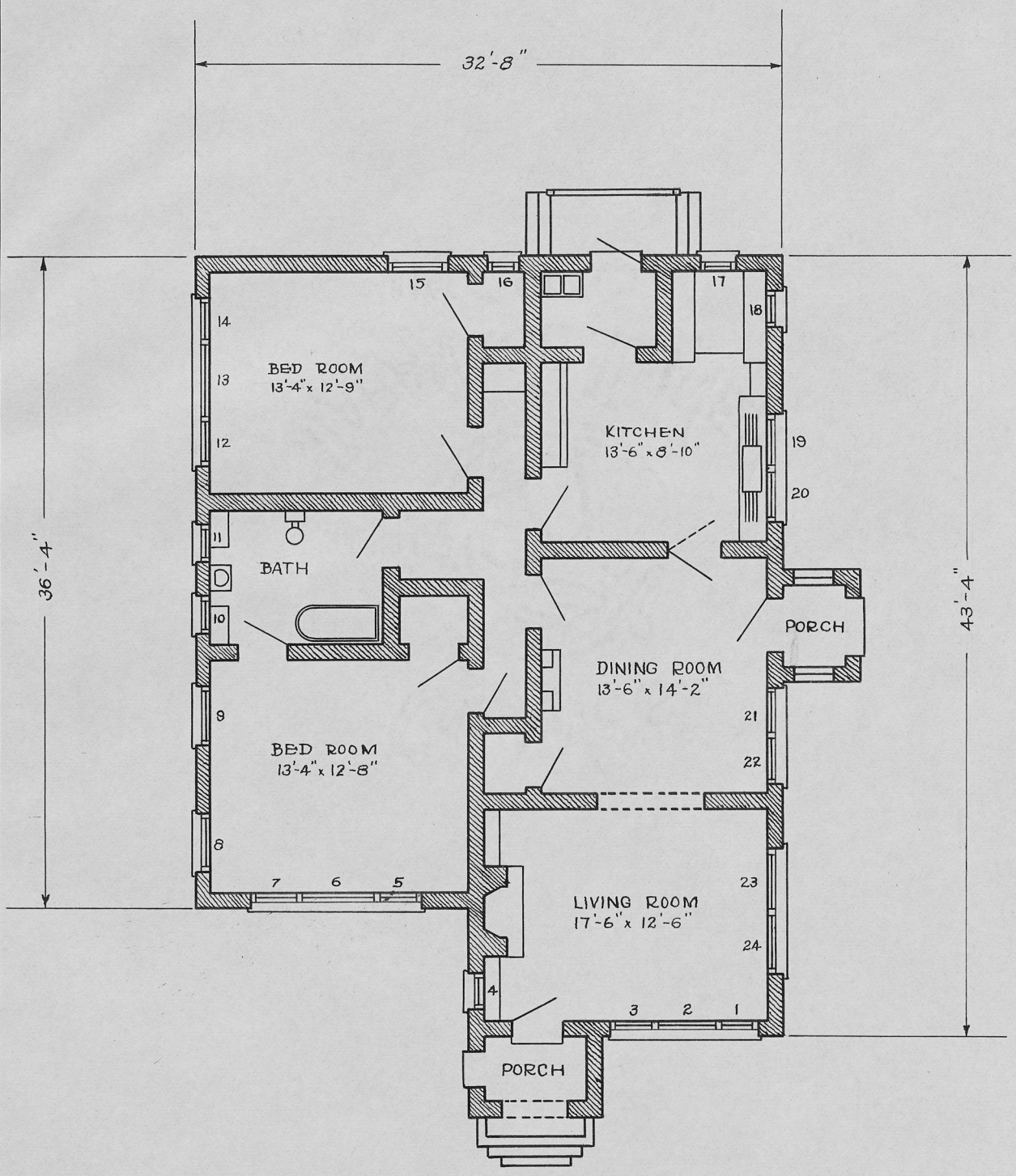
32'-8"
36'-4"
43'-4"
BED ROOM
13'-4" x 12'-9"
KITCHEN
13'-6" x 8'-10"
BATH
PORCH
DINING ROOM
13'-6" x 14'-2"
BED ROOM
13'-4" x 12'-8"
LIVING ROOM
17'-6" x 12'-6"
PORCH
1
2
3
4
5
6
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Fig. 3

Calculation of Heat Losses

1. Heat loss through outside wall subjected to 15 mile wind. From table p. 187 Fan Engineering, C = thermal conductance = 0.258 B.T.U. per hour per sq. ft. per ° F. difference. From table p. 54 surface coefficients for still and moving air are f_i = 1.34 B.T.U. per hour per sq. ft. per ° F and f_o = 4.02 B.T.U. per ..... respectively.
The overall coefficient of heat transmission is

$$U = 1 / [(1 / C) + (1 / f_i) + (1 / f_o)]$$

$$U = 1 / [(1 / 0.258) + (1 / 1.34) + (1 / 4.02)]$$

$$U = 1 / 4.88 = 0.205 \text{ B.T.U. / hr. / sq. ft. / °F}$$

Wall area (left side) is 308 sq. ft. and $t_i - t_o$ is 70 - 0 = 70°F.
Therefore H = (308)(0.205)(70) = 4420 B.T.U. / hour.

2. Heat loss through glass windows subjected to 15 mile wind. From table p. 190 "Fan Eng." C = 1.13 B.T.U. / hr. / sq. ft. / °F.

f_i = 1.5 per .....

f_o = 4.5 per .....

$$U = 1 / [(1 / 1.5) + (1 / 4.5) + (1 / 1.13)]$$

$$U = 1 / 1.77 = 0.565 \text{ B.T.U. / hr. / sq. ft. / °F.}$$

Window area (left side) is 125 sq. ft.

$$H = (125)(0.565)(70) = 4940 \text{ B.T.U. / hour.}$$

3. Heat losses through walls subjected to 5 mile wind. "The effect of wind is to increase both convection and infiltration losses, and therefore to make necessary a

greater supply of heat to maintain the desired inside temperature. This may be approximated by considering the effect as equivalent to an increased drop in outside temperature"[1]. The National District Heating Association reports that the equivalent drop in temperature per mile increase in wind velocity for an outside temperature of 0°F. is 1.3°F.

As stated above, the front, rear, and right side of the house considered are subjected to a wind velocity of 5 miles per hour. The tables used in these calculations are based on a a wind velocity of 15 miles, so that the outside temperature will be increased by the amount (15 - 5) (1.3) = 13°F, making t_o = 13°F. and (t_1 - t_o) = (70 - 13) = 57°F. The overall heat transmission coefficient is 0.205 B.T.U. / hr. / sq. ft. / °F

Wall area is 832 sq. ft. and t_1 - t_o is 70 - 13 = 57°F.

H = (832) (0.205) (57) = 9,730 B.T.U. / hour.

4. Heat loss through windows subjected to 5 mile wind.

Overall heat transmission coefficient is 0.565

Window area is 255 sq. ft. and t_1 - t_o is 57°F.

H = (255) (0.565) (57) = 8210 B.T.U. / hour.

5. Heat loss through roof. From table on p. 190, "Fan Eng". for pitched roofs over unheated attics

C = 0.172 B.T.U. / hr. / sq. ft. / °F.

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1. R. D. Madison, Fan Engineering, p. 191.

f_i = 1.34 B.T.U. / hr. / sq. ft. / °F.

f_o = 4.02 " " " " "

U = 1 / [(1 / 0.172) + (1 / 1.34) + (1 / 4.02)]

U = 1 / 6.81 0.147 B.T.U. / hr. / sq. ft. / °F.

Area of ceiling is 1440 sq. ft.

H = (1440) (0.147) (70) = 14,800 B.T.U. / hour.

6. Heat loss through floors. From table p. 188, "Fan Engineering" C = 0.339 B.T.U. / hr. / sq. ft. / °F.

Assume still air conditions and $t_i - t_o$ = 65 - 40 = 25°F.

U = 1 / [(1 / .339) + (1 / 1.34) + (1 / 1.34)]

U = 1 / 4.44 = 0.225 B.T.U. / hr./ sq. ft./°F.

Floor area is 1440 sq. ft.

H = (1440) (0.225) (25) = 8100 B.T.U. / hour.

7. Infiltration through walls.

From the table p. 195 infiltration for this type of wall construction with a 15 mile wind is 0.22 cu.ft./ hr./ sq. ft. of wall surface. The wall area subjected to a 15 mile wind is 308 sq. ft.

H = (sp.ht.) (density) ($t_i - t_o$) (cu. ft. / hr.) (wall area)

H = (0.24) (0.0731) (70) (0.22) (308) = 83 B.T.U. / hour.

From table on p. 195 infiltration for this type wall construction with a 5 mile wind is 0.04 cu. ft. / hr. / sq. ft. of wall area. The wall area subjected to 5 mile wind is 832 sq. ft.

H = (0.24) (0.0731) (70) (0.04) (832) = 41 B.T.U./ hr.

Note: Page references unless specified refer to "Fan Eng".

8. Infiltration through window cracks.

From the table on p. 196 infiltration for a wind velocity of 15 miles is 12.8 cu. ft. / hr./ ft. of crack.

Feet of crack subjected to 15 mile wind = 156

H = (0.24)(0.0731)(70)(12.8)(156) = 2455 B.T.U. / hour.

From the same table infiltration for a wind velocity of 5 miles per hour is 2.2 cu. ft. / hr. / ft. of crack.

Feet of crack subjected to 5 mile wind = 413

H = (0.24)(0.0731)(70)(2.2)(413) = 1120 B.T.U. / hour.

Heat produced by cooking.

It is estimated that the heat produced by cooking will furnish 50% of the heating requirements for the kitchen. The heating requirements for the kitchen will be computed.

9. Heat loss through walls of kitchen

The overall heat transmission coefficient U is 0. 205

Total wall area - window area is 290 - 44 = 246 sq. ft.

$t_i - t_o = 70 - 13 = 57°F$

H = (246)(0.205)(57) = 2870 B.T.U. / hour.

10. Heat loss through windows of kitchen.

Overall heat transmission coefficient U is 0.565

Window area is 44 sq. ft. and $t_i - t_o = 57°F$.

H = (44)(0.565)(57) = 1420 B.T.U. / hour.

11. Infiltration through walls of kitchen.

Infiltration is 0.04 cu. ft. / hr. / sq. ft.

H = (0.24)(0.0731)(70)(0.04)(246) = 12 B.T.U. / hour.

12. Infiltration through window cracks.

Infiltration is 2.2 cu. ft. / ft. of crack.

Feet of crack = 65

H = (0.24)(0.0731)(70)(2.2)(65) = 176 B.T.U. / hour.

13. Heat loss through roof.

Overall heat transmission coefficient U is 0.147

Ceiling area = (13)(16) = 208 sq. ft.

H = (208)(0.147)(70) = 2140 B.T.U. / hour.

14. Heat loss through floor.

Overall heat transmission coef. U is 0.225

$t_i - t_o$ = 65 - 40 = 25°F.

Floor area is 208 sq. ft.

H = (208)(0.225)(25) = 1170 B.T.U. / hour

Total heat requirements for kitchen = 7788 B.T.U. / hour.

7788 / 2 = 3900 B.T.U. / hr. supplied by heat produced by cooking.

Itemized heat losses:

4420 B.T.U./hr. through wall subjected to 15 mile wind

4940 " " " windows " " " " "

9730 " " " walls " " 5 " "

8210 " " " windows " " " " "

14800 " " " roof "

8100 " " " floors

83 " " " due to infil. thru walls (15 mi.)

41 " " " " " " " (5 mi.)

2455 B.T.U./hr. loss due to infil. thru window cracks (15mi.)

1120 " " " " " " " " " (5 mi.)

53,900 B.T.U. / hr. = total loss

53,900 - 3900 = 50,000 B.T.U./ hr. required for heating this particular house with an outside temperature of 0°F and an inside temperature of 70°F.

Refrigeration requirements:

"The calculation of cooling load is made in a manner similar to that for a heating load with the exception of sun effect which is a relatively large factor. The heat gained by transmission into the building is obtained by multiplying the square feet of each type of surface by its transmission coefficient and also by the difference between the outside and inside air temperatures"[1].

A maximum outside temperature of 96°F. is assumed. An inside temperature of 72°F. will be maintained, so that $t_o - t_i = 96 - 72 = 24°F.$

1. Heat gain through wall subjected to 15 mile wind.

U = 0.205 B.T.U./hr./sq.ft./°F.

Wall area = 308 sq. ft.

H = (308)(0.205)(24) = 1520 B.T.U./hr.

2. Heat gain through windows subjected to 15 mile wind.

Note: Values of heat transmission coefficients used in these calculations are of course the same as used in the previous calculations of heating requirements.

1. R.D.Madison, Fan Engineering, p. 271.

U = 0.565 B.T.U./hr./sq.ft./°F.

Window area = 125 sq. ft.

H = (125)(0.565)(24) = 1700 B.T.U. / hour.

3. Heat gain through walls subjected to 5 mile wind.

U = 0.205 $t_o - t_i$ = 96 - (72 + 13) = 11°F. (refer to item 3 under heating requirements)

Wall area = 832 sq. ft.

H = (832)(0.205)(11) = 1870 B.T.U. / hour

4. Heat gain through windows subjected to 5 mile wind.

U = 0.565 window area = 265 sq. ft.

H = (255)(0.565)(11) = 1580 B.T.U. / hour.

5. Heat gain through roof.

U = 0.147 ceiling area = 1440 sq. ft.

H = (1440)(0.147)(24) = 2120 B.T.U. / hour.

6. Heat gain through floors.

No loss is assumed through the floors.

7. Infiltration through walls.

Infiltration (15 mile wind) = 0.22 cu. ft./hr./sq.ft.

Wall area = 308 sq. ft.

H = (0.24)(0.0731)(24)(0.22)(308) = 29 B.T.U. / hour.

Infiltration (5 mile wind) = 0.04 cu. ft. / hr./sq.ft.

Wall area = 832 sq. ft.

H = (0.24)(0.0731)(24)(0.04)(832) = 14 B.T.U. / hour.

8. Infiltration through window cracks.

Infiltration (15 mile wind) = 12.8 cu. ft./ft. of crack.

Feet of crack = 156

H = (0.24)(0.0731)(24)(12.8)(156) = 842 B.T.U. / hour.

Infiltration (5 mile wind) = 2.2 cu. ft./ft. of crack.

Feet of crack = 413

H = (0.24)(0.0731)(24)(2.2)(413) = 383 B.T.U./ hour.

It has been estimated that heat produced by cooking will equal 3900 B.T.U./ hour. This is probably a fair estimate, since the cooking element on the stove will be well hooded in the summer to carry off as much heat as possible.

Sun effect:

In the calculation of sun effect for maximum conditions it is necessary to figure the gain in heat from the roof and one wall only - that wall subjected to the greatest sun effect. It will be assumed that the left wall will be subjected to the greatest sun effect, and this will be the west wall. The windows are equipped with awnings, so that the sun effect on them will be negligible.

Note: The values of heat flow given below are more or less inaccurate because accurate tables could not be obtained for rate of flow through the specific materials used in the construction of this house.

1. Heat gain through left wall.

Heat flow per hr./sq.ft. wall surface is 13.2 B.T.U.

Wall area = 308 sq. ft.

H = (308)(13.2) = 4060 B.T.U. / hour.

2. Heat gain through roof.

It will be assumed that due to sun effect the temperature in the attic under the roof will be raised to 110°F. The heat flow through the ceiling will now be calculated where $t_o - t_1 = 110° - 72° = 38°$

From the table on page 188 C is 0.298 and for this type of ceiling

$U = 1 / [(1 / 0.298) + (1 / 1.34) + (1 / 1.34)]$

$U = 1 / 4.854 = 0.206$ B.T.U./hr./sq.ft./)°F.

Ceiling area is 1440 sq. ft.

$H = (1440)(0.206)(110 - 72) = 11{,}300$ B.T.U. / hour.

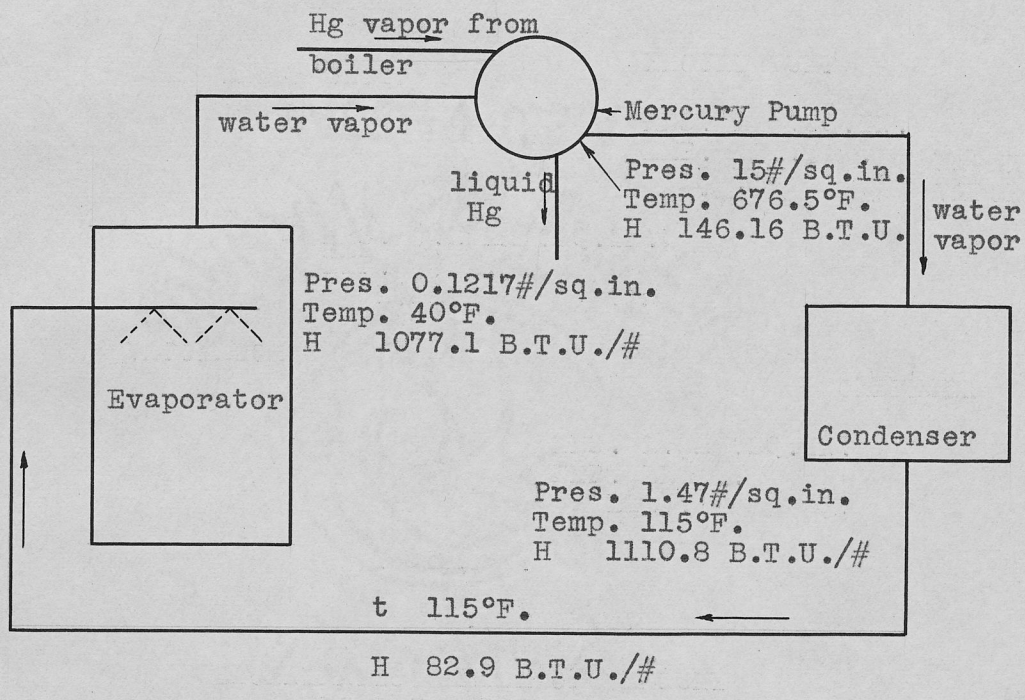
Total refrigeration required per hour:

1520 B.T.U. wall subjected to 15 mile wind.
1700 " windows " " " " "
1870 " walls " " 5 " "
1580 " windows " " " " "
2120 " gain through roof.
29 " infil. thru wall subjected to 15 mile wind.
14 " " " walls " " 5 " "
842 " " " window cracks sub. to 15 mi. wind.
383 " " " " " " 5 " "
3900 " produced by cooking.
4060 " sun effect on left wall.
11300 " " " " roof.
29,318 B.T.U. Total per hour.

Having now the heating and refrigeration requirements, and the theoretical and actual efficiencies of the mercury ejector refrigerating system, we are ready to calculate the size unit required.

Chapter V

CALCULATIONS



Diagrammatic Sketch of Mercury
Ejector Refrigerator
Fig. 4

Refrigeration required = 29318 B.T.U. / hour.

Water necessary to evaporate per hour:

Total heat at 0.1217# / sq.in = 1077.1 B.T.U./ #

Heat returning to evaporator = 82.9 B.T.U./ #

Heat absorbed by evaporation of 1# of water =

1077.1 - 82.9 = 994.2 B.T.U.

Therefore 29318 / 994.2 = 29.5# of water evaporated per hr.

Mercury necessary to evaporate per hour:

Assume that mercury returns to the boiler at 200°F.

The heat content of this mercury as returned at 200°F is

(0.0373)(200 - 32) = 6.278 B.T.U./# H = 146.16 B.T.U./#

146.16 - 6.27 = 140 B.T.U./# necessary to evaporate 1# hg.

Theoretical efficiency of apparatus = 21%

Theoretical coefficient of performance = 4.6

29318 / [(0.21)(4.6)(140)] = 217# mercury necessary to evaporate per hour (theoretical). Operating at 32% of the theoretical efficiency:

29318 / [(0.32)(0.21)(4.6)(140)] = 675# necessary to evaporate per hour (real).

Hot water produced per hour:

In the water condenser the water vapor gives up its latent heat in condensing of 1027.9 B.T.U./#. In the mercury condenser the mercury vapor gives up its latent heat in condensing of 122.12 B.T.U./#. It also gives up some of its sensible heat. Assume the temperature of the mercury after condensing to be 400°; then its heat content would be (sp.heat)(t - 32) = (0.0373)(400 - 32) = 13.7 B.T.U. The mercury is returned to the boiler at 200° and its heat content is (0.0373)(200 - 32) = 6.27 B.T.U./#. Therefore the sensible heat given up is 13.7 - 6.27 = 7.43 B.T.U./#

Total heat available for heating:

(1027.9)(29.5) = 30,300 B.T.U./hr. given up by water vapor.

(122.12 + 7.43)(217) = 28,100 B.T.U./hr. given up by hg.

Total = 58,400 B.T.U./hr. (theoretical).

(1027.9)(29.5) = 30,300 B.T.U./hr. given up by water vapor.

(122.12+ 7.43)(675) = 87,500 B.T.U./hr. given up by hg.

Total = 117,800 B.T.U./hr. (actual).

Water is desired at 190°F for heating purposes.

For summer conditions:

Average temperature of water entering condenser is 70°F

58,400 / (190 - 70) = 487#/hr. (theoretical).

117,800 / (190 - 70) = 982#/hr. (actual).

For winter conditions:

The average temperature of the water entering condenser is 40°F.

58,400 / (190 - 40) = 389#/hr. (theoretical).

117,800 / (190 - 40) = 785#/hr. (actual).

Power required to operate the system:

Total heat content of mercury at 15#/sq.in. abs. is 146.16 B.T.U. per #, and the heat content of the mercury returning to boiler is 6.27 B.T.U. per #.

146.16 - 6.27 = 139.89 B.T.U./# supplied by boiler.

Theoretical amount of mercury evaporated per hour is 217#

Actual " " " " " " " 675#

Assuming a boiler efficiency of 70%

(217)(139.89) / 0.70 = 43,300 B.T.U./hr. reqd. (theoretical)

$\frac{(675)(139.89)}{0.70}$ = 134,700 B.T.U./hr. reqd. (actual)

Coal gas has a heat content of about 550 B.T.U./cu. ft. and natural gas a heat content of about 750 B.T.U./cu.ft.

43,300 / 550 = 78.8 cu. ft./hr. coal gas reqd. (theoretical)

134,700 / 550= 245 " " " " " " (actual)

43300 / 750 = 57.8 " " " natural " " (theoretical)

134700 / 750 = 179 " " " " " " (actual)

Chapter VI

CONCLUSIONS

The functions that the mercury ejector refrigerating system will perform in the home may be enumerated as follows:

1. It will cool the entire house during the summer months.
2. A small cold storage room may be operated by coils from the evaporator.
3. It will supply more than a sufficient amount of heat for heating purposes during the winter.
4. A supply of hot water for household purposes will be had throughout the year.

During the winter the heating requirements will determine the rate of operation of the system. Obviously the evaporator would be placed outside the house during the winter, and in the present example approximately 30,000 B.T.U. per hour would be absorbed from the atmosphere and rejected into the house.

During the summer the refrigeration requirements would be the determining factor. In practically all homes the utilization of the large quantity of hot water would be impossible. Of course a portion could be used for household purposes. For best economy the water used for the condensers would have to be recirculated, thus necessitating a cooling tower. It might be practical to use air as a cooling medium for the condensers during the summer, although a condenser so designed to alternate

between air and water as a cooling medium might be difficult to produce.

It is estimated that the initial cost of the system would be very little more than any other type of refrigeration system of the same capacity. The cost of the mercury is the only item which might run the cost above that of the other systems. It is estimated that 150# of mercury would be sufficient to operate the system. It may be noted that this system operates at a low efficiency. It is likely that a much higher efficiency will soon be obtained in the mercury ejector system.

The present homes as constructed are not suitable for refrigerated air conditioning because they are not properly insulated."In the future, refrigerated homes will be well insulated; they will be completely closed and a slight pressure will be built up within to prevent infiltration losses"[1].

It is the opinion of the writer that a more detailed analysis of the problem would be profitable than this brief examination has been able to make.

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1. Professor Wilkinson, Professor Mech. Eng., U. of T.

Chapter VII

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Chapter VIII

APPROVAL

The foregoing thesis is hereby approved as a creditable study of an engineering subject, carried out and presented in a manner sufficiently satisfactory to warrant its acceptance as a prerequisite to the degree for which it has been submitted. It is to be understood that by this approval the undersigned does not necessarily endorse or approve any statement made, opinions expressed, or conclusions drawn therein, but approves the thesis only for the purpose for which it is submitted.

Name F. L. Wiltrout Jr

Title Asst. Prof., Mechanical Engineering.