DOAN
STEAM CONSUMPTION TEST ON AN 9x12

Steam Consumption Test

on

An 8 x 12 Skinner Engine.

- Submitted by -

George Doan and Lloyd Kanters,

as

Thesis Work.

Michigan Agricultural College.

- Spring Term -

1913.

Steam Consumption Test

on

An 8 x 12 Skinner Engine.

- Submitted by -

George Doan and Lloyd Kanters,

as

Thesis Work.

Michigan Agricultural College,

- Spring Term -

1913.

OBJECT:- The object of this thesis is to determine the steam consumption per Brake Horse Power and per Indicated Horse Power under different conditions of back pressure.

Description of Apparatus.

Engine:- The engine used was an 8" x 12" engine manufactured by the Skinner Engine Co. of Erie, Pa. The engine is automatically governed by changing the point of cut off, by means of an inertia governor located in one of the flywheels. The steam ports are 3/4" x 7 3/4" while the exhaust port is 1 3/4" x 7 3/4". The lead for the crank end was 13/64", head end was 3/16".

The engine is automatically lubricated by a splash oiling system.

Steam is taken from the college power house about 400 feet distant. The pressure in the header is kept at approximately 100 lbs. per square inch. The engine is piped so as to exhaust either into a condenser, the heating system of the Engineering Building, or directly into the air.

Dimensions:- Engine.

Diameter of cylinder ---------------------- 8".

Stroke ---------------------------------- 12"

Diameter of piston rod ------ -------------1 7/16".

Diameter of Fly Wheel -------------------- 48".

Diameter of Brake Wheel ------------------ 48".

Cylinder Displacement:-

Head End (H.E.) - 603.12 cu.in. or .349 Cu. ft.

Crank End (C.E.) 583.68 cu.in. or .3375 cu. ft.

Clearance Volume:-

H. E. - 89.09 cu. in. - .0515 cu. ft. equals 14.75 %.

C. E. - 96.59 cu. in. -.0559 cu. ft. equals 16.55 %.

I. H. P. Constants equal K.

H. E. Constant equal $\frac{L\ A}{33000}$ equals $\frac{1 \times 50.265}{33000}$ equals .001527.

C. E. Constant " $\frac{L\ A}{33000}$ equals $\frac{1 \times (50.265 - 1.623)}{33000}$ = .001472.

I. H. P. equals K n P.

B. H. P. equals C.

C equals $\frac{2\ 1}{33000}$ equals $\frac{2 \times \quad \times 4.256}{33000}$ equals .00081.

B. H. P. equals C n W.

The engine is rated at 300 R. P. M.

Condenser:-

The condenser used was a surface condenser manufactured by C. H. Wheeler Company, Philadelphia, Pa.

The pump used in connection with the condenser was a simple cylinder, double acting, steam actuated pump, manufactured by Knowles, Warren, Mass.

The condensing water was pumped from the Red Cedar River by a Duplex pump manufactured by H. R. Warthington, New York, N.Y.

The indicator used was a Crosby manufactured by the Crosby Indicator Company.

A 60# indicator spring was used throughout the series of tests. Both the indicator and spring were calibrated and found to be correct.

The thermometers used were one 200° dairy thermometer and a 400° thermometer. These were calibrated by comparing with a standard, and correction curves plotted as shown on blue print sheets No's 10 and 11.

The steam gauge used was a 200 # Test Gauge, manufactured by Scheffer & Budenberg Company, New York, N. Y. This was calibrated and found to be correct through the range of steam pressures used in the tests.

The vacuum gauge registered from 30" vacuum to 40 # pressure. This was likewise calibrated and found to be correct.

Three platform scales were used, all of which weighed correctly within the range desired.

Two weighing tanks with valve in bottom, were used in connection with the scales, for the purpose of weighing the condensed steam.

METHOD.

Before the actual tests were started the engine was tested for leakage in the following manner. In testing for leaky piston, two methods were used. One, by taking the cylinder head off and turning steam into the opposite end, the actual leakage could be observed. In the other method, the engine was placed on dead center, giving a small amount of lead at that end of the cylinder. The indicator cock at the opposite end of the cylinder was opened and steam turned on. The amount of steam escaping through the indicator cock showed the amount of leakage. The piston leaked badly, due to one of the rings being stuck. The piston was taken out and washed in gasoline which losened the ring. This reduced the leakage by a small amount.

To test the valves, we placed the valve in mid-position and blocked the engine. Steam was then turned on. The amount comping through the indicator cocks showed the leakage. The valves proved to be fairly tight.

The clearance volume at each end of the cylinder, including steam port, drain pipe and indicator pipe was found as follows:

The steam chest cover was taken off and packing was placed of rubber over the steam port and blocked, the valve also being removed, so as to prevent leakage. The engine was placed on dead center and water was poured into the clearance space through the indicator pipe until the same was full and time to fill noted. Then water was poured from another source, just fast enough to keep the clearance space full for the same length of time that it took to fill it first.

The latter amount was taken to be the leakage and was subtracted from the original amount required to fill. The result gave the required lbs of water from which the clearance volume was calculated.

In running the tests, the engine and condenser were started in the usual manner and allowed to run for about ten minutes under the desired conditions for the actual test. The engine was run under six different conditions of back pressure, namely, 5# above atmosphere, atmospheric pressure, 5", 10", 15" and 20" of vacuum.

The different conditions of back pressure were secured by throttling the exhaust before it entered the condenser. The throttling was accomplished by means of a gate valve in the exhaust pipe leading from engine to condenser.

Under the first three conditions of back pressure, loads of 50, 100, 150, and 200 lbs were placed on the scales. Under the last three conditions loads of 50, 100, 150, 200, and 250 lbs were used. Using the 250 lbs load, it was found impossible to obtain a 20" vacuum 18" being the highest obtainable.

Using the 50 and 100 # loads each test was run for one hour. For the other loads the tests were 40 minutes in length From 5 to 10 minutes were left between tests having different brake loads. All the tests using the same back pressure were run consecutively. Indicator cards were taken at the beginning middle and end of each test. Readings were taken as foll-ows, brake load, time, steam pressure, back pressure, R. P. M., temperature of condensed steam, temperature in calorimeter, barometric pressure.

The quality of the steam used, was found by means of a throttling calorimeter.

Knowing the total amount of steam condensed for any one test and the quality, the amount of dry steam per B. H. P., and I. H. P. could be calculated. The results are found tabulated on sheets 12 and 13 of blueprints.

The following curves were plotted.

B. H. P. lbs of dry cond. steam / B. H. P. / hour.

I. H. P. lbs of dry cond. steam / I. H. P. / hour.

Vacuum " " " " " / B. H. P. / " .

" " " " " " / I. H. P. / " .

" " " " " " per hour.

" Rankine Eff.

Sample indicator cards are shown on blue prints.

Saturation curves are shown plotted on blueprints 25 - 30 inclusive. They are plotted as follows:-

First an indicator card was drawn.

The length of the card was taken as representing the displacement of piston. Then the line representing clearance was drawn to the same scale. The lines of 0 lbs pressure absolute was also drawn. (See blueprint 25 for example). At any point as "D" on the compression curve drop a perpendicular to the O# pressure line and also to the clearance line. From these perpendiculars the volume of steam and pressure can be scaled, and from a steam table the weight of this volume of steam can be obtained. Then the cylinder feed is obtained by dividing the total lbs of steam condensed per hour by the number of strokes per hour. The sum of the cylinder feed and cushion steam which has been obtained gives the total amount of steam in the cylinder. From steam table, the weight of steam in the cylinder by the weight gives of 1 cu ft. at any given pressure gives the volume of steam in the cylinder at that pressure as " A B". Using this method the curve S S is plotted.

which is the saturation curve and would represent the expansion of the steam in the cylinder providing no condensation took place.

CONCLUSION.

From a study of the curves, we would conclude that the steam consumption decreases with an increase of vacuum.

The steam per B. H. P. and per I. H. P. per hour, drops up to the rated load after which it increases slightly, the rated load being about 40 H.P.

The clearance volume is higher than is usually found in high speed engines, Kent giving from 8 - 12 % while this engine has from 15 - 16 %.

From a study of the saturation curves, we were able to determine the quality of the steam in the cylinder at any point in its expansion (i.e.) at any pressure.

By drawing the atmosphere line on each card, we found that a great deal of condensation took place before the steam entered the cylinder, probably due to valves and turns. This was especially noticable on the light loads. For the heavy loads, the condensation was very little as is shown by the drawings. On many of the light loads the saturation curve touched or even intersected the expansion line at release. Thus showing 100 % dry when it touched or superheated when it intersected. For the heavy loads was not so noticable.

The quality at an 80 # absolute pressure and 200 # load was found from the sample cards for the various back pressures. On account of the irregularity of the cards, no definite diffence could be observed. However the quality seemed to be high-

er for higher vacuum.

Thus for H. E. 200 # load 80# pressure the quality was as follows:-

5 #	-	76.2 % quality.	
0 #	-	73. %	" .
5 #	-	75.4 %	" .
10"	-	78.8 %	" .
15"	-	76.0 %	" .
20"	-	79.6 %	" .

and for H.E. 200 # load 60 # pressure the quality was:-

5 #	-	75.8 % Quality.	
0 #	-	69.5 %	" .
5"	-	74.8 %	" .
10"	-	82.8 %	" .
15"	-	74.1 %	" .
20"	-	80.3 %	" .

Also comparing a light load and a heavy load with the same back pressure, we found that the quality was higher for the heavy load as shown by the following figures.

Load	Back Pressure	% Quality. H.E. 60# Pressure.	
200#	5#	75.8	" .
"	0#	69.5	"
"	5#	74.8	" .
"	10"	82.8	" .
"	15"	74.1	" .
"	20"	80.3	" .
50#		72.	" .
50#		61.	" .

Load.	Back Pressure	% Quality.
50 #	5"	57.5 " .
100#	10"	76. " .
100#	15"	72. " .
100#	20"	64. " .

CALCULATIONS.

For 50 # load and 0 # back pressure.

Symbols used.

N equals- R. P. M.

n equals- number of strokes.

W equals-weight on scale beam.

w equals-unbalanced weight of brake.

A equals-are a piston in square inches.

I.H.P. equals-Indicated Horse Power.

B.H.P. equals-Constant.

K equals-I.H.P. constant.

C equals-B.H.P. constant.

M.E.P. equals - Mean Effective Pressure.

H.E. equals - Head End.

C.E. equals - Crank End.

x equals - Quality of steam.

H equals - total heat corresponding to calorimeter temperature.

C_p equals - S ecific Pressure = .48

T_s equals - Temperature in Calorimeter.

T_2 equals - " " " corresponding to pressure.

q equals - Heat of Liquid at throttle pressure.

T - Heat of liquid at throttle pressure.

e_r- Rankine Effeciency.

rg- Latent heat at exhaust pressure.

x_c- Quality of steam at release.

q_1^1- Heat of liquid at throttle pressure.

q_2^1- Heat of liquid at exhaust pressure.

r - Latent heat at inlet pressure.

x_b- Quality of steam at inlet.pressure.

1. I.H.P. equals K x M.E.P. x n.

I.H.P. (H.E.) equals .001527 x 8.5 x 298.6 equals 3.88.

I.H.P. (C.E.) equals .001472 x 9.58 x 298.6 " 4.21.

Total I.H.P. equals 8.09.

2. B.H.P. equals C x N x (W - w)

B.H.P. " .00081 x 298.6 x (50 - 27.2) equals

.00081 x 298.6 x 22.8 equals 5.51.

3. Quality of steam equals x.

$$x_1 \text{ equals } \frac{H \text{ plus } C_p (T_s - T_2) - q}{r_1}$$

$$\text{equals } \frac{1149.96 \text{ plus } .48 \ (271.4 - 211.) - 306.9}{881.4}$$

$$\text{equals } \frac{872.06}{881.4} \text{ equals } 99.\%.$$

4. Lbs of steam used per hour, equals 428 #.

5. Lbs of dry steam used per hour,
equals .99 x 428 equals 423.7 #.

6. Lbs of dry steam per I.H.P. / hour. equals,

$\frac{423.7}{8.09}$ equals 52.2 #.

7. Lbs of dry steam / B.H.P. / hour equals,

$\frac{423.7}{5.51}$ equals 76.8 #.

8. Rankine Efficiency.

(C_r for 50 # load at 5 # back pressure.

$$C_r = 100 - \left(\frac{961.8 \times .88}{306.9 - 194 \text{ plus } (881.4 \times .99} \right) 100 \text{ equals}$$

$$100 - \left(\frac{851}{984.9} \right) 100$$

100 - 86.4 equals 13.6 %.

9. Clearance Volume.

(H.E.)

Cyl Displacement equals $8^2 \times .7854 \times 12$ equals 603.12. cu. in or .349 cu. ft.

Water to fill head end equals 3.175 plus .04 equals 3.215# (temp. of water used 65.5° F.)

Water at 65.5° F weighs 62.3575 # / cu. ft.

$\frac{3.215}{62.3575}$ equals .0515 cu. ft or 89.09 cu. in.

$\frac{.0515}{.349}$ equals 14.75 %.

(C.E.)

Weight of water to fill clearance space, equals,

3.45 # plus .04 # equals 3.49 #.

$\frac{3.49}{62.3575}$ equals .0559 cu ft. or 96.59 cu. in.

C. E. cyl. displace ment 603.12.

($(1\ 7/16)^2 \times .7854 \times 12$) equals 583.68 cu. in. or .3375 cu. ft.

$\frac{.0559}{.3375}$ equals 16.55 % Clearance.

Total volume H. E. equals .349 plus .0515 equals .4005 cu ft
C. E. equals .3375 plus .0559 equals .3935 "

REFERENCES.

"Engine Tests" by Barrus.

"Transactions of A. A. M. E. of 1902."

"Kent".

"Thermodynamics" by Goodenough.