HEAT TRANSFER IN

STICKWATER CONCENTRATORS

by

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I N D E X

S U M M A R Y.

Multi-effect, natural circulation evaporators are widely used for concentrating weak liquors during the production of commodities such as sugar, tomato paste, meat extract, and more recently fish meal. In the latter case the solids content of certain fish juices (termed stickwater) has to be increased from roughly 8% to 50%, as a prerequisite to their commercial utilization.

Of all problems connected with the design and operation of conventional stickwater concentrators, that of low heat transfer appears to have received least attention. A study of the subject appeared particularly promising, in view of the scarcity of relevant information in respect of similar equipment used by other processing industries.

A laboratory-sized replica of a typical industrial unit was constructed, and equipped for the measurement of all relevant processing variables. Using commercial stickwater at practical ranges of temperatures and solids concentrations it was found that the heat transfer rates in the first, second and third effects were roughly in the ratios of 4:2:1. Even with clean tubes, the overall U value in the third effect occasionally dropped below 50 BTU/sq.ft. °F hr. Data on the physical properties of stickwater and its concentrate is presented which suggests that the low heat transfer rates are associated with excessively high product viscosities.

Using the same apparatus, five alternative methods for improving the heat transfer in existing plant were tested. It was found that :

(a) Reverse - instead of forward - feeding was only moderately promising.

(b) Reducing the viscosity of the product by enzyme treatment resulted in a rise of evaporative capacity by from 3% to 37%.

(c) Forced circulation caused the heat transfer rates to rise, provided the circulation was accelerated beyond about 6 ft. per second.

(d) Acceleration of the natural circulation rates by injection from 2.0 to 4.0 litres of air per minute into the lower end of the third effect tube caused the capacity of the plant as a whole to rise by from 17% to 47%.

(e) Throttling the natural circulation by means of a metering pump caused an improvement in heat transfer in an individual effect of from 20% to 84%, depending on the solids concentration of the working fluid.

To account for some of the seemingly inconsistent results of these tests a theory is proposed which quantitatively applies the concept of the establishment of distinct boiling and non-boiling zones for liquids flowing upwards in vertical heated tubes. The analysis is extended to include the case when the working fluid is so viscous that (in the absence of vapour generation) laminar flow must be expected to prevail in the non-boiling section of the tube.

Numerical examples are provided to illustrate how the change from turbulent to laminar flow in the non-boiling tube section profoundly affects the heat transfer mechanism, and may lead to a reversal of the trends associated with variables such as viscosity and circulation velocity.

The experimental results are re-examined in the light of this theory, and reasonable explanations are advanced for the observed phenomena.

The work is concluded with a critical examination of the role of heat transfer within the economic structure of stickwater utilization.

CHAPTER I

INTRODUCTION

The increasing importance of fish as a source of nourishment for the world's population is amply documented (1). The total fish catch by the major fishing nations has increased from 20 million tons in 1938 to 55 million tons in 1961.

Up to about 50 years ago the bulk of the catch was directly utilised for human consumption. The balance, in the form of processing offal was generally discarded, or used as manure. In recent years it has been recognised that practically any kind of fish or fish offal can be turned into fish meal, which in its crude form is a valuable source of protein in many animal rations.

The demand for fish meal has increased as rapidly as the growth of the Fishing Industry as a whole; while only 656,000 tons were produced in 1938, the figure for 1960 had risen to 1,669,000 tons.[1]

The latest development in the utilisation of fish meal has been the perfection of a process (2) to refine the crude product and render it fit for human consumption directly, instead of via the animal organism. Serious efforts are presently being made on an international level to make adequate supplies of this so-called "fish flour" available to millions of the world's population who suffer from a chronic dietary protein deficiency. If these efforts succeed, it is estimated that the demand for fish meal will further increase by several millions of tons per annum.

The spectacular upsurge of fish meal consumption is expected to continue as long as its price remains competitive with other - e.g. vegetable - sources of similar nutrients.

Fish meal production has greatly benefited from mechanisation and technological progress. The flow sheet for a typical modern fish reduction plant is shown in Fig. 1. The approximate composition of the terminal and intermediate products is illustrated by means of Fig. 2.

The Whole Fish Storage must be adequate to balance fish landings against plant capacity. If landings are regular and plant capacity oversized (as with the menhaden industry in the U.S.A.) holding tanks may virtually be dispensed with. If landings are intermittent and plant capacity limited (as with the Norwegian winter herring industry) each factory may have to store up to 6,000 tons of raw fish at a time.[2]

The Steam Cookers were originally screw conveyors fitted with jets for injecting live steam into the moving mass of raw fish. But for/

(1) 2,036,000 tons in 1961.

(2) In South Africa the capacity of the fish pits allows for approximately 48 hours production.

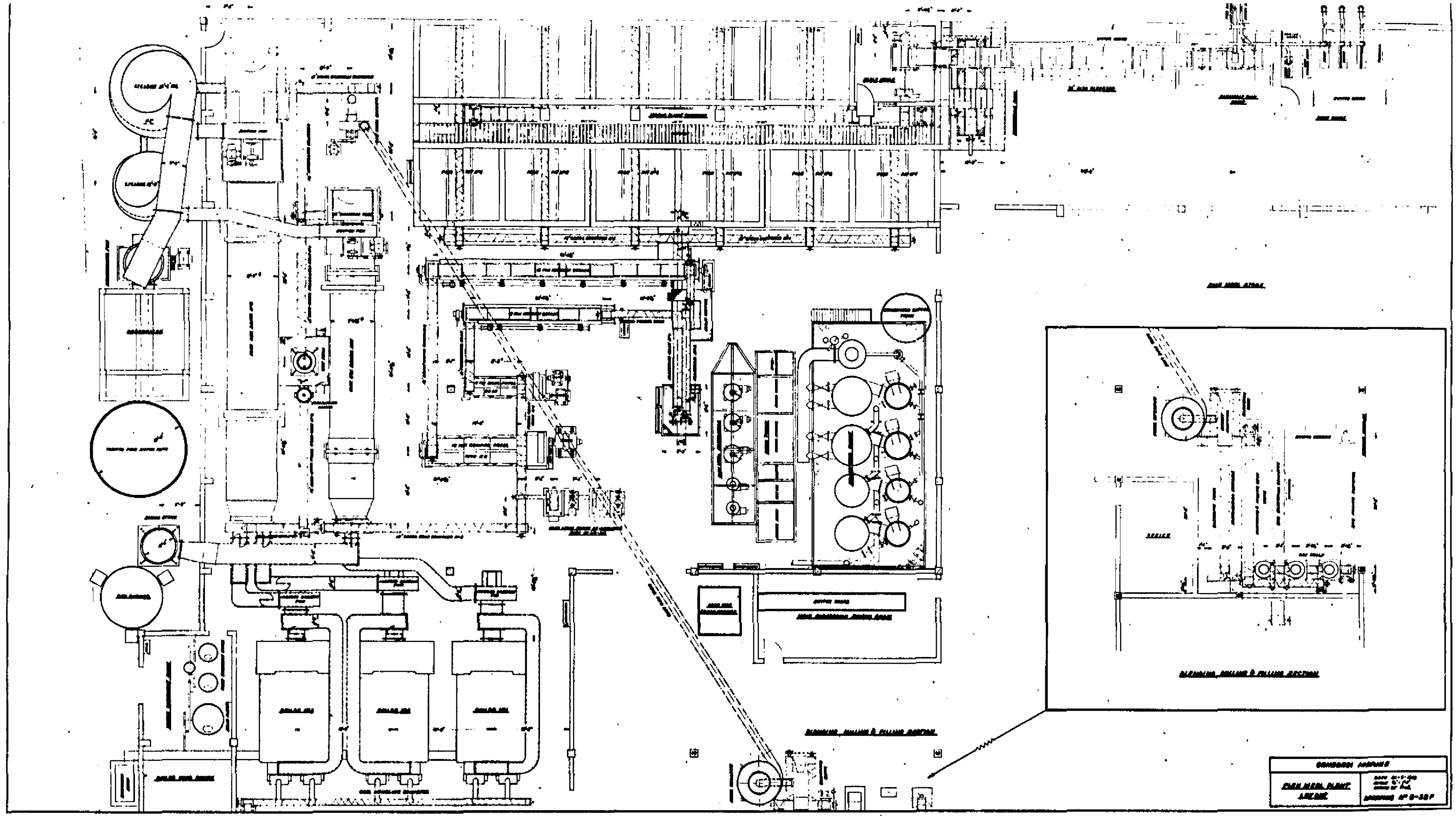

FIG I MODERN FISH REDUCTION PLANT

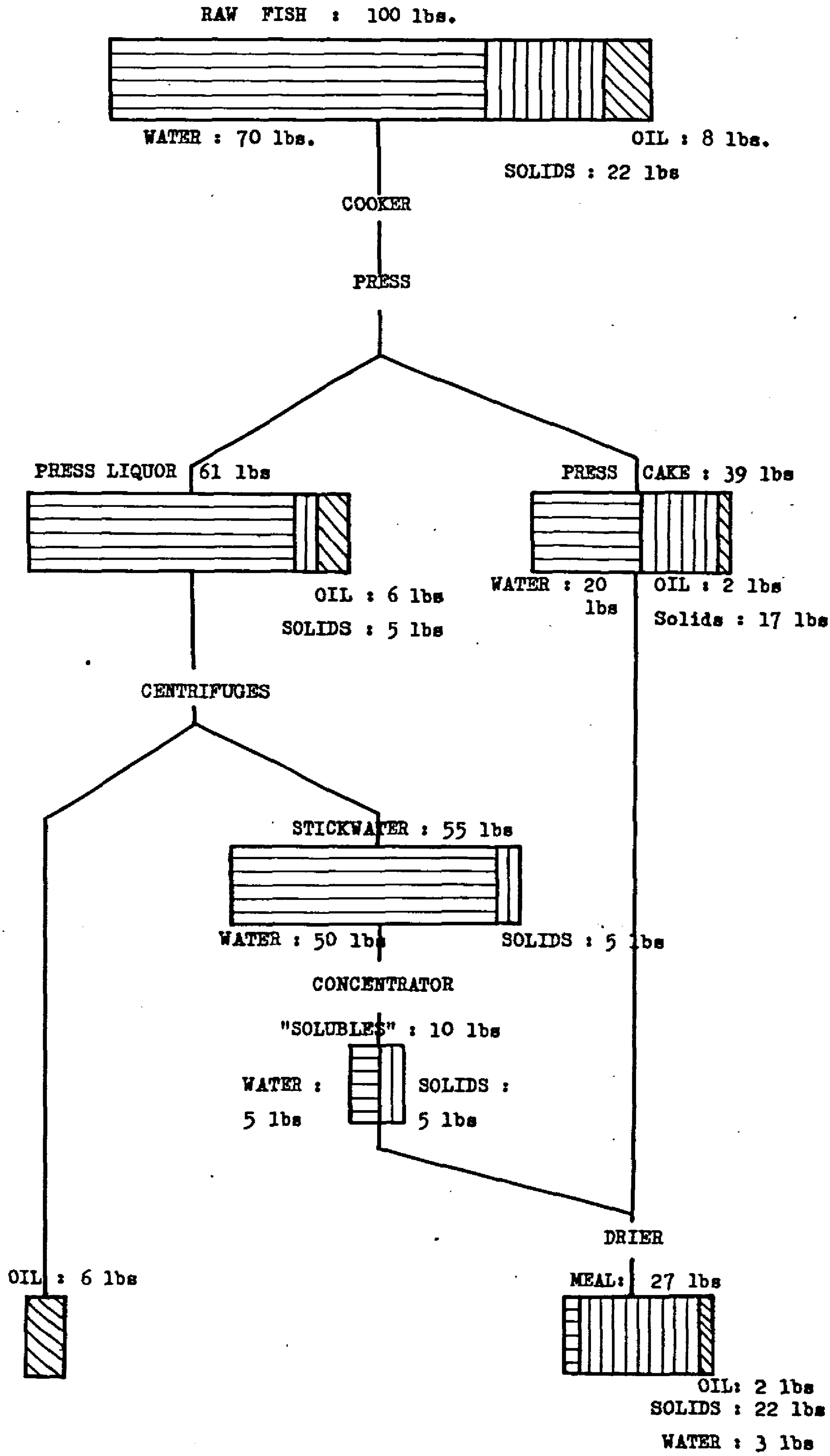

FIG. 2 : MATERIAL BALANCE OF FISH MEAL PRODUCTION BASED ON AVERAGE VALUES

for stickwater recovery the live steam condensate imposes an additional, unnecessary burden on the concentrator, and jacketed cookers with hollow, heated flights are now standard.

The Screw Presses serve to separate the solid and the liquid phases of the cooked fish. The primary purpose of pressing is the recovery of the body oil, which in the case of some species, such as herring and pilchard, constitutes a major source of revenue. (When handling species of low oil content, commonly referred to as "White fish", pressing is generally dispensed with).

The products leaving the presses are known as presscake and press-liquor respectively. The former - the "solid" phase - still contains appreciable quantities of moisture and oil, while the latter - the "liquid" phase - contains the bulk of the free oil originally present in the raw fish, as well as a fair percentage of suspended and dissolved proteinaceous matter. (See Fig. 2).

The Driers reduce the moisture content of the presscake to a level where it can safely be stored without danger of bacterial degradation. Continuous rotary driers are standard, with the heat supplied either by steam generated in a separate boiler plant, or by integral oil burners, or by the flue gases from the boiler plant which generates the steam required for cooking and stickwater concentration.

The Centrifuges recover the free oil in the press-liquor. The oil-free underflow from the centrifuges is known as Stickwater, and was formerly run to waste, as the value of its dissolved and suspended constituents was not appreciated (and water pollution was not yet a problem). Research has shown, however, that by using multi-effect evaporators the concentration of the stickwater solids could be achieved at a profit. As a result, stickwater recovery has become an integral part of the fish meal process, particularly when processing oily fish.

The Stickwater Plant generally takes the form of a multi-effect, natural circulation evaporator, capable of raising the solids content of the product to between 50 and 60 per cent. The stickwater concentrate is usually mixed with the presscake just before the latter enters the drier. The fish meal thus produced is termed "full" meal, as distinct from "ordinary" meal produced without the addition of stickwater concentrate. The concentrate may also be marketed as a separate product called "solubles", for which purpose it is either drum dried to a moisture content of less than 2.5%, or chemically preserved and sold as a liquid.

Stickwater recovery has taken the Industry a big step forward. The average yield of fish meal per ton of raw fish has been increased

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by 25%; a conspicuous source of harbour water pollution has been eliminated, and profits have benefited handsomely.

Stickwater concentration is, however, not without its problems; since the cost of the stickwater plant may easily amount to $12\frac{1}{2}$% of the cost of the fish meal plant as a whole, the Industry is keenly aware of the economic aspect. The perfection of suitable equipment has, however, been hampered by a lack of information to guide both the manufacturers and the operators.

In 1959 the Technical Committee of the Fishing Industry[1] initiated an investigation into the exasperating habit of maasbanker[2] concentrate to choke the tubes in the final effects of the evaporators solidly without warning to the operator.

The reason for this was naturally sought in the high viscosity of the concentrate, (see Appendix I), and early experiments were designed to render this material more fluid, either by reverse feeding or by the action of proteolytic enzymes.

It soon became apparent that scaled-down laboratory tests were not a reliable means of estimating the effectiveness of either enzyme treatment or reverse feeding. It seemed that the phenomenon of tube blockage was critically affected by the physical dimensions and operating conditions of the evaporator concerned.

Authority was therefore given for the construction of a replica of a typical triple effect commercial concentrator, comprising a single tube of standard dimensions per calandria.

The earliest tests with this apparatus, when operating under conditions of temperature and product concentration as apply in practice, revealed surprisingly low rates of heat transfer, particularly in the final effect. A few checks confirmed that these experimental heat transfer coefficients were indeed representative of those occurring in commercial plants.

The centre of interest thereupon shifted from the search for a solution to a specific, and not very common, operating problem to the study of heat transfer in stickwater concentrators generally.

The purpose of this study was :

a) the estimation of heat transfer coefficients in commercial stickwater concentrators for a fully representative range of operating conditions;

b) the identification of the factors which are responsible for the abnormally low heat transfer rates prevailing in the later effects; and

c) the evaluation of the effectiveness and profitability of certain measures designed to improve the heat transfer rates in existing plant.

The results of the tests which were carried out in fulfilment

(1) An advisory organ of the Fishing Industry Research Institute, University of Cape Town, Rondebosch. South Africa.

(2) TRACHURUS TRACHURUS.

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of this task could not be interpreted without an understanding of the principles underlying the functioning of natural circulation evaporators. A thorough search of the literature revealed a remarkable paucity of basic information regarding the performance of this type of plant, particularly when applied to the concentration of viscous fluids. Although the division of the heated tube into a boiling and a non-boiling section had been postulated and confirmed experimentally, no evidence was found of an attempt to formulate precisely the implications of this concept. Neither was mention found of a distinction between laminar and turbulent flow conditions in the non-boiling tube section.

In order to find a satisfactory explanation for some of the apparently contradictory results of the experiments with stickwater, some of the tests were repeated with working fluids with more clearly defined ("Newtonian") physical properties. Finally, a theory was proposed to account for the apparently inconsistent changes in heat transfer coefficients when the viscosity of the working fluid exceeds a certain limit.

In so far as the latter part of this investigation has been divorced from the physical properties of a particular working fluid, it is possible that the results may be profitably applied to processes which require the concentration of liquors other than stickwater.

CHAPTER II

THE FUNCTION AND CHOICE OF STICKWATER PLANT

The purpose of stickwater recovery is two-fold; it is intended to increase the profits earned from the processing of fish to meal and oil, and to reduce the pollution of the riparian and coastal waters by the factory effluents. If the factory is situated in a well populated area, and particularly if the polluted waters tend to be stagnant, the Public Health authorities will generally not permit the return of stickwater to the sea under any circumstances.

Irrespective of the motive for stickwater recovery, sound business sense demands that the plant chosen for the purpose shall perform its function in the most economical manner. In other words, it must yield a product of maximum market value at minimum production cost.

The saleable constituents in raw stickwater total only about six to twelve per cent of its weight. Regardless of the form in which they are ultimately marketed (as Solubles, as roller-dried powder, or incorporated with the ~~rest of the~~ meal), the first step in their recovery calls for the evaporation of vast amounts of water. For instance, referring to Fig. 2, it will be seen that for a plant of say twenty tons raw fish capacity per hour to concentrate its stickwater to a 50% solids content, requires about nine tons of water to be evaporated per hour. (Note: methods other than evaporation to concentrate the key constituents of stickwater are as yet of only academic interest).

Evaporators are amongst the most widely used units of process engineering, and many excellent text books have been devoted to their design. But, as with other specific commodities, the concentration of stickwater is governed by practical as well as theoretical considerations.

Among the physical, chemical and nutritional properties of the product which restrict the choice of an evaporator for stickwater processing, the most important are:

a) the heat sensitivity of its constituent amino acids, vitamins and other - as yet unidentified - "growth factors". The impairment of the nutritive value of stickwater by heat is a complex subject. It has most clearly been demonstrated in the case of Vitamin B_{12}. The Norwegian Herring Industry Research Institute has reported for instance (3) that by maintaining herring stickwater at 150°C for two hours its Vitamin B_{12} content was reduced from 0.75 γ/g. dry matter in the unheated control to 0.50 γ/g. dry matter, whereas at 100°C it was only reduced to 0.66γ/g. dry matter. Evidence of this nature, coupled with the lingering uncertainty regarding the degree of heat sensitivity of the other stickwater constituents, have caused the Industry to favour plant operating at temperatures not exceeding about 280°F in the first, and 150°F in the final effects;

b) the remarkably rapid rise of the apparent viscosity of the product with solids concentrations (described in detail in Appendix I). This calls for conservative estimates of heat transfer coefficients, and practically rules out forced circulation as an aid to heat transfer;[1]

c) the corrosiveness of the liquid and its distilled vapours. This may require special materials of construction, or chemical treatment of the corrosive media;

d) the fouling of heating surfaces by organic as well as inorganic matter (loosely termed "scale"). All heating surfaces must, therefore, be accessible, and must not require elaborate cleaning methods. A construction that permits cleaning of individual heating units in rotation so as not to interrupt production, is also highly desirable.

Apart from these fundamental limitations imposed by the properties of the product, a number of pertinent practical considerations also affect the choice of plant.

When an existing factory decides to recover its stickwater, it must face not only the cost of the evaporator, but also the expense of integrating it with the rest of the plant. This may involve the replacement of direct by indirect cookers, the extension of the boiler plant and the installation of larger driers.

In view of the diversity of factors which had to be taken into account, it is not surprising that the early days of stickwater recovery were notable for the large number of widely differing processes offered, and the - sometimes extravagant - claims that were made for them.

This is illustrated by two diametrically opposed systems which were developed and tested during this period. They are :

a) the Sharples-Lassen process. This involved the acidification of the stickwater to a pH of 4.5 and its subsequent clarification by means of centrifuging. The clarified liquor was then concentrated in a multi-effect evaporator of acid-proof construction. The concentrate thus produced was sufficiently stable to be marketed directly, without further preservative

treatment/

[1] For economic reasons as discussed on Page 56.

treatment, or it could be neutralised and added to the press-cake to produce full meal. This process achieved a tolerably low operating cost - by virtue of employing multi-effect evaporation - but the capital cost of the equipment, notably the special centrifuges, was high;

b) the Lysosund process. In this process the unconcentrated raw stickwater was mixed with a portion of the fish meal, which was then recirculated through the drier. The only plant modifications required were an enlarged drying capacity, which called for the logical minimum of capital expenditure. As, however, the evaporation of a pound of water in a rotary drier requires the equivalent of about 1.5 pounds of steam, whereas a vacuum concentrator comprising say two effects requires only about 0.65 pounds, the operating cost of this process was so high that it could not be considered except in the presence of abnormally low fuel prices. The comparison is even more unfavourable in the case of a triple effect plant, which achieves a steam economy of roughly 0.4 pounds per pound evaporation.

In South Africa a single factory on the West Coast installed a Sharples-Lassen plant in 1950. As this factory is situated on a river estuary, the decision to recover its stickwater was prompted more by the necessity to prevent water pollution than by the profit motive. The Sharples-Lassen process was chosen because a market for acidified solubles already existed in the U. S. A., whereas the production of full meal was still clouded by uncertainty regarding its stability, nutritive value and production cost.

In 1953, when the profitability of stickwater recovery had already been established overseas, a sub-committee of the S. A. Food Canners' Council was formed, comprising the technical leaders of the Fishing Industry, headed by the Director of the Fishing Industry Research Institute. This committee was to study and to report on the merits of different types of stickwater plant which were on offer at the time, and on the desirability of bringing one or more of these pilot installations to South Africa.

It was through the guidance provided by this committee that the Industry was largely spared the expense of experimentation with plant and processes which subsequently proved uneconomical or inherently unsuitable. Among other things, a draft specification was drawn up which contained a capacity clause stating that the rated evaporation had to be maintained for six days continuously without cleaning, and that the total solids content of the product was to be maintained at 50% or better.

The/

The Fishing Industry Research Institute issued its Progress Report No. 15 entitled: "The adaptation of existing fish reduction plants for the manufacture of whole meal", which critically analysed the advantages and disadvantages of various combinations of evaporators and modifications of existing fish meal plants.

The report rejects outright the Lysosund process on account of its high fuel cost. It also warns of the possible heat damage to the product in the case of one type of plant employing high temperatures (284°F 140°C in the first and 262°F 128°C in the final effect), although the re-use of the high pressure vapours in the fish cookers and driers renders this combination economically attractive.

While the report does not single out a particular plant as being the most suitable, it subscribes in principle to the choice of double or triple effect, natural circulation vacuum concentrators. It so happens that this has become the pattern for modern stickwater concentrators not only in South Africa but throughout the world.

As an example, the following specification for a stickwater concentrating plant of a type which is currently in favour in South Africa was kindly supplied by Messrs. Rock Engineering Works (Pty.) Ltd. of Parow, Cape Province:

<u>Type:</u> Four separate effects (or "stages"), each comprising two calandria, are connected to individual vertical separating drums. The calandria are vertical and fitted with loose top and bottom covers for easy cleaning. The four stages are so interlinked that any three can be operated together as a triple effect evaporator, thus releasing the fourth stage for cleaning.

<u>Capacity:</u> When operated as a triple effect unit: 38,000 pounds 17 236.5 kg evaporation per hour, assuming a feed temperature of 176°F. 80°C

<u>Feed Method:</u> Forward feed, i.e. product and distillate pass through successive effects in the same order.

<u>Steam Supply:</u> The first effect receives live steam at 35 241kPa to 60 413kPa psi gauge pressure. All later effects are heated by distillate from the previous effects.

Steam Con-<u>sumption</u>: 16,000 pounds per hour. Hence the specific consumption equals $\frac{16,000}{38,000}$ = 0.42 pounds of steam per pound of evaporation. 0.42 kg

<u>Temperatures:</u>

	Calandria	"Separator" (i.e. liquid receiver)
First effect	298°F 147.8°C	271°F 132.8°C
Second effect	271°F 132.8°C	227°F 108.3°C
Third effect	227°F 108.3°C	115°F 46.1°C

Number/

Number of Tubes	:	230 per calandria, i.e. 460 per effect
Tube Dimensions	:	Length: 10 ft. 2 ins.; Diameter: 35 mm I.D., 38 mm O.D.
Material of Heating Tubes	:	Grade 316 Stainless Steel.
Heating Surface	:	1,750 sq. ft. per effect
Concentrate Removal	:	By "Mono" Pump, rated at 1,900 gals. per hour against a head of 150 feet, requiring a 10 H.P. motor.
Vacuum Pump	:	Sulzer, rated at 110 cfm against 27 ins. Vacuum, requiring a 15 H.P. motor. 11.19 kW
Cooling Water Pump	:	SL 1 Pump, rated at 56,000 gals. per hour against a head of 45 feet, requiring a 20 H.P. motor.
Condensor	:	Barometric spray type
Instruments	:	Indicating instruments are supplied for calandria pressure, drum pressure (or vacuum) and drum temperature for each effect.
Fabrication	:	All vessels working under pressure comply with the British Standard Specification for pressure vessels.
Cost	:	(subject to minor adjustments). R58,000 ex Works.
Guarantee	:	The plant is guaranteed against defects in materials and workmanship in the usual manner.

An assembly drawing of the plant is shown in Fig. 3.

A remarkable feature about most specifications for stickwater plant - including the one above - is that no reference is made to the product, and its final solids concentration in particular is not specified. This is indicative of the uncertainty surrounding the physical properties of stickwater concentrate, and its behaviour in evaporators.

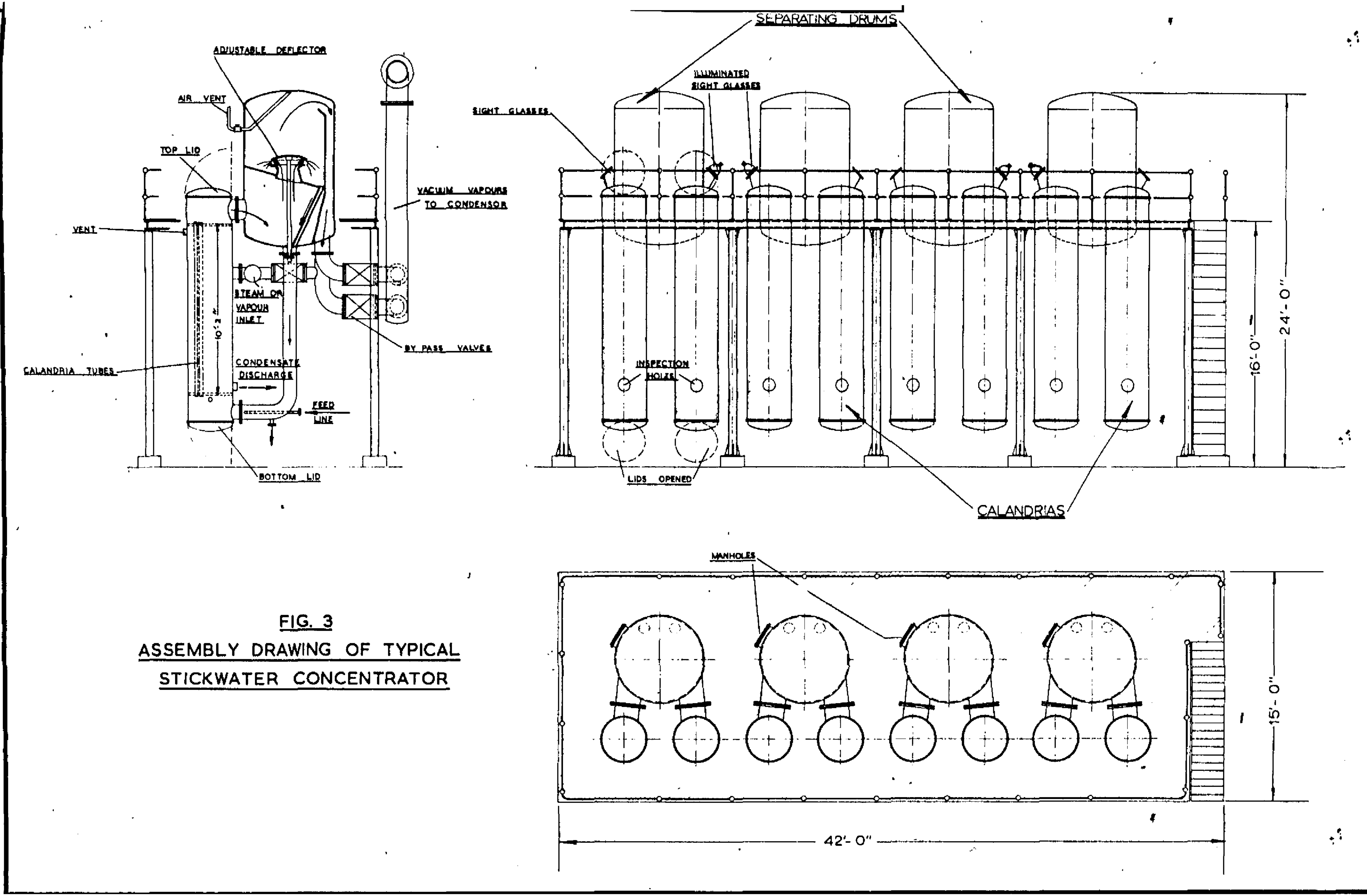

FIG. 3

ASSEMBLY DRAWING OF TYPICAL STICKWATER CONCENTRATOR

CHAPTER III

THE PROBLEMS OF STICKWATER CONCENTRATION

The problems to be dealt with in this chapter concern plant construction and operation. The economics of stickwater recovery are dealt with in Chapter IX.

Until recently the most disturbing problem of stickwater concentration was corrosion. The average life of the body of a concentrator constructed in mild steel is six to seven years. The average life of mild steel tubes ($2\frac{1}{2}$ mm wall thickness) is only three to four years; i.e. one shell will generally outlast two sets of tubes.

The corrosiveness of stickwater, stickwater concentrate and its distilled vapours has been thoroughly investigated and reported on by FIRI (4). It appears that the tubes corrode not only on their insides (particularly at their inlets and outlets), but even more severely on their outsides. Corrosion seems to be worst near the lower ends, and on the steam side, of the tubes in the calandria following the first effect.

Stickwater vapour and stickwater distillate are the chief corrosive agents. It was found that while the pH of distillate at room temperature is 9.0, it drops to about 6.8 at 175°F, and even lower at higher temperatures. The attack by the distillate is most severe in the presence of oxygen, traces of which are present in the vapours at all stages of stickwater concentration.

In a typical test the weight loss over 96 hours of a strip of annealed mild steel with a surface of 3 sq. ins., immersed in distillate at 175°F, amounted to110 mg under aerobic, and 65 mg under anaerobic conditions.

A straightforward solution to the corrosion problem is the use of stainless steel tubes (and possibly a stainless steel body, although the cost of such a plant may render it uneconomic). Stainless steel tubes are widely used in South Africa, but their cost remains high. The price of a plant equipped with AISI 316 stainless steel tubes is generally 60% above that of a similar plant with mild steel tubes.

Less expensive grades of stainless steel are available, and tests are planned to determine their corrosion resistance as compared with grade 316. But the maximum saving that could thus be achieved would hardly be expected to exceed 10% of the installed plant cost.

Another promising possibility is the chemical treatment of the corrosive liquids and vapours in order to neutralise them. This has also been investigated by FIRI (5), and the recommended treatment consists of injecting a 10% solution of caustic soda into the vapour space of the entrainment

separator/

separator of each calandria to maintain a pH of 9.0 at all operating temperatures. Scrubbing of the vapours rather than direct addition of caustic soda has certain theoretical advantages, but might be more difficult to apply in practise. Injection of NaOH also protects the vapour passages in addition to the distillate and condensate lines.

This treatment is being successfully applied by several South African factories, but it is not yet certain that its greater economy entirely outweighs all the advantages of stainless steel.

Chemical treatment of the corrosive media, and the use of stainless steel have reduced the urgency of the corrosion problem, but no satisfactory solution has yet been found for another of stickwater's distressing properties - its proneness to scale formation.

The opportunity was taken during the tests reported in Chapters V and VI to estimate the rate of scale formation under normal operating conditions. In altogether twelve cases the scale removed after runs of known duration was dried and weighed. The thickness of the layer of scale deposited per sq. ft. during 24 hours was estimated by assuming a specific gravity of 2 (a reasonable average for most scale-forming salts). A summary of these figures is shown in Table I.

From these data the maximum, minimum and average rates of scale deposition for each of the three effects has been calculated as follows:

	Millimetres scale deposited during 24 hours		
	1st Effect	2nd Effect	3rd Effect
Maximum	0.23	0.12	0.08
Minimum	0.05	0.02	Negligible
Average	0.1	0.06	0.02

Assuming a thermal conductivity of this scale of 1.3 BTU ft./sq.ft. °F. hr. (6), and with a typical overall coefficient for a first effect starting with clean tubes of 250 BTU/sq.ft. °F. hr., the average reduction during the first 24 hours of normal operation will be given by

$$\frac{1}{U_{(24\ hrs.)}} = \frac{1}{250} + \frac{1}{\frac{25.4 \times 12 \times 1.3}{0.1}} \quad (1)$$

From this $U_{(24\ hrs.)}$ = 235 BTU/sq.ft. °F. hr., i.e. a reduction of 6 per cent. Whilst this may not appear serious, calculation shows that after an uninterrupted run of say five days (as frequently occurs in practice), the overall U value will have been reduced to 190 BTU/sq.ft. °F. hr., which is 24% lower than for clean tubes.

An interesting finding, and one that confirms reports from the factories, is that most scale is formed in the first effect. This is probably due to a higher liquid temperature, which coupled with an inverted solubility curve for the major scale components causes their more rapid

deposition/

(1) Assuming that scaling proceeds at a uniform rate.

T A B L E I

RECORD OF SCALE REMOVED FROM INDIVIDUAL EFFECTS AFTER RUNS OF KNOWN DURATION

Run No	DRY WEIGHT OF SCALE - grams -			TOTAL WEIGHT grams	OPERATING TIME hours (approx)	DRY WEIGHT SCALE PER SQ. FT. PER 24 HOURS OPERATION - grams -				CALCULATED SCALE THICKNESS millimetres/24 hours.		
	1st Effect	2nd Effect	3rd Effect			1st Effect	2nd Effect	3rd Effect	Total	1st Effect	2nd Effect	3rd Effect
1, 2 & 3	29.7	6.3	neg.	36.0	12	22.0	4.7	neg.	26.7	0.12	0.025	neg.
4 & 5	10.1	5.2	0.9	16.2	8	11.3	5.9	1.5	18.7	0.06	0.03	0.008
6 & 7	13.6	6.4	3.4	23.4	8	15.0	7.1	3.8	25.9	0.08	0.038	0.02
8 & 9	30.1	24.4	neg.	54.5	10	26.8	21.7	neg.	48.5	0.142	0.115	neg.
10, 11 & 12	28.1	25.3	7.7	61.1	12	20.8	18.7	5.7	42.2	0.11	0.10	0.034
13, 14 & 15	12.5	5.0	6.8	24.3	10	11.2	4.4	6.0	21.6	0.06	0.024	0.032
16, 17 & 18	17.1	10.4	9.4	36.9	10	15.2	9.2	8.4	32.8	0.08	0.049	0.045
19	24.1	6.7*	neg.	30.8	5	43.0	12.0	neg.	55.0	0.228	0.063	neg.
20,21,22, 23,24,25	13.1	14.9	20.0*	48.0	12	9.7	11.0	14.8	35.5	0.052	0.059	0.08
26, 27 & 28	14.4	7.1	0.7	22.2	8	16.0	8.0	0.8	24.8	0.085	0.043	0.004
29 & 30	26.7	19.9	neg.	46.6	8	29.7	22.1	neg.	51.8	0.157	0.117	neg.
Odd Test	15.7	5.1	6.0*	26.8	10	13.9	4.6	5.3	23.8	0.074	0.025	0.028

* Scale very dark - presumably due to presence of concentrate.

deposition. It is probably not due to limitations in the amounts of scale-forming material present in the stickwater, as there is no difference in the pattern of scale formation between forward and reverse feed operation.

It is important to note that these estimates apply only to stainless steel tubes. Factory managers have often reported that the rate of scaling of mild steel tubes is more serious, and this is confirmed by another series of tests carried out by FIRI in Walvis Bay (7).

The first effect calandria of a low temperature concentrator operating at 218°F was experimentally fitted with four stainless steel tubes, which were thus subject to the identical conditions as the remaining, mild steel tubes.

After 55 hours of operation - during which period both sets of tubes had been cleaned, and the scale from each tube weighed nine times - it was found that on the average about 1.7 times as much scale had been removed from the mild steel tubes as from the stainless steel tubes.

There appears to be a tendency for the scale formed inside stainless steel tubes to partially flake off before it has attained a substantial thickness. This is most strikingly illustrated by a corollary to the above test. The plant was this time operated for extended periods before cleaning. After running for 48 hours, the average weight of scale per single tube removed from four stainless steel and six adjacent mild steel tubes was 1.24 gms. and 4.31 gms. respectively. After another run of 60 hours without cleaning, these figures had risen to 3.0 gms. and 27.0 gms. respectively.

Stainless steel tubes not only appear to foul less rapidly, but are also reported to be easier to clean. Laboratory tests at FIRI indicate that the adhesion of scale to the tube surface depends largely on its smoothness. This partly explains why scale in stainless steel tubes, which do not suffer from surface roughening due to corrosion, comes away in flakes, instead of having to be drilled or scraped out, down to the bare metal. It follows from this observation, that tubes with a highly polished internal surface should be particularly easy to clean. Tests at FIRI have confirmed this (8) and factory tests are planned. It will have to be seen to what extent the high surface finish can be maintained under normal operating conditions.

The use of stainless steel is not expected to stop tubes fouling altogether, and remains expensive. Numerous attempts have, therefore, been made to reduce the rate of scale deposition by chemical treatment of the stickwater. To evaluate different compounds, the composition of the scale must be known.

Few/

Few complete analyses of scale samples have been made, but judging from a number of approximate analyses carried out spectrographically by the Physics Department of the University of Cape Town (9), the following appear to be its chief constituents:

Substance	Content on dry basis: per cent
Ash	57 - 70
Fat	3.2 - 2.5
Organic Nitrogen	2.4 - 4.5
Phosphorus Pentoxide	20 - 24
Silica Dioxide	traces - 1
Magnesium	14 - 20
Calcium	traces - 3
Iron	0.4 - 10

Among the cations, magnesium was invariably the main constituent. The anionic fraction was mainly phosphate, and it is assumed that the inorganic parts of the scale are magnesium phosphate and some iron.

It would thus appear that the most promising scale inhibitors would be sequestering agents. It has in fact been reported that the only chemical treatment which has any beneficial effect consists of the addition to the raw stickwater of about 30 ppm of a commercially available product, which consists largely of sodium hexa-metaphosphate. This treatment is said to result in the formation of a softer and less tenacious type of scale, although the total volume is apparently unchanged.

The search for other treatments is complicated by the fact that all compounds to be considered must be non-toxic and must be fully acceptable to the ultimate user of the fish meal or solubles. Amongst the substances which have been tested without success are ferric chloride and various types of coagulating and precipitating agents.

Physical treatment of the stickwater to reduce scaling has also been considered. Preliminaty tests at FIRI indicated that by flocculating and clarifying the raw stickwater (by heating and decanting) the quantity of scale deposited could be significantly reduced.

Until such a time as scale formation can be prevented altogether, the factories will have to contend with the cleaning problem. The most important advanse in cleaning technique has been the provision of one extra - or "floating"-effect per stickwater plant. The effects are so interconnected that each one of them can be isolated in proper rotation for cleaning. In a triple effect plant equipped with one floating effect, the change-over is made roughly at 12 hour intervals, each calandria thus being cleaned every 36 hours.

For routine cleaning (i.e. while the rest of the plant is operating) the tubes are drilled out with rotary cutters. It takes two men up to eight hours to drill out a calandria with 450 tubes. In spite of the fact that smooth stainless steel tubes are easier to clean than mild steel tubes, the factories prefer to adhere to their 12-hourly change-over cycle.

At weekly to 3-weekly intervals (depending on the fish landings), the plant is stopped altogether and boiled out for 8 hours with a 10% caustic soda solution. This is followed by drilling of the tubes in the ordinary way. The caustic soda treatment helps to soften and partly remove the organic incrustations which permeate the true, inorganic scale.

There is another chemical treatment which is known to loosen or soften the hard scale deposits. This consists of exposing the tubes for 3 to 4 hours to unheated solutions of hydrochloric or sulphuric acid. Details have been reported by FIRI (10). The relative effectiveness of a 2% sulphuric acid solution was found to be similar to that of a 7% hydrochloric acid solution. Either acid is only safe to use in the presence of an inhibitor to protect those parts of the evaporator which are not covered by scale. During use and re-use the inhibitor is consumed, and the inhibited acid eventually reaches a stage when it can again attack the exposed metal. As no satisfactory chemical method for detection of the inhibitor appears to exist, its presence can only be checked by observing the action of the solution on bare metal specimens. The lingering uncertainty about the (accidental) attack on the evaporators by insufficiently inhibited cleaning solutions has impeded the general acceptance of this treatment by the factories.

The point to note is that no combination of tube material and chemical de-scaling has yet completely eliminated the necessity for regular cleaning by mechanical methods.

Reference must also be made to a problem which a few years ago was a source of constant annoyance to many factory managers, and in fact initiated this investigation into heat transfer in stickwater concentrators.

At the time, the landings of maasbanker in South Africa were considerable (130,000 tons in 1954 compared with 19,000 tons in 1959). When concentrating stickwater derived from this species, operators frequently had the experience that the evaporator ceased to function suddenly and without apparent reason. When the calandria were opened up, the tubes of the final effect were found to be solidly blocked with a tenacious, rubbery mass, which could only be dislodged by repeated drilling, scraping and boiling with caustic soda solutions. The clearing of a calandria thus fouled took several days and yet the same breakdown was liable to occur again, after only a few hours' operation.

Samples/

Samples of the rubbery matter removed from blocked tubes were analysed at FIRI (11). Their most striking feature was their gelatine content of about 6.5%, based on a 70% moisture content. The high viscosity (presumably due to the higher gelatine content) of maasbanker stickwater as compared with pilchard stickwater was subsequently confirmed by direct measurements (Appendix I).

What appeared to happen in practice was that the movement of the concentrate in the tubes of the third effect became progressively more sluggish, until it ceased altogether, causing the material in the tubes to be vacuum dried. This view was supported by the absence of significant quantities of true, inorganic scale-forming compounds in the specimens submitted for analysis.

The obvious solution of this problem was to reduce the viscosity of the concentrate, thus maintaining the circulation and forestalling tube blockage.

There are three known ways of reducing the viscosity of stickwater concentrate. It can be subjected to heat, which results in a breakdown of the large protein molecules into smaller, more soluble fractions. The Norwegian Herring Industry Research Institute (12) demonstrated that by keeping samples of herring solubles of 50% T.S. at 150°C, the viscosity dropped from 84 centipoises to 79 centipoises after one hour, to 58 cp after two hours and to 19 cp after four hours. This treatment is however not favoured owing to the risk of impairment of the nutritive quality of the product.

During the course of this investigation (12) it was discovered that the insoluble (as opposed to the soluble) matter in the concentrate contributed very substantially to its viscosity. It was concluded that an insoluble matter content of more than 15% of the total solids would cause the (herring) solubles to be too thick to handle by conventional methods. Although clarification or filtration of the stickwater would also reduce the rate of scale formation, the factories seem reluctant to apply this knowledge.[1]

The third method of reducing the viscosity of stickwater concentrate viz. by subjecting it to the action of certain proteolytic enzymes has come into general use. Particulars of this treatment is given in Appendix IV.

The use of enzymes to reduce the viscosity of maasbanker stickwater has virtually eliminated the risk of tube blockages. Together with the decline in maasbanker landings in recent years, this has for practical purposes caused the problem to disappear.

But this cannot be said of the final problem in this series: the low plant capacity due to poor heat transfer rates. It is unfortunate that both the evaporative efficiency (as pounds of evaporation per sq. ft. of

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heating surface), and the thermal efficiency (as pounds of steam required to evaporate one pound of water) are considered of secondary importance by factory personnel. Deficiencies in either of these two categories are too easily ameliorated; by installing larger plant if the capacity is too small, or by ordering an extra boiler if there is a shortage of steam.

In either case money is spent that could be more usefully employed elsewhere; and factory managers are becoming reconciled to conditions which would not be accepted without query by other industries.

FIRI has occasionally been called upon to test the performance of concentrators which were reported to be operating below their rated capacity. At one factory the Gross U Value was estimated as 40 BTU/sq.ft. °F hr.; at another it was only 30 BTU/sq.ft. °F hr.

The Gross U Value is defined as :

$$\frac{\text{Total Heat of Evaporation}}{\text{Overall Temperature Difference x Heating Surface per Effect}}$$

It is equivalent to the Overall U Value that would apply if a multi-effect plant were operated as a single effect.

The lowest Overall U Value quoted in the literature (13) for this type of equipment operating under the worst conditions is 60 BTU/sq.ft. °F hr. Under normal conditions (when concentrating thin liquors) the same source quotes a U Value of 300 BTU/sq.ft. °F hr.

The factory tests indicated - although accurate figures were difficult to obtain - that the heat transfer coefficients in the final effects were abnormally low. This was confirmed in preliminary tests made with the triple effect pilot plant described in the next chapter. In a typical test, operating under ordinary factory conditions the Overall U Values for the First, Second and Third effects were 256, 122 and 59 BTU/sq.ft. °F hr. respectively.

Referring to the specification for a stickwater plant quoted at the end of the previous chapter, the nominal temperature differences between steam and liquid for the three effects were :

First Effect	Second Effect	Third Effect
27°F	44°F	112°F

Assuming a Latent Heat of evaporation for water of say 1000 BTU/lb, the Gross U Value for this plant would be .

$$\frac{38,000 \text{ x } 1,000}{1,800 \text{ x } 183} = 115 \text{ BTU/sq.ft. °F hr.}$$

Making the simplifying assumption that the same quantity of heat is transferred in each effect, it follows that:

Overall U Value for the First Effect $= \frac{115 \text{ x } 183}{3 \text{ x } 27} = 260$ BTU/sq.ft. °F hr.

" " " " " Second " $= \frac{115 \text{ x } 183}{3 \text{ x } 44} = 160$ "

" " " " " Third " $= \frac{115 \text{ x } 183}{3 \text{ x } 112} = 63$ "

The/

The manufacturers evidently have no illusions about the evaporative efficiency of their plant and are aware of the fact that the individual coefficients for the three effects are roughly in the ratio 4 : 2 : 1. It is remarkable how the designers as well as the users appear to have resigned themselves to this situation without inquiring into its causes. Nevertheless, it appears short-sighted to pretend that the problem does not exist merely because the factories can still afford to ignore it.

CHAPTER IV

PLANNING AND EXECUTION OF EXPERIMENTS

PURPOSE.

The purpose of the experiments was:

a) to determine the range of heat transfer coefficients for typical commercial stickwater plant;

b) to test the effectiveness of certain changes in operating technique which on practical and theoretical grounds promised to improve the heat transfer coefficients in existing plant; and

c) to confirm a proposed theory for the operating principles of natural circulation evaporators, particularly when concentrating fluids of high viscosities.

APPARATUS

The most convincing results would be expected from tests on full-scale commercial plant. In practice it is virtually impossible to dovetail an experiment calling for exact control and measurement into the routine production activity of a factory. Furthermore, the instrumentation of commercial plant is generally inadequate, and finally it is impractical to use an industrial stickwater concentrator for tests with other working fluids. It was therefore unavoidable for the tests to be performed in the laboratory.

Fortunately, studies in heat transfer can readily be made with apparatus comprising only the essential elements of the larger commercial units. Where multi-tube calandria are involved it is often sufficient to construct the experimental equipment round one single tube of typical dimensions. This expedient was adopted in the present case.

The apparatus that was used consisted of a triple effect evaporator, with each effect based on a single heated tube. The assembly of heated tubes, return tubes, liquid receivers, condensers, etc., is diagrammatically shown in Fig. 4. Where single effect tests had to be performed, the appropriate section was isolated and operated as required.

As the scope and the purpose of these tests widened, it became necessary from time to time to improve the construction of certain components. Similarly, the experience gained during the four years devoted to this investigation made it possible to simplify some of the earlier methods. Such modifications frequently necessitated recalibrations and the revision of correcting factors, but they were never sufficiently drastic to destroy the value of earlier results. All changes made, as well as their implications, are fully discussed under the appropriate headings.

The <u>heated tubes</u> used for all tests were of a type which is popular with plant manufacturers. They were made of solid drawn, unpolished grade AISI 316 stainless steel. Their length was 8 ft., internal diameter 35 mm., and wall thickness 1½ mm.

Although/

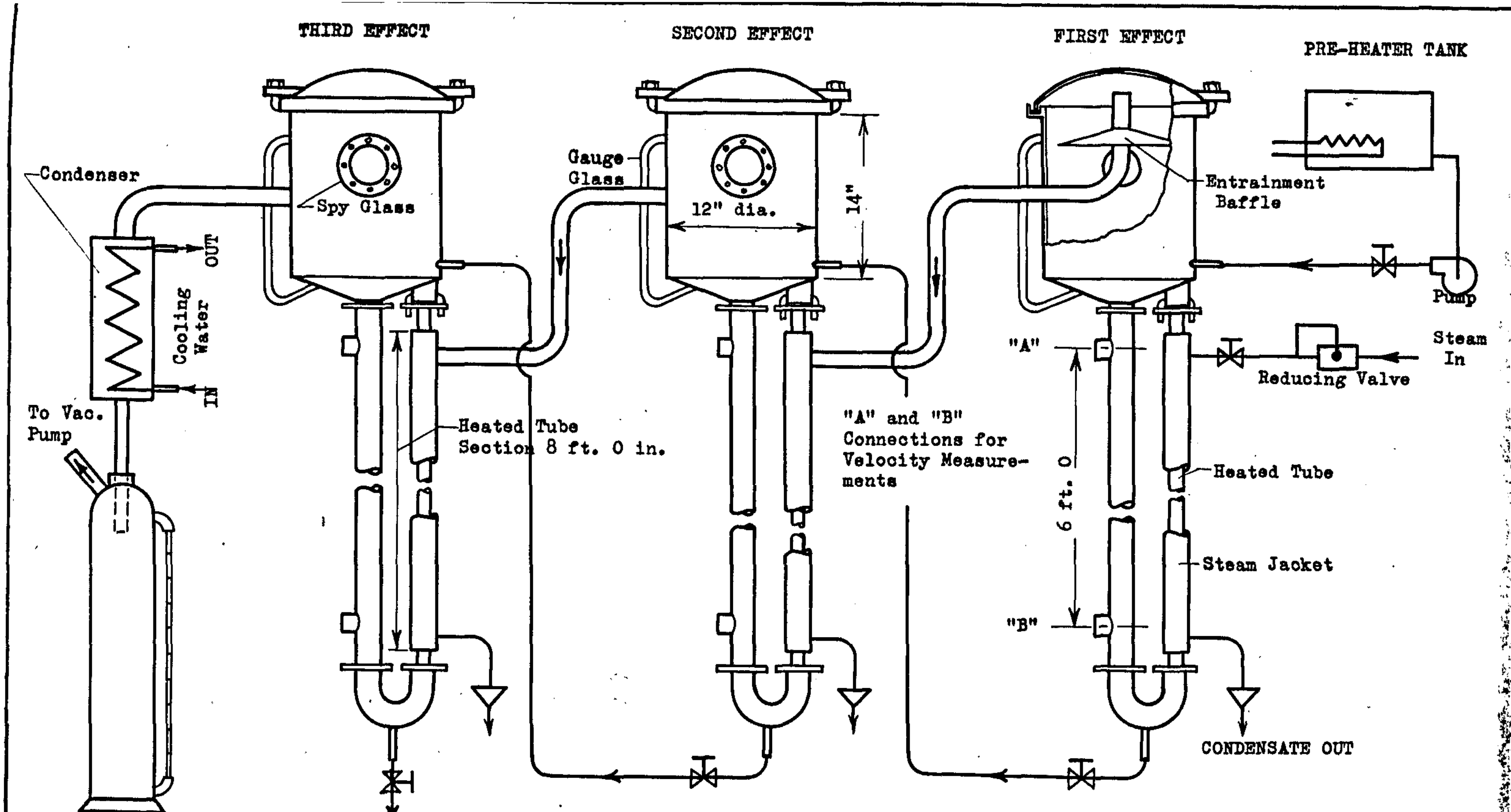

FIG. 4 : APPARATUS : ASSEMBLY DRAWING

Although tubes made of mild steel are also used in practice, stainless steel tubes were preferred as the corrosion of the mild steel (particularly on the inaccessible outside of tubes heated by distilled vapours) would have led to undesirable complications.

The <u>source of heat</u> was steam, supplied to a jacket concentrically surrounding the heated tube. As originally constructed, the jacket covered 7 ft. 6 in. of the heated tube, and the ends were designed to facilitate the substitution of different types of heated tubes, as shown in Fig. 5. At that stage it was intended to test the effect of different tube materials and surface finishes on the blockages caused by maasbanker concentrate, as described in the previous chapter. When interest was subsequently focussed on the more fundamental aspects of heat transfer, the improved construction shown in Fig. 6 was adopted. This had the advantage of utilising the full length of the heated tube, and yet providing a clear break between heated and unheated tube sections. With the earlier construction, the exact length of the heated tube section was somewhere between 7 ft. 6 in. and 8 ft., giving rise to a maximum possible error of $\frac{6 \times 100}{96}$ or say 7 per cent.

The <u>return tube</u> diameter of 47.5 mm. I.D. represents a compromise between the desire to concentrate the pressure drop of the system in the heated tube and the requirements for measuring the liquid velocity by the methods described in Appendix II.

To compare the pressure drops in the heated and the unheated sections of the circuit, take the length of the straight portion of the return tube as 8 ft. 6 in. and assume the tight return bend to be equivalent to 60 tube diameters (14). Hence the total equivalent length of the unheated section is 18 ft. Take the overall length of the heated section as 8 ft. 6 in. Also assume that the contraction coefficient at the inlet to the return tube is 0.5, the contraction coefficient at the inlet to the heated tube is 0.28, and the discharge coefficient from the heated tube is 1.0 (15). The ratio of the pressure drops in the two sections can then be expressed as:

$$\frac{\frac{4 \times 18}{0.156} \times f_1 + 0.5 + 0.28}{\frac{4 \times 8.5}{0.115} \times f_2 + 1.0} \times \frac{v_1^2}{v_2^2}$$

(where f_1, f_2, and v_1, v_2 refer to the friction factors and liquid velocities in the unheated and heated sections respectively).

On simplification this becomes: $\frac{460f_1 + 0.78}{295f_2 + 1.0} \times 0.295$

The ratio of the Reynolds Number for the flow in the unheated and the

heated/

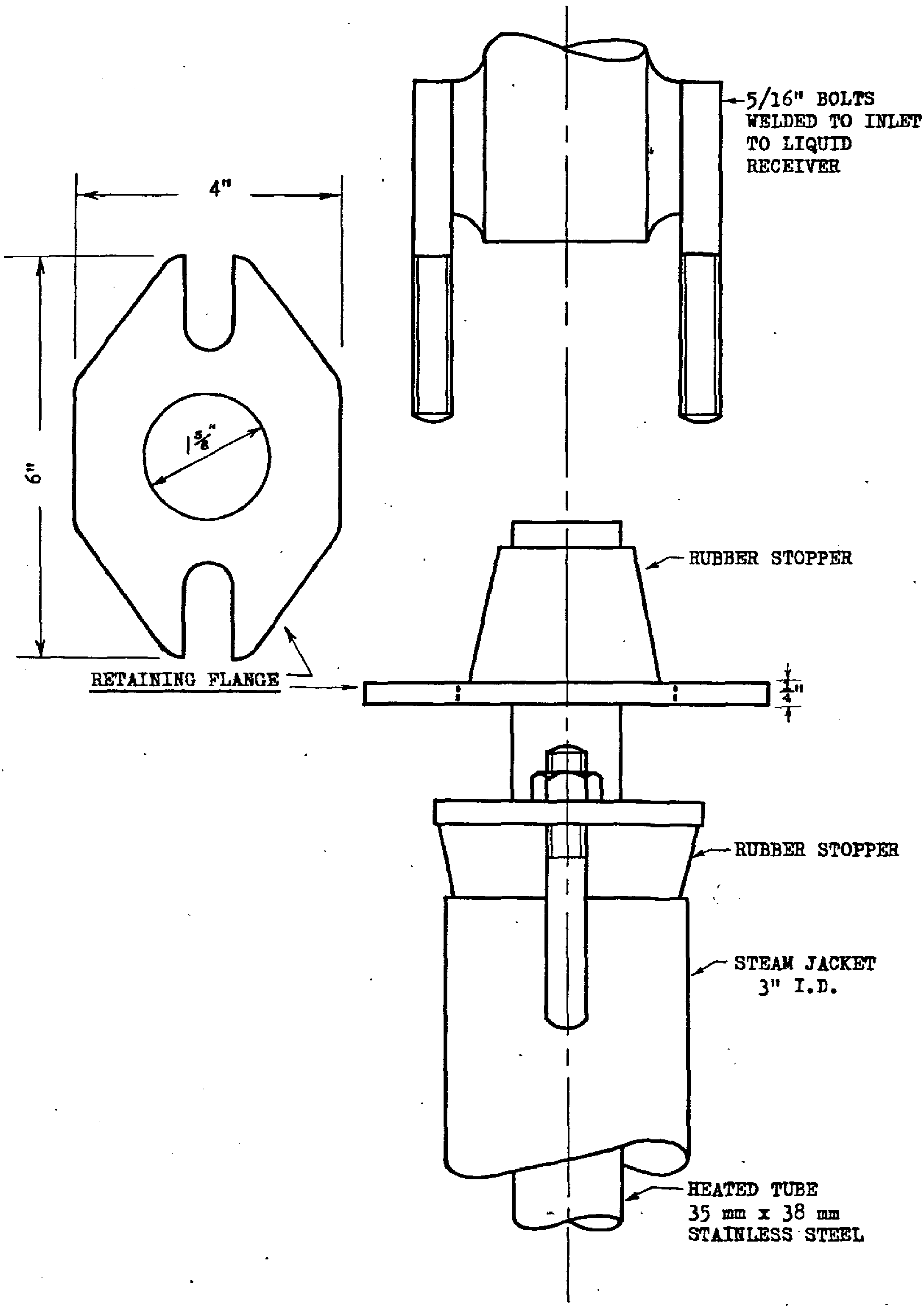

FIG. 5 : ORIGINAL TUBE AND JACKET ASSEMBLY

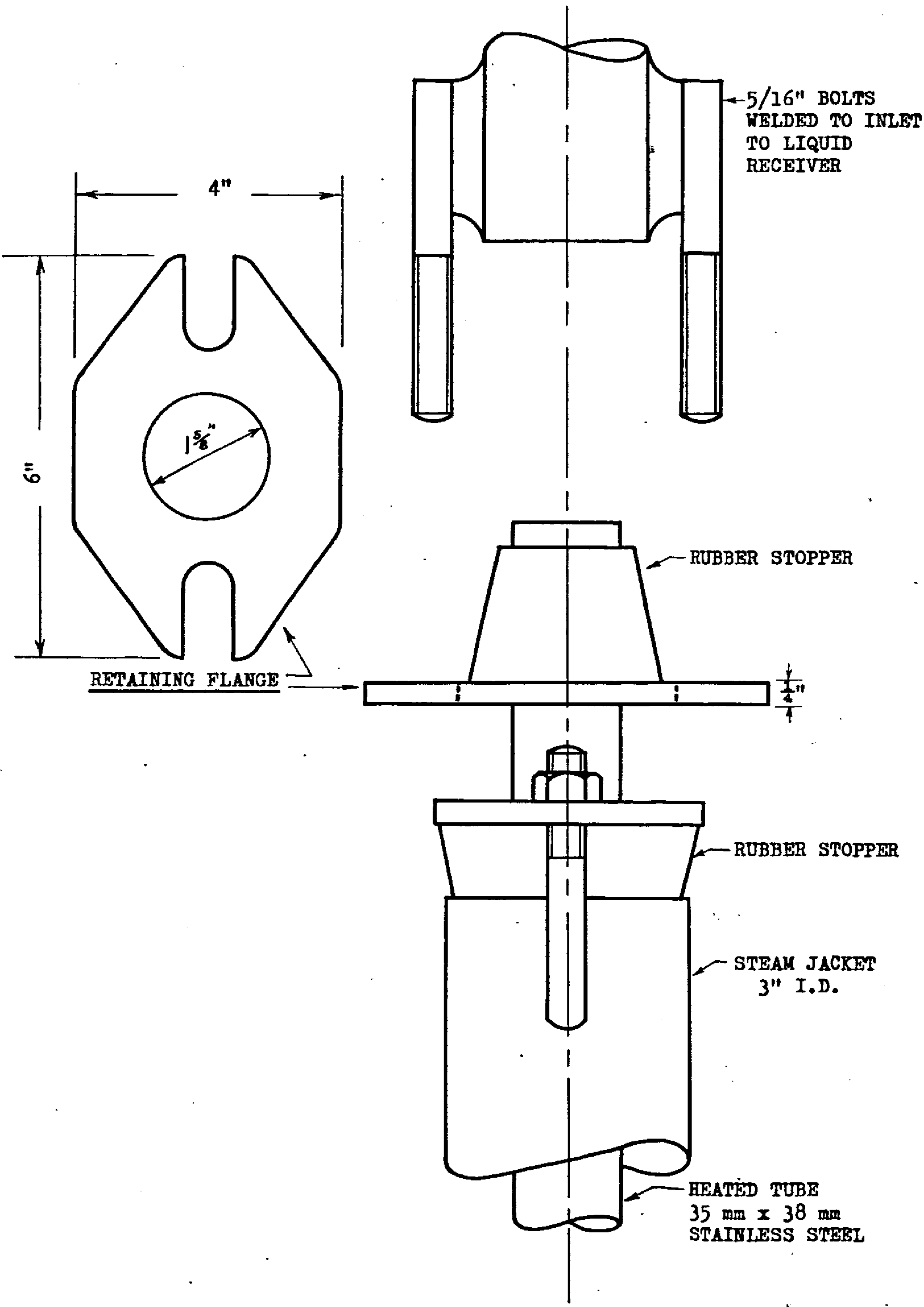

FIG. 5 : ORIGINAL TUBE AND JACKET ASSEMBLY

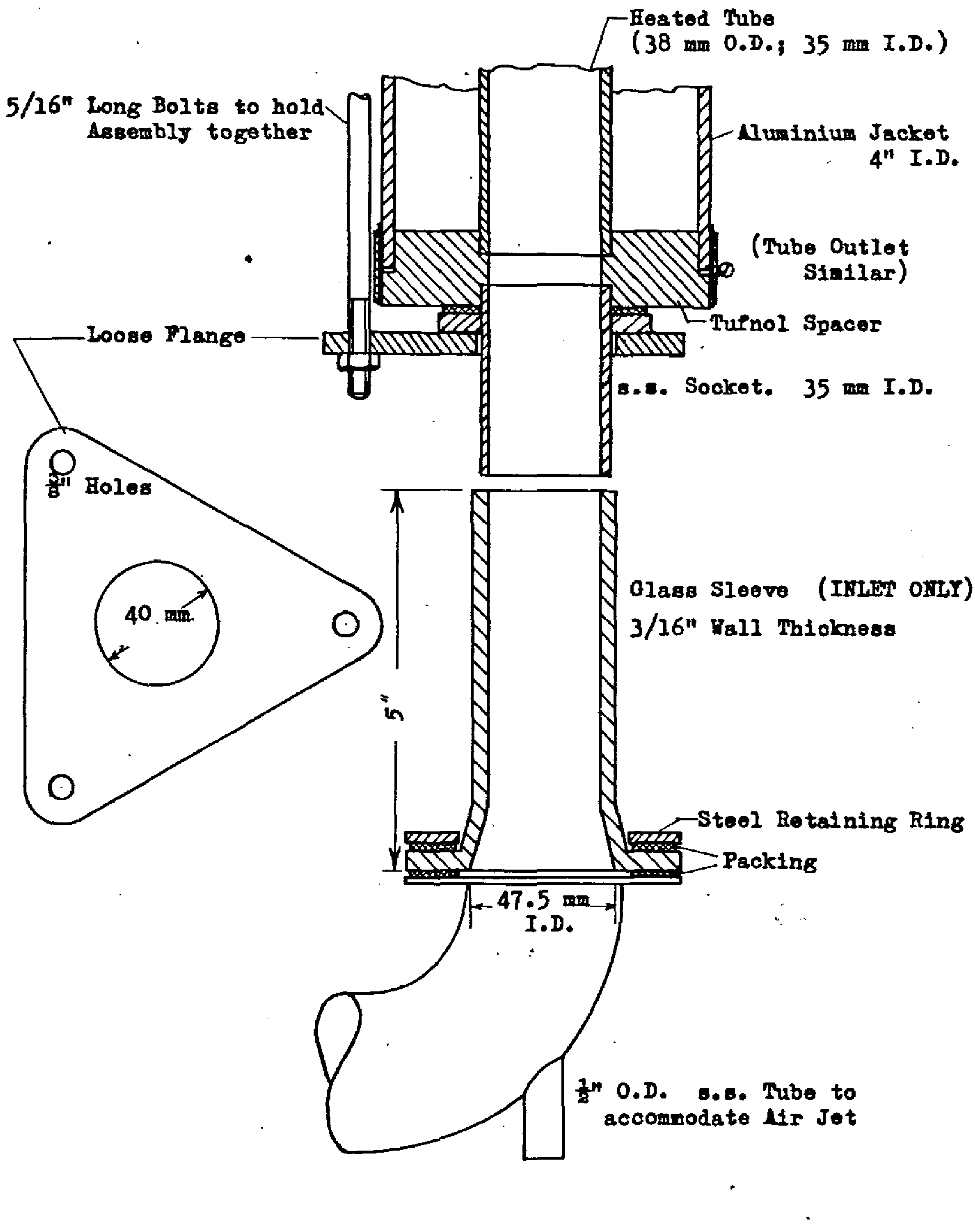

FIG. 6 : IMPROVED TUBE AND JACKET ASSEMBLY (INLET)

heated sections is the inverse of the ratio of the tube diameters i.e.

$$\frac{0.115}{0.156} \text{ or } 0.74$$

A typical f - Re curve for tubes with smooth surfaces (16) shows that the ratio of the two pressure drops for fully turbulent conditions in the heated tube (Re = say 100,000) will be approximately

$$\frac{460 \times 0.0047 + 0.78}{295 \times 0.0044 + 1.0} \times 0.295$$

$$= \underline{0.38}$$

The corresponding ratio for fully developed laminar flow (Re = say 100) will similarly be approximately

$$\frac{460 \times 0.204 + 0.78}{295 \times 0.15 + 1.0} \times 0.295$$

$$= \underline{0.6}$$

Although the friction pressure drop in the unheated section is not negligible for single phase flow, it practically becomes so when boiling commences in the heated section. It is shown in Appendix IV that even with quite moderate vapour:liquid ratios the two-phase friction pressure drop is several times as high as the corresponding single phase pressure drop. It is therefore reasonable to assume that in this apparatus the flow resistance offered by the non-heated portion of the circuit did not significantly change the flow pattern in the heated (boiling) section.

That similar conditions probably apply in practice appears from a comparison with a typical calandria described previously. The calandria comprises 176 tubes, 35 mm. i.d. and 10 ft. 2 in. long. The equivalent length of the unheated section of the circuit is approximately 15 ft., and the diameter of the return tube is 10 in. The ratio of the liquid velocities in the return tube and in the heated tubes is therefore:

$$\frac{176 \times 0.0117}{0.694} = 2.97$$

The ratio of the Reynolds Numbers in the unheated section and in one of the heated tubes is the ratio of the tube diameters multiplied by the velocity ratio i.e.

$$\frac{10 \times 25.4 \times 2.97}{35} = 22$$

Taking a typical case of turbulent flow with a Reynolds Number in the heated tube of say 20,000, the corresponding friction factor is 0.0062. The Reynolds Number in the return tube will be 460,000, and the friction factor = 0.0032. Hence, ignoring the expansion and contraction losses - which tend to cancel each other - the ratio of pressure drops under non-boiling conditions will be approximately

$$\frac{0.0032 \times 15 \times 0.108}{0.0062 \times 0.834 \times 10.2} \times (2.97)^2 = 0.8$$

By the same reasoning as before, it would appear that the flow pattern in the tubes of the calandria of this commercial plant is not seriously affected by the flow through the return tube.

(NOTE/

(<u>NOTE</u> that the case for laminar flow in both the heated and unheated tubes is unrealistic owing to the large difference in the appropriate Reynolds Number. If such hypothetical conditions are assumed to be possible, the ratio of pressure drops would be:

$$\frac{15 \times 2.97 \times 0.0117}{10.2 \qquad 0.694} = 0.07$$)

The <u>liquid receiver</u> was of simple design, with a removable cover to facilitate de-scaling of the heated tube. The entrainment separator, which proved to be surprisingly effective, was fitted inside the liquid receiver, to minimise radiation losses.

The <u>condenser</u> consisted of a specially strengthened motor car radiator suspended inside a tank of running water.

The <u>distillate</u> from the first effect jacket was collected in an open vessel after discharge from a "Sarco" thermostatic steam trap. The distillate from the second effect was recovered in a similar manner, except when the pressure in the jacket was sub-atmospheric. In the latter case it was found after some experimentation, that the distillate could best be collected in a graduated 1 litre separating funnel, which was periodically drained into an evacuated 5 litre Erlenmeyer flask. Non-condensable gases were purged from the system at the end of eabh distillate collection (or roughly at 5-minute intervals). The distillate from the third effect jacket was collected in a converted, disused gas cylinder of 30 litres capacity, fitted with a gauge glass and connected to the laboratory vacuum system. When the condensate was collected in open vessels, these were periodically weighed, and the readings are accurate to within $\pm 1\%$. When collected in graduated Erlenmeyer flasks, the readings are accurate to $\pm 5\%$. Readings taken with the graduated gas cylinder are accurate to $\pm 1\%$.

The following <u>pumps</u> were used for the purposes indicated:

1) a $\frac{3}{4}$ in. Kono Pump, Type HM 11, for feeding the first effect during triple effect tests;
2) a $\frac{1}{2}$ in. "Jabsco" Flexrotor pump, Model 584, driven via a set of cone pulleys giving 12 combinations for a maximum and minimum delivery of 6 and 17 gals/min. respectively, was used for forced circulation; and
3) a $\frac{1}{2}$ in. Gear Pump, made by American Machine Products, was used for "reverse feeding" and "throttled flow" tests.

These pumps performed satisfactorily when the discharge was above atmospheric pressure. But with suction and discharge both at sub-atmospheric pressure, as when reverse-feeding from the third to the second effect, or for forced or throttled circulation under vacuum, it proved extremely difficult to eliminate the air leaks which persisted, particularly through the shaft seals and stuffing boxes. It appears in retrospect, that for such unusual purposes pumps fitted with magnetic

couplings/

couplings or similar special seals would have well justified their extra cost.

MEASUREMENTS, CONTROLS AND INSTRUMENTATION

Heat transfer coefficients were calculated from the fundamental relationship $q = U_a \times A \times \Delta T_o$ or $q = h_a \times A \times \Delta T$

where q is the rate of heat transmission to the heated liquid ... BTU/hr.

U_a is the average overall heat transfer coefficient from the condensing steam to the bulk of the liquid ... BTU/sq.ft. ^oF hr.

h_a is the average coefficient of heat transfer between the tube wall and the bulk of the liquid ... BTU/sq.ft. ^oF hr.

A is the wall area of the tube ... square feet

ΔT_o is the temperature difference between the condensing steam and the bulk of the liquid ... degrees F.

ΔT is the temperature difference between the tube wall and the bulk of the liquid degrees F.

While the meaning of these parameters is clear, their exact definition and accurate determination is a difficult task, and the already formidable amount of literature on this subject is growing steadily. There is no limit to the ingenuity, the time and the money that can be expended on increasing the number and the accuracy of observations for a single experiment. But the craving for perfection must necessarily be bridled by the sober understanding of the purpose of the work in hand.

In the present instance the scope of the experiments was limited by the more immediate needs of the Industry. This meant that the answers to the problems that were posed had to be given in terms which could be translated into practice. Unless stated otherwise, the conventional terminology used in evaporator design has, therefore, been adhered to.

The heat transfer rate, q, was based on the weight of condensate drained from the respective steam jackets, and corrected for "flashing" and radiation losses.

The overall heat transfer coefficient, U_a was obtained by dividing q by the product of A and ΔT_o.

ΔT_o was taken as the difference between the average temperature measured in the steam jacket and the temperature of the liquid entering the heated tube. It stands to reason, and it has been shown e.g. by Boarts et al (17) that the temperature of liquids flowing upwards in vertical, heated tubes is not constant. However, the true liquid temperatures at different tube levels are difficult to determine, and are not required for commercial plant specifications.

The/

The error that can arise by assuming the bulk liquid temperature in the heated tube to be constant can be estimated by applying the appropriate dimensionless relationships, as discussed in Chapter VII.

It is largest in the final effects, as the liquid saturation temperature curve rises most steeply at low vapour pressures.

Assuming an overall temperature difference between the condensing steam and the bulk of the liquid of say 100°F, and a liquid bulk temperature of say 120°F, the maximum temperature rise of the liquid, assuming it has a viscosity of say 500 cp, would be about 20°F.

The maximum temperature would be reached near the tube outlet; hence the true average temperature difference between the steam and the bulk of the liquid would be about $100 - \frac{20}{2} = 90$°F instead of 100°F.

The error is therefore not negligible (although it is smaller at higher liquid temperatures and lower viscosities); but it is consistent and tends to cancel in comparative tests. Furthermore, the U_a values that are calculated when this error is ignored are too low rather than too high, which is an advantage for practical purposes.

The area of the heated tube section, A, was based on the internal tube diameter, as is usual in cases where the major thermal barrier is the liquid film on the inner tube wall. The results of the tests which indicated film coefficients for stickwater of the order of 50 to 300 BTU/sq.ft. °F hr., as compared with estimated steamfilm and tube wall coefficients of 1,000 and 2,000 BTU/sq.ft. °F hr. respectively, confirmed that this was a reasonable assumption. [The maximum error that could have arisen by basing the tube area on the internal diameter instead of on the wall centre diameter is $\frac{36.5 - 35}{36.5} \times 100 = 4\%$.]

The U_a values calculated in this manner are useful for measuring and comparing the performance of industrial plant. For a more fundamental analysis of the test figures it is desirable to estimate h_a, the film coefficient in terms of T, the average temperature difference between the inner tube surface and the bulk of the liquid. In the case of the earlier experiments, when only the steam jacket temperature was measured, h_a was, where necessary, derived from U_a by the use of assumed wall- and steam-film coefficients. In later tests the temperature of the tube wall was measured by means of thermocouples embedded therein, as shown in Fig. 7. This was a comparatively simple refinement, which also served to check the assumed steam film coefficient.

Apart from the measurements required to determine U_a or h_a, a number of secondary readings were also taken to complete the picture. When operating with stickwater or solutions of sucrose or corn syrup, the solids concentration was checked at half-hourly intervals by refractometer. Whenever convenient, the circulation velocity was measured (see Appendix II), and the barometer reading was recorded at the end of each run.

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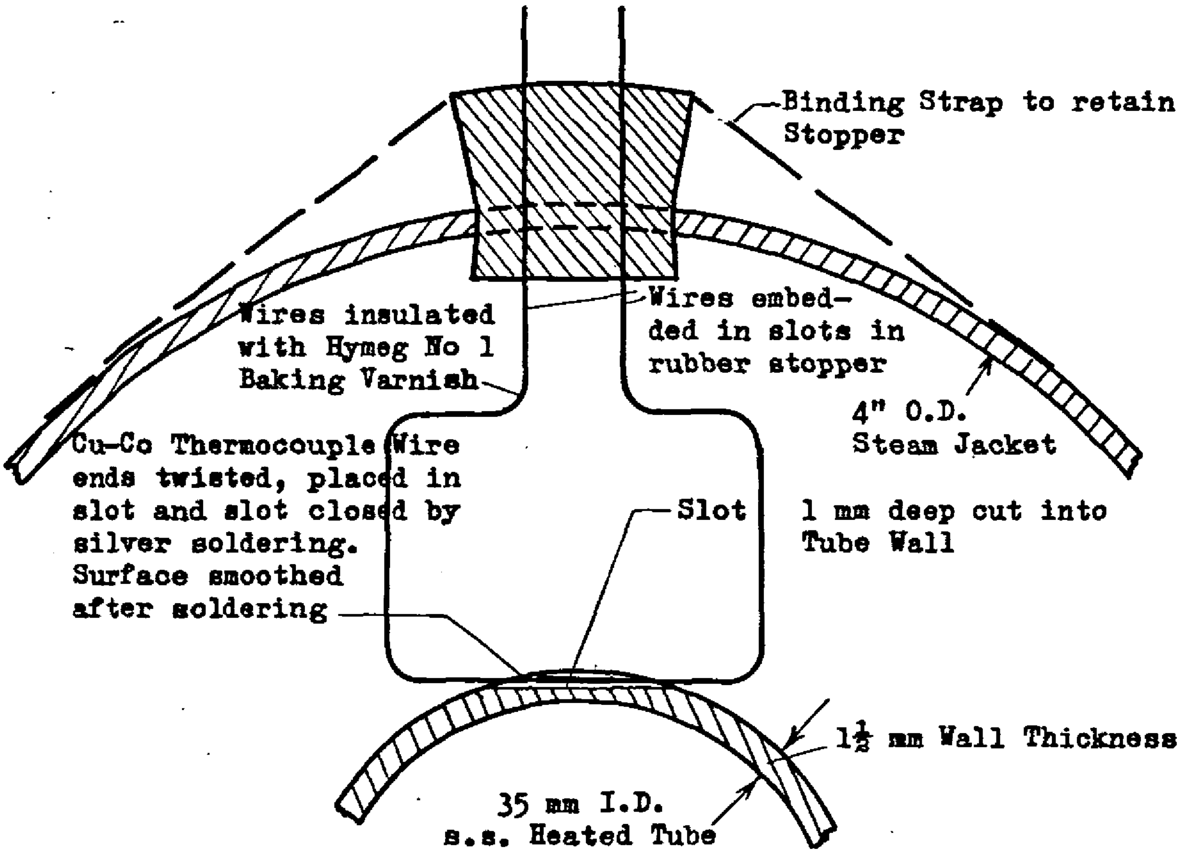

FIG. 7 : METHOD OF LOCATING THERMOCOUPLES IN TUBE WALL

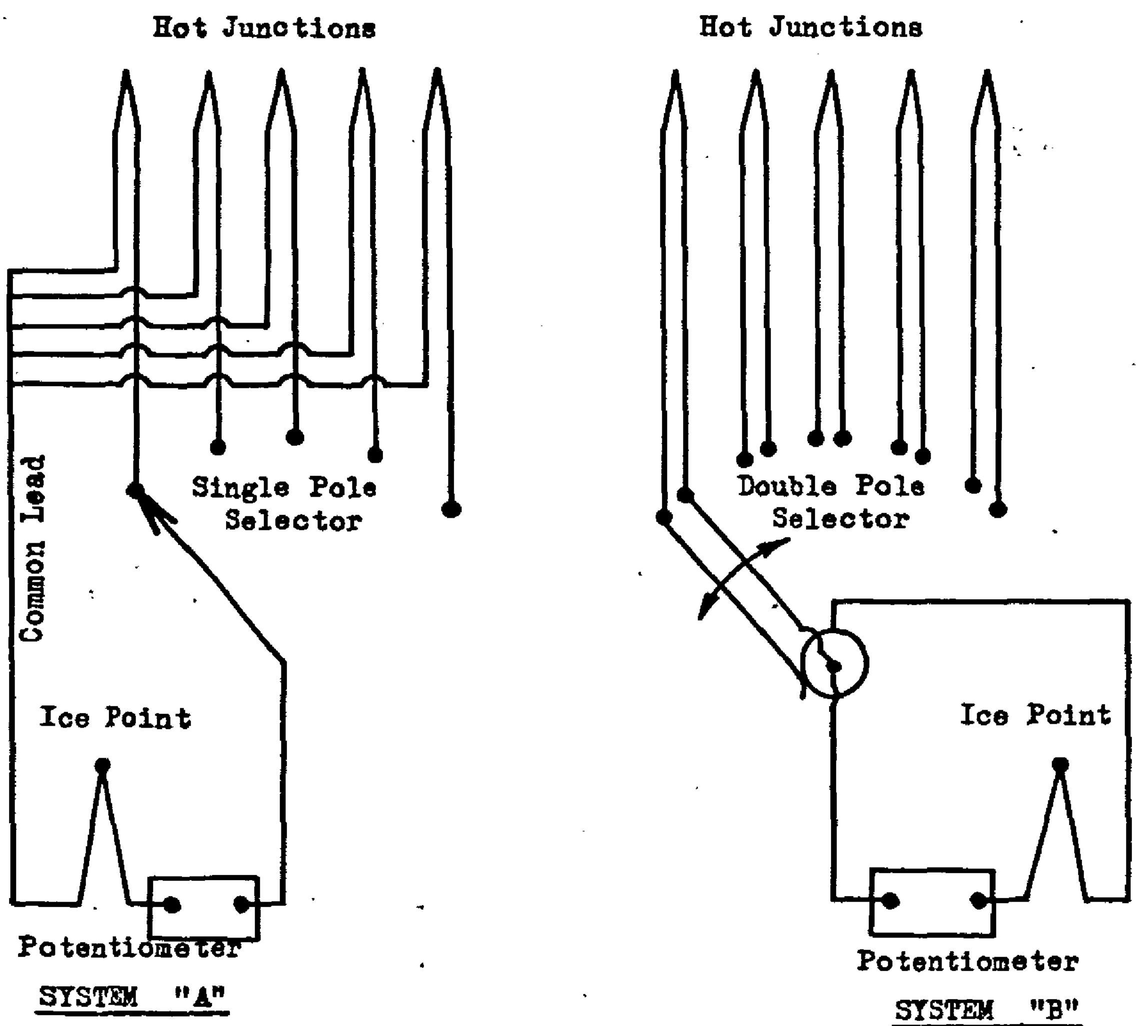

FIG. 8 : METHODS OF SELECTING THERMOCOUPLES

The following instruments were used:
For temperature measurements: A Negretti & Zambra quick-reading potentiometer; No. E-25111; range, 0 - 53 M. volts. Alternatively, a Leeds & Northrupp direct-reading potentiometer; Type 8663-S; triple range instrument, the range used being 50 - 250°F. Both potentiometers were used in conjunction with standard copper-constantan and iron-constantan wire of 0.022 in. diameter, supplied by Messrs. B.I. Cables.

Thermocouples were placed in the following positions:

a) in the liquid in the plenum space;
b) in the liquid about 1.5 in. below the entrance to the heated tube;
c) in the steam, 4 in. from the top of the jacket;
d) in the steam, 4 in. from the bottom of the jacket; and
e) evenly spaced at 30 in. interval embedded in the wall of the heated tube.

The method in which the tube wall thermocouples were attached is illustrated in Fig. 7. The thermocouples were connected to the potentiometer via a selector switch, as shown in Fig. 8. In some of the earlier tests a single pole selector switch was used (System "A"), and this had the effect of creating one complex couple out of the four individual couples in the tube wall. The individual wall temperatures determined during the tests were, therefore practically identical, and represent in fact the averag temperature for the four positions. Although this is not a disadvantage for the purpose of calculating the coefficient h_a, it creates the impression that the tube wall temperatures were constant.

The single pole selector box was therefore replaced by the double pole switch (System "B"), which causes the individual couples to be completely isolated from each other, and true instead of average wall temperature readings to be obtained. The effect is illustrated in Appendix III.

Secondary temperature readings were taken with laboratory mercury-in-glass thermometers supplied by H.G. Zeal. All thermometers and thermocouples were calibrated against a set of standard thermometers certificated by the N.P.L. The overall accuracy of all temperature readings was not less than ± 1°F.

The circulation velocity was measured by the devices described in Appendix II.

The airflow required for air-lift circulation, was measured with one of a series of rotameters supplied by Negretti & Zambra. The instruments were calibrated against the water displacement from an inverted measuring cylinder before use. While in use the floats of the instruments tended to hover around the average position, and the accuracy of the readings was about ± 5%.

Pressure and vacuum readings were taken either by mercury manometer, or by means of commercial, bourdon tube pressure gauges, which were calibrated on a dead weight tester supplied by Messrs. Ferris Products of Cape

Town/

Town. Manometer readings were accurate to ±0.5 mm Hg., and pressure gauge readings to ± 0.5 psi.

The refractive index (or %T.S.S.) of the working fluids was measured either with a Zeiss Abbe refractometer, or with a Bellingham & Stanley pocket refractometer with a sucrose scale up to 50%. T.S.S. readings of raw stickwater, sucrose and corn syrup were accurate to ± 0.5%. At high solids concentrations the stickwater ceased to give a clear line, and the accuracy of T.S.S. readings dropped to about ±1%.

The viscosity of the working fluid was read from viscosity vs. %T.S.S. curves, or else estimated as described in Appendix I.

The vacuum in the laboratory vacuum system was maintained by an Ingersoll Rand pump, type 30, model V244, driven by a 2 H.P. motor.

The following variables were controlled during individual runs:

1) The jacket temperature in the first effect. This was done manually by regulating the live steam stop valve, the steam pressure having previously been reduced from boiler pressure (75 psi) to 20 psi. A search was made for a thermostatic steam regulator with a sensing element small enough to fit inside the steam jacket, but such a device could not be supplied locally. With manual operation, the steam pressure was controlled to ± 2 psi when the pressure was above atmospheric, and to ± 1 psi when it was below atmospheric. In most of the single effect tests, the jacket outlet was open to the atmosphere and control presented no problem.

2) The vacuum in the final effect. A home-made vacustat adequately served this purpose. It consisted of a platinum wire which could be raised or lowered inside a mercury manometer to make contact at a selected level. Closing the circuit caused a relay to open a solenoid valve, which admitted air to the vacuum line. By regulating a throttling valve on the air line, the on and off cycles of the system could be adjusted to be nearly equal, and the vacuum fluctuations inside the liquid receiver were kept to less than 2 mm. Hg.

3) The temperature of the feed. This was controlled thermostatically, and generally maintained to within 5°F of the liquid temperature in the receiver. All but minor feed temperature variations were allowed for in the Heat Balance.

4) The liquid level in the receiver. In most cases this was done manually, and the level could generally be held to within ± 1 in. of the mark on the gauge glass. When working with water, or other working fluids up to a solids concentration of about 25%, the level could be maintained automatically by means of a device similar to the vacustat described under 2). But at higher concentrations the liquids invariably became too sticky, and the make and break of the contact with the level control wire became erratic.

5) The/

5) The solids concentration of the working fluid. During the triple effect tests this was maintained at the target level by opening and closing of the feed and drain valves on the last effect. The T.S.S. of the liquid in the first and second effects is determined by the evaporation rates, and therefore beyond control. The solids concentration fluctuated by about ± 1% in the first, and ± 2% in the second and third effects. When operating a single effect only, the T.S.S. of the liquid was kept to within ± 0.5% by feeding water to the liquid receiver at the same rate at which distillate was being collected.

<u>EXPERIMENTAL [illegible]</u>

The tests were divided into groups of from two to five runs. For one or more runs within each group, the apparatus was always operated with natural circulation, forward feed and with untreated stickwater or one of the other working fluids. The effect of changing the variable which characterised each test could thus be measured against its appropriate control. This procedure was adhered to (although it involves a certain amount of duplication), as it provided a running check on the proper functioning of the apparatus.

The duration of each run, from the time when all temperatures and pressures gave steady readings, was initially 1½ hours (Tests 1 - 38; 41 - 42; 45 - 53; 64 - 67; 71 - 76; 107 - 110). Later, when the operating technique had been improved, this time was reduced to ½-hour (Tests 39 - 40; 43 - 44; 54 - 63; 68 - 70; 77 - 106; 111 - 167).

Four sets of readings were taken during each run, at 30-minute intervals for the longer and at 10-minute intervals for the shorter runs. Freak readings were considered on their merits; when traced to an instrument error they were included at the average value of the remaining readings. If a freak reading could not be reasonably explained and adjusted, or if the readings collectively showed a trend away from the initial steady state, the results were verified by repeating the runs.

At the end of each group of runs with stickwater the tubes were descaled; in some cases the scale was collected to provide information about the rate of scale formation in commercial plant, as discussed in Chapter III. An experiment was conducted to estimate the effect of scaling on heat transfer in the apparatus. The results are shown in Table II. It was concluded that, for the limited operating time required to complete each group of runs, the error due to scaling was not likely to exceed 2%.

For all runs (Nos. 1 - 19; 26 - 30; 45 - 49; and 64 - 67) in which the apparatus was operated as a triple effect unit, the working fluid was raw stickwater obtained from one of the factories at Hout Bay. Runs 20 - 25 and 71 - 76 (triple effect), and all single effect runs with

stickwater/

TABLE II

EFFECT OF SCALING ON HEAT TRANSFER RATE

RUN NO.	TEMPERATURES °F				"U" VALUE BTU/sq.ft.°F.hr.				CHANGE in GROSS "U" VALUE	REMARKS
	Live Steam	I Liquid	II Liquid	III Liquid	I Effect	II Effect	III Effect	GROSS		
31	277	234	186	115	478	328	225	324	Control	Run made with water and clean tubes
32	274	225	180	115	375	325	231	301	-7%	Run made with water, but tubes not cleaned after roughly 10 hours operation with stickwater

CONCLUSION: The duration of each test run was about 2 hours - including starting up time. The estimated error due to scaling is therefore say 2% per run. This is less than the anticipated experimental error.

stickwater were made with concentrate thinned down to the average solids content (5% - 10%) of stickwater. Although no significant difference was detected between the physical properties of raw stickwater and the fluid reconstituted from concentrate, the tests to determine the performance of commercial plant, which are dealt with in Chapter V, were all made with unprepared stickwater.

CALCULATIONS

The first step in calculating the required heat transfer coefficients was to itemise the heat sources and losses. In all triple effect tests a heat balance was established by estimating both the total heat supplied to, and the total heat delivered by each individual effect. In a few cases where these figures failed to tally, the particular runs were repeated. Owing to the smaller number of variables involved, and the better control that could thus be achieved, in the single effect runs, it was found that it was unnecessary to establish a heat balance, and the heat transfer coefficients were calculated directly from the heat delivered by the apparatus.

In either case the major item was the latent heat given up by the condensate collected from the steam jackets of the individual effects, or from the water-cooled condenser. The following heat sources and losses were, however, also taken into account:

the sensible heat of the condensate;
the "flash" heat released by discharging the jacket condensate to a lower pressure;
the sensible heat in the liquid feed and discharge from each effect;
the radiation loss from the body of each effect; and
the radiation loss from the steam jackets and lines.

The coefficients U_a and h_a were calculated by dividing the nett figure for the heat transmitted by the effective area of the heated tube and by the nominal temperature difference.

Specimen sets of readings and calculations are given in Appendix III.

In certain cases supplementary calculations were made to estimate Reynolds, Prandtl and Nusselt Numbers, to determine the Gross U Values (defined by : $\dfrac{\text{Total Heat Transferred}}{\text{Total Temperature Drop x Heating Surface per Effect}}$) and to compute the power required to achieve given rates of forced circulation and air injection. These calculations are fully discussed in Chapter VIII.

TRIAL RUNS

Apart from preliminary tests to check the functioning of the various components, a number of trial runs with water were made to determine the reproducibility of the results obtained with the apparatus. Particulars of these tests are shown in Tables III and IV.

Table/

TABLE III

REPRODUCIBILITY TESTS

TRIPLE EFFECT

RUN NO.	OPERATION	TEMPERATURES °F				LIQUID VELOCITY FT./SEC.			"U" VALUE BTU/Sq.ft.°F.Hr.				
		LIVE STEAM	I LIQUID	II LIQUID	III LIQUID	I EFFECT	II EFFECT	III EFFECT	I EFFECT	II EFFECT	III EFFECT	GROSS	
33	NATURAL CIRCULATION PURE WATER CLEAN TUBES	259	225	195	155	2.1	1.7	1.45	324	277	190	258	VARIATION 5%
34		258	224	193	161	2.3	1.8	1.4	304	258	229	264	
35		258	223	189	150	2.2	1.95	1.6	325	268	225	270	
36		264	223	178	102	2.9	2.35	1.1	456	342	208	308	VARIATION 1%
37		258	221	171	99	2.9	2.0	1.2	500	297	210	305	
38		278	235	184	112	2.9	1.95	1.2	448	301	222	306	VARIATION 6%
31		277	234	186	115	Not measured			478	328	225	324	

Table III relates to the earlier use of the apparatus in triple effect, whilst Table IV relates to the later single effect tests. It is evident that in the earlier tests the control of the live steam and the condenser temperatures were fairly erratic, and it is chiefly for this reason that the reproducibility was no better than about 5%. It must be remembered, however, that the chief purpose of the apparatus at that stage was to demonstrate the causes and the prevention of tube blockages. Yet the results obtained in those tests were sufficiently accurate to supply some hitherto unobtainable data on the performance of typical commercial plant.

The better reproducibility (about 1%) reflected in Table IV is due not only to better control but also to single effect as opposed to triple effect operation. The heat gain and loss corrections to be made in the former case are smaller in number and significance, and it is for this reason that the most critical tests of air-lift and throttled circulation were performed in single effect.

NOTE: All runs were originally identified by code numbers and letters relating to the purpose of the tests. For the sake of clarity, they have been re-numbered consecutively in the order in which they are referred to in the text.

TABLE IV

REPRODUCIBILITY TESTS

SINGLE EFFECT

RUN No.	OPERATION	TEMPERATURES °F LIVE STEAM	TUBE WALL	LIQUID	DIFF. TUBE WALL – LIQUID	LIQUID VELOCITY FT./SEC.	h_a BTU/SQ.FT. °F. HR.	
39	NATURAL CIRCULATION PURE WATER CLEAN TUBES	214	208	175	33	Not	418	VARIATION 1%
40		214	209	174	35	Measured	414	
41		215	-	160	55	1.93	382*	VARIATION 1%. Note * These are U values, not h_a values, as wall temperatures not measured in earlier tests.
42		221	-	165	56	1.90	386*	
43	NATURAL CIRCULATION CORN SYRUP TSS (Refractometer) 75%	213	210	120	90	Not	102	VARIATION 1%
44		213	209	119	90	Measured	103	

CHAPTER V

ESTIMATED HEAT TRANSFER COEFFICIENTS IN COMMERCIAL PLANT

As explained in Chapter IV, all tests consisted of a number of runs, at least one of which was a control run. Particulars for all control runs made with undiluted, untreated stickwater, and operating the experimental plant in triple effect and with natural circulation have been collated in Table V. The relevant data for a further group of control tuns in which the apparatus was operated in single effect have been recorded in Table VI. As the apparatus had purposely been designed along the lines of a full-scale plant, and as the operating conditions are representative of those found in practice, the data presented in the two tables can with reasonable certainty be accepted as applicable to most commercial concentrators.

An inspection of the experimental U_a values suggests a comparison with figures quoted in the literature. It appears however that little has been published about the performance of natural circulation evaporators, and virtually nothing about their use for concentrating viscous liquors. The only working fluids that have been tested to any extent are sugar solutions, but even they have not been used at concentrations at which laminar flow conditions are likely to have become established in the non-boiling sections of the heated tubes.

The situation is best summed up in the words of Oliver Lyle: "There is a great jungle of ignorance regarding heat transfer that badly needs explaining. It is important that heat transfer rate figures obtained on one plant and on one material should not be used to foretell the heat transfer rate on another plant and/or on another material". (19)

In spite of these reservations a summary has been made, as shown in Table VII, of overall heat transfer coefficients determined or recommended by different investigators under conditions which - in the most general way - might be considered applicable to stickwater concentrators. The most striking feature of this cross-section of expert opinion is the tenfold increase from the lowest to the highest recorded U values.

Overall heat transfer coefficients quoted for all types of heat exchangers are invariably flexible, but the maxima and minima rarely approach the ten to one spread observed for natural circulation evaporators.

The reason for this is that heat transfer in this type of plant depends on an interrelation of viscosity, bulk temperatures, temperature gradients, etc., which is more complex that in other conventional heat exchangers. This aspect is fully discussed in Chapter VII.

When comparing Tables V and VI with Table VII, it is evident that

stickwater/

TABLE V

TEST SUMMARY : CONTROL RUNS : STICKWATER CONCENTRATE

(concentrated at laboratory from raw stickwater)

TRIPLE EFFECT

Run No.	Effect	Temperatures °F			TSS % (Refractometer)	$P^{\frac{1}{3}}$ (P in lbs/sq. in)	U_a BTU/sq. ft. °F. hr.	$\frac{U_a}{\Delta T_o \cdot P^{\frac{1}{3}}}$
		Jacket	Liquid	ΔT_o				
45	1st	263	231	32	18	4.6	215	1.46
	2nd	229	180	49	29	2.8	71	0.52
	3rd	179	108	71	47	1.1	46	0.59
9	1st	273	232	41	16	4.6	210	1.11
	2nd	232	176	56	25	2.7	98	0.65
	3rd	176	91	85	46	0.85	67	0.93
46	1st	275	242	33	13	5.1	318	1.89
	2nd	242	190	52	21	3.1	123	0.76
	3rd	190	94	96	46	0.9	66	0.76
4	1st	273	233	40	14	4.7	250	1.33
	2nd	232	180	52	25	2.8	126	0.87
	3rd	180	89	91	48	0.8	72	0.99
6	1st	273	235	38	11	4.8	266	1.46
	2nd	234	185	49	18	2.9	139	0.98
	3rd	183	88	95	45	0.8	74	0.97
10	1st	270	233	37	13	4.7	256	1.47
	2nd	232	184	48	24	2.9	122	0.88
	3rd	183	91	92	46	0.85	59	0.75
13	1st	277	244	33	13	5.2	324	1.89
	2nd	244	194	50	18	3.2	146	0.91
	3rd	194	99	95	46	0.95	76	0.84
47	1st	277	236	41	15	4.8	256	1.30
	2nd	236	194	42	22	3.2	164	1.22
	3rd	194	106	88	46	1.1	68	0.70
48	1st	272	233	39	8	4.7	326	1.78
	2nd	233	196	37	13	3.3	229	1.88
	3rd	196	97	99	31	0.9	83	0.93
49	1st	272	238	34	8	4.9	372	2.23
	2nd	238	200	38	14	3.4	217	1.68
	3rd	200	95	105	31	0.9	72	0.76
16	1st	270	230	40	13	4.6	236	1.28
	2nd	228	174	54	22	2.6	111	0.79
	3rd	174	87	87	46	0.8	69	0.99

TABLE VI

TEST SUMMARY : CONTROL RUNS : STICKWATER CONCENTRATE

(concentrate ex Factories)

SINGLE EFFECT

Run No.	Temperatures °F			TSS % (Refractometer)	$P^{\frac{1}{3}}$ (P in lbs/sq. in.)	U_a BTU/sq. ft. °F. hr.	$\frac{U_a}{\Delta T_o . P^{\frac{1}{3}}}$
	Jacket	Liquid	ΔT_o				
50	186	100	86	46	1.0	73	0.85
51	179	107	72	37	1.1	65	0.82
52	181	108	73	37	1.1	59	0.74
53	180	105	75	45	1.1	78	0.95
54	213	124	89	57	1.4	77	0.62
55	213	121	92	41	1.3	129	1.08
56	213	120	93	40	1.3	118	0.98
57	213	121	92	50	1.3	139	1.16
58	213	122	91	61	1.3	137	1.15
59	213	118	95	60	1.3	135	1.09
60	213	118	95	63	1.3	122	0.99
61	213	124	89	42	1.4	119	0.95
62	213	124	89	40	1.4	118	0.95
63	213	118	95	60	1.3	128	1.04

T A B L E VII

OVERALL HEAT TRANSFER COEFFICIENT : AS REPORTED IN THE LITERATURE FOR NATURAL CIRCULATION EVAPORATORS

WORKING FLUID	TEMPERATURE OF LIQUID °F	TEMPERATURE DIFFERENCE STEAM-LIQUID °F	U BTU/SQ.FT. °F.HR.	S O U R C E
Water	140	20	205	Badger & Sheppard. Studies in Evaporator Design. Part I. Trans. Am. Inst. Chem. Eng., 1920, 13, 139
Water	167	20	320	
Water	212	20	440	
Fluids of low viscosity	-	-	100 - 300	Walter Arnold. "Der Apparatebou". Carl Hauser Verlag, 1959. p. 70
Fluids of high viscosity	-	-	60 - 160	
Water and dilute liquors	"Pressure"	-	500	O. Lyle. Efficient use of Steam. p. 296. (Reference 19)
	"Vacuum"	-	300	
Sugar Solutions: 1st Effect	215	10	630	R. Royds. Heat Transmission in Boilers, Condensers and Evaporators. Constable & Co. Ltd., London, 1921. p. 269
2nd "	204	11	450	
3rd "	162	41	200	
Deg. Brix				
Sugar Solution: 0	160	32	410	R. Royds. Ibid. p. 270
20	156	38	368	
30	150	49	310	
65-70	144	61	175	
Water	212	36	570	E. Kirschbaum. Chemie, Ingenieur, Technik. March 1962. 34. p. 183
Water	122	36	290	
Sugar Solution: 50% W/W	212	36	270	
Sugar Solution: 50% W/W	122	36	145	

stickwater behaves in a similar manner to most other fluids; i.e. U values increase with bulk temperature and temperature gradient, and decrease with viscosity. However, there does not appear to be any simple correlation. The important difference is that the U values for stickwater clearly tend towards the lower end of the range listed in Table VII for other working fluids.

The principal difficulty about interpreting the data in Tables V and VI is that the figures for different runs can only be compared in a general, unsatisfactory, way. In particular, the role of the most important variable, viscosity, cannot be clearly distinguished from that of the others, i.e. bulk temperature and temperature gradient. This difficulty can be overcome to a limited extent by grouping the two types of variables together, and presenting their correlation graphically.

Kirschbaum (20) and many other investigators have shown that for liquids with viscosities up to about 4 cp the liquid film heat transfer coefficient h_a is primarily a function of the bulk temperature (or saturation pressure), the viscosity and the temperature gradient, and can without gross error be written

$$h_a \propto \frac{P^a \times \Delta T^b}{\mu^c}$$ where a, b, and c are constants.

This fact cannot directly be applied to the current data, because U_a values were determined instead of h_a values, and because the true viscosity of the liquid could scarcely be defined, much less measured (See Appendix I). But both parameters may be defined in a manner which has a theoretical basis yet satisfies the needs of the designer and operator.

The error introduced by employing U_a values instead of h_a values assumes major proportions only at the higher levels. Assuming, for instance, an average steam film coefficient of 2000 BTU/sq.ft. °F hr., and a conductance through the tube wall of 1000 BTU/sq.ft. °F hr.,[1] the h_a value corresponding to a U_a value of 300 BTU/sq.ft. °F hr. will be given by:

$$h_a = \frac{1}{\frac{1}{300} - \left(\frac{1}{2000} + \frac{1}{1000}\right)} = 550 \text{ BTU/sq.ft. °F hr.}$$

$$\text{an error of } \frac{550 - 300}{550} = 45\%.$$

Similarly, for a U_a value of say 60 BTU/sq.ft. °F hr., the h_a value will be 66 BTU/sq.ft. °F hr., and error of 9%. Whilst the error introduced by using U_a values instead of h_a values is by no means negligible, it is consistent, and hence more like a correcting factor, the significance of which diminishes with decreasing U_a values.

The most logical definition of viscosity in terms of known measurements is in terms of the total solids content. It has been shown (21) that an exponential form of correlation often exists between the solids concentration and the viscosity of a colloidal solution. Data presented in Appendix I justify the assumption that over the range of values covered by these tests/

(1) See Marks L.S., "Mechanical Engineers Handbook" Macgraw-Hill, New York 1951. Pages 366 and 375

tests, the apparent viscosity of the concentrate is more dependent on its solids concentration than on its temperature. As an approximation we can therefore write:

$$\mu = e^{a_1 \times TS}$$

$$\text{or } U_L \propto \frac{P^{a_2} \times \Delta T_e^{a_3}}{e^{a_1 \times TS}}$$ where a_1, a_2 and a_3 are constants. TS = Total solids content.

From this $$\log \frac{U_a}{P^{a_2} \cdot \Delta T_e^{a_3}} \propto -(TS)$$

By trial and error the best numerical values for the coefficients (a_2) and (a_3) were found to be roughly $\frac{1}{2}$ and 1. The term $\frac{U_a}{P^{\frac{1}{2}} \cdot \Delta T_e}$ was calculated for all runs listed in Tables V and VI, and plotted as shown in Fig. 9. The corresponding figures for a number of runs made with water (TS = 0%) were obtained for the sake of comparison, and are shown in Table VIII and included in Fig. 9.

When allowance is made for the sweeping approximations - particularly in relation to viscosity - a correlation can be read into the curve of $\frac{U_a}{P^{\frac{1}{2}} \cdot \Delta T_e}$ vs. (TS) up to a TS value of about 30%. Beyond that the points are scattered in a seemingly random fashion between the ordinate values of 0.5 and 1.2

The fact that the best curve that can be drawn through the points between zero and 30% TS deviates from a straight line at low TS values, can be ascribed to the increasing weight of the steam-film and tube wall coefficients which are included in the overall U_a values. The scatter in the $\frac{U_a}{P^{\frac{1}{2}} \cdot \Delta T_e}$ values for pure water are probably due to variations with temperature of secondary parameters such as density, surface tension, etc.

The most interesting feature of Fig. 9 is, that the correlation which seems to hold for thin liquor breaks down when a certain viscosity is exceeded. It is remarkable that if the correlation did in fact extend into the regions of high TS - as shown dotted in Fig. 9 - the corresponding U_a values would become so low that evaporation would practically cease. It appears that it is only by virtue of the break in the accepted correlation of U_a and viscosity that commercial concentrations can successfully produce concentrate up to 60% TS. It is significant that at the critical solids concentration of 30%, the flow pattern in the non-boiling section of the heated tube would be expected to change from turbulent to laminar.

This can be illustrated by the following example. Referring to Fig. AI(1) in Appendix I, and Table IX in Chapter VI, it can be assumed that the

average/

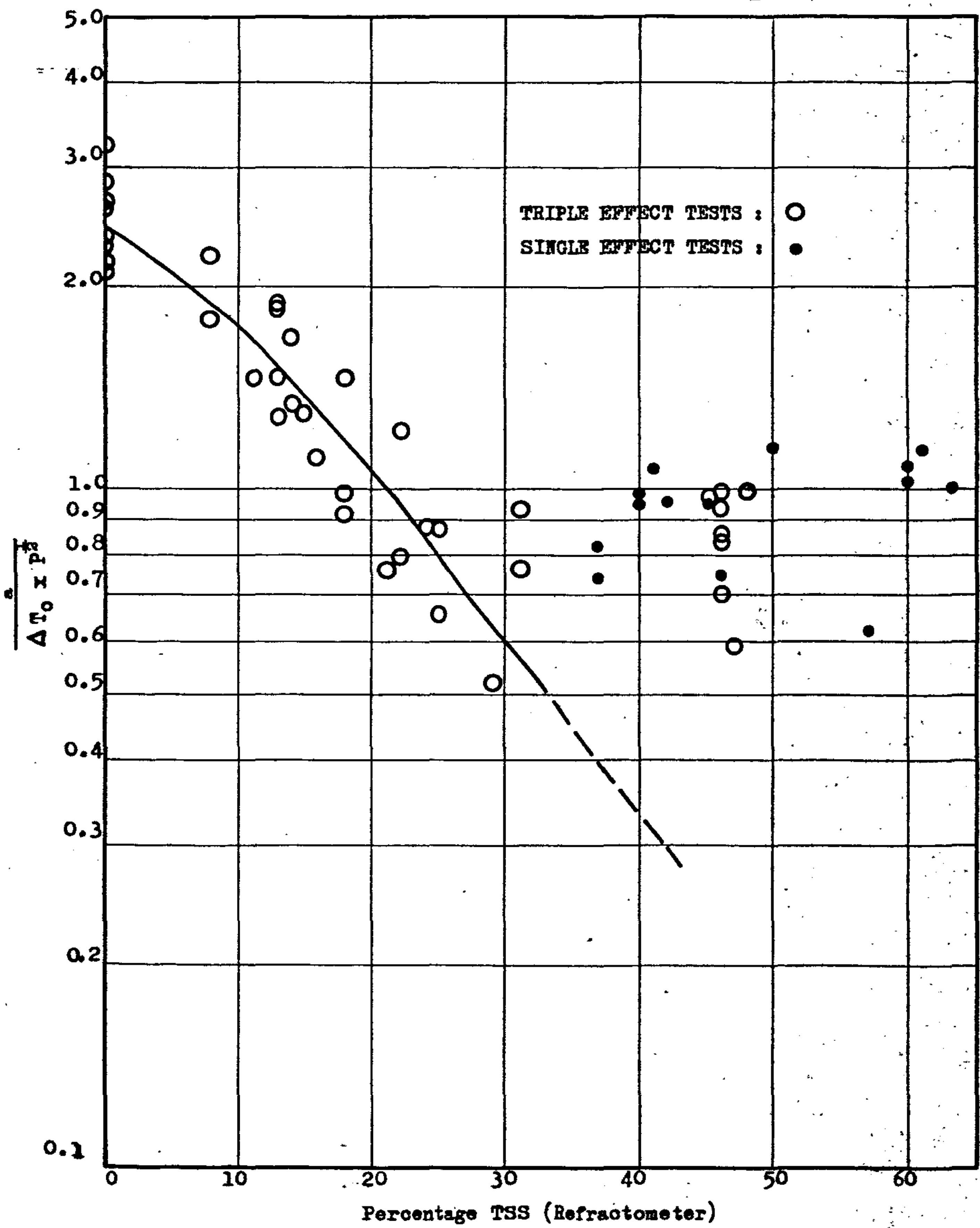

FIG. 9 : SUMMARY OF CONTROL TESTS

TABLE VIII

TEST SUMMARY : CONTROLS : WATER

TRIPLE EFFECT

Run No.	Effect	Temperatures °F			$P^{\frac{1}{3}}$ (P in lbs/sq. in.)	U_a BTU/sq. ft. °F.hr.	$\frac{U_a}{\Delta T_o \cdot P^{\frac{1}{3}}}$
		Jacket	Liquid	ΔT_o			
31	1st	277	234	43	4.7	478	2.37
	2nd	234	186	48	3.0	328	2.28
	3rd	186	115	71	1.2	225	2.64
33	1st	259	225	34	4.3	324	2.22
	2nd	225	195	30	3.2	277	2.88
	3rd	195	155	40	2.1	190	2.26
34	1st	258	224	34	4.3	304	2.08
	2nd	224	193	31	3.2	258	2.60
	3rd	193	161	32	2.2	229	3.25
35	1st	258	223	35	4.3	325	2.16
	2nd	223	189	34	3.0	268	2.63
	3rd	189	150	39	2.5	225	2.30

average viscosity of concentrate at 30% TS is about 10 cp, and the liquid velocity in the heated tube would be of the order of 1.0 ft/sec. Assuming the density to be 65 lbs/cubic ft

$$Re = \frac{0.115 \times 1 \times 65 \times 10^4}{10 \times 6.72} = \underline{1,100}$$

The extremely rapid rise of viscosity with solids concentration would be paralleled by an even sharper drop in the corresponding Reynolds Number, owing to the simultaneous slowing down of the circulation rate.

In spite of the scatter of about $\pm$ 20% of the experimental figures around the average values within the range of zero to 30% TS, and the absence of a recognisable correlation at higher TS values, Fig. 9 successfully combines all the data contained in Tables V, VI and VIII. It also serves to indicate the range of U_a values that may be expected under all practical operating conditions.

CHAPTER VI

EXPERIMENTS TO PROMOTE THE HEAT TRANSFER IN EXISTING PLANT

It is widely held, e.g. by Lyle (29), that the drop in the heat transfer coefficients that commonly occurs in the later effects of natural circulation evaporators is caused by the steep rise of the viscosity of the product with decreasing temperature and increasing solids concentration. It is believed that the increase in viscosity reduces the circulation rate, on which chiefly depends the rate of heat transfer. This reasoning suggests that if either the viscosity of the product is sufficiently reduced, or if a reasonable circulation rate can be maintained in spite of its high viscosity, better heat transfer coefficients will be achieved.

The application of this argument to existing stickwater concentrators resulted in the following alternatives being selected for testing:

(a) Reverse feeding

(b) Enzyme treatment of the stickwater

(c) Forced circulation

(d) Air-lift circulation.

A fifth alternative which appeared promising as a result of the study of the working principles of the apparatus outlined in Chapter VII, and which was tested after the main experimental work had been concluded, was :

(e) Throttled circulation.

This chapter contains details of all the experimental application of alternatives (a) to (e), and brief summaries of the results. A full discussion of the significance of the findings is difficult without an understanding of the operating mechanism of the apparatus, and is therefore deferred to Chapter VIII.

(a) Reverse feeding(instead of forward feeding).

The liquid lines of the three effects of the apparatus were reconnected to permit raw stickwater to be drawn into the third effect, and liquid from the third effect to be pumped into the second, and from the second into the first effect respectively. The steam lines were not changed, so that the feed now entered at the cold end, and the concentrate left at the hot end of the plant.

It was expected that the increase in the U_a value in the final effect (owing to the lower solids concentration of the liquid) would

more/

more than offset the decrease of the U_a value in the first effect (owing to the higher solids content of the concentrate).

The results, shown in Table IX, were disappointing. The effect of reverse feeding ranged from an increase of 13% to a decrease of 5% in the Gross U value as compared with the respective control runs. While reverse feeding may have possibilities under certain operating conditions, it did not appear the most promising solution and was not further investigated.

(b) Enzyme Treatment of Stickwater.

Interest in the use of enzymes to promote heat transfer under normal operating conditions was aroused during the investigation of tube blockages, described in Chapter III. The use of enzymes on a routine basis appeared particularly attractive in view of the possibility of using mascerated pilchard viscera as the source of enzymes.

A full description of a typical test with enzyme-treated stickwater, including all readings and calculations, will be found in Appendix III(a).

It is evident from Table X that the treatment of the stickwater with enzymes resulted in an increase of the gross U value (for triple effect operation) of from 3% to 37%. Both the highest and the lowest increases were obtained with homogenised pilchard gut. In view of its inconsistency as a thinning agent (see Appendix IV) it cannot be relied upon to increase plant capacity by a fixed amount. However, in circumstances where supplies of pilchard gut are available at low cost, e.g. in the neighbourhood of pilchard canneries, where the disposal of the gut sometimes presents a problem, the use of this material for treating stickwater may be a distinct advantage.

(c) Forced instead of Natural Circulation.

Oliver Lyle (30) states "The aim of good design in a pan or evaporator should be circulation, circulation and still more circulation". This, and similar statements extracted from the literature, suggested that a substantial improvement in heat transfer could be expected by the application of forced circulation to the third effect, which seemed to be the bottleneck in the performance of commercial plant.

It was gathered from the literature (31) that the lowest practical liquid velocity in forced circulation evaporators is about 6 ft./sec., and with a view to restricting pumping costs to a minimum, this was the velocity employed in the triple effect tests.

The results shown in Table XI indicate that forced circulation resulted in an increase of about 17.5% in the Gross U value. A significant feature of the triple effect tests is, that the application of forced circulation to the third effect gave rise to considerable changes in the heat transfer coefficients of the first and second effects. The causes

for/

T A B L E IX

EFFECT OF REVERSE FEEDING ON HEAT TRANSFER RATE

(TRIPLE EFFECT TESTS) (FIRST, SECOND AND THIRD EFFECTS DENOTED BY I, II and III)

	Run No.	Mode of Operation	TSS by Refractometer %				Temperature °F				Circulating Velocity Ft. per Second			U_a BTU/sq.ft.°F.Hr			U_G $\frac{BTU}{Sq.ft.°F.Hr}$	CHANGE IN U_G due to Reverse Feeding
			Raw Stick-water	I	II	III	Live Steam	I Liquid	II Liquid	III Liquid	I	II	III	I	II	III		
TEST A	45	Forw.	12	18	29	47	263	231	180	108	1.75	0.8	0.5	215	71	46	88	Control
	64	Rev.	12	46	24	16	274	222	174	98	not measured			126	110	43	84	- 5%
TEST B	26	Rev.	7	47	14	8	288	225	154	110	1.7	1.35	1.2	167	137	167	148	+ 1%
	27	Forw.	7	11	16	49	298	262	204	110	2.4	1.4	0.4	350	139	77	147	Control
	28	Rev.	7	47	16	9	287	224	152	110	1.5	1.3	0.9	179	197	198	166	+ 13%
TEST C	29	Rev.	9	48	18	12	286	230	158	102	1.8	0.9	0.6	179	122	115	130	- 3%
	30	Forw.	9	12	21	50	287	242	191	108	not measured	1.3	0.4	238	144	82	134	Control

T A B L E IX

EFFECT OF REVERSE FEEDING ON HEAT TRANSFER RATE

(TRIPLE EFFECT TESTS) (FIRST, SECOND AND THIRD EFFECTS DENOTED BY I, II and III)

	Run No.	Mode of Operation	TSS by Refractometer %				Temperature °F				Circulating Velocity Ft. per Second			U_a BTU/sq.ft.°F.Hr			U_G $\frac{\text{BTU}}{\text{Sq.ft.°F.Hr}}$	CHANGE IN U_G due to Reverse Feeding
			Raw Stick-water	I	II	III	Live Steam	I Liquid	II Liquid	III Liquid	I	II	III	I	II	III		
TEST A	45	Forw.	12	18	29	47	263	231	180	108	1.75	0.8	0.5	215	71	46	88	Control
TEST A	64	Rev.	12	46	24	16	274	222	174	98	not measured			126	110	43	84	- 5%
TEST B	26	Rev.	7	47	14	8	288	225	154	110	1.7	1.35	1.2	167	137	167	148	+ 1%
TEST B	27	Forw.	7	11	16	49	298	262	204	110	2.4	1.4	0.4	350	139	77	147	Control
TEST B	28	Rev.	7	47	16	9	287	224	152	110	1.5	1.3	0.9	179	197	198	166	+ 13%
TEST C	29	Rev.	9	48	18	12	286	230	158	102	1.8	0.9	0.6	179	122	115	130	- 3%
TEST C	30	Forw.	9	12	21	50	287	242	191	108	not measured	1.3	0.4	238	144	82	134	Control

EFFECT OF ENZYME TREATMENT ON HEAT TRANSFER RATE

(TRIPLE EFFECT TESTS) (First, Second and Third Effects denoted by I, II and III)

Test	Run No.	TREATMENT(1)	TSS by Refractometer %				Temperature °F				Apparent Viscosity by U-Tube Viscometer Centistokes (2)			U_a BTU/sq.ft.°F.Hr.			U_o BTU/Sq.ft.°F.Hr.	Change in U_o due to Enzyme Treatment
			Raw Stickwater	I	II	III	Live Steam	I Liquid	II Liquid	III Liquid	I	II	III	I	II	III		
TEST A	4	Untreated control	9	14	25	48	273	233	180	89	not	measured		250	126	72	126	Control
	5	10% Pilchard gut	9	13	19	47	272	233	187	93	not	measured		282	167	77	144	+ 14%
TEST B	6	Untr. Control	8	11	18	45	273	235	185	88	0.55	1.6	.82	266	139	74	127	Control
	7	10% Pilchard gut	9	11	18	45	273	235	190	88	0.5	0.75	20	289	171	77	142	+ 12%
TEST C	10	Untr. Control	8	13	24	46	270	232	184	91	not	measured		256	122	59	116	Control
	11	5% Pilchard gut	8	12	20	46	272	230	191	94	not	measured		287	216	82	156	+ 34%
	12	10% Pilchard gut	9	12	20	47	272	233	203	107	not	measured		295	259	75	159	+ 37%
TEST D	13	Untr. Control	9	13	18	46	277	244	194	99	0.6	1.3	105	334	146	76	143	Control
	14	0.2% Commercial Enzyme prep.	9	13	21	47	274	238	199	103	0.5	1.0	46	316	196	74	153	+ 7%
	15	5% Pilchard gut	9	13	17	48	274	234	203	99	0.5	0.8	70	330	280	79	170	+ 19%
											Liquid Velocity ft/sec.							
TEST E	47	Untr. Control	8	15	22	46	277	236	194	106	1.95	1.7	0.6	256	164	68	137	Control
	65	0.3% Com. Enzyme	8	12	20	47	273	234	197	99	1.7	1.4	1.2	324	244	88	173	+ 26%
	66	5% Pilchard Gut	8	13	22	47	273	234	188	99	2.05	1.4	1.25	280	158	73	141	+ 3%

(1) For description of treatment see Appendix IV. (2) See Appendix I.

TABLE XI

EFFECT OF FORCED CIRCULATION, IN THIRD EFFECT, ON HEAT TRANSFER RATE

(TRIPLE EFFECT TESTS) (First, Second and Third Effects denoted by I, II and III)

	Run No.	Mode of Operation	TSS by Refractometer %				Temperature, °F				Apparent Viscosity by U-Tube Viscometer Centistokes			U_a BTU/sq.ft.°F.Hr			U_G BTU/Sq.ft.°F.Hr	Change in U_G due to Forced Circulation in 3rd Effect
			Raw Stick-water	I	II	III	Live Steam	I Liquid	II Liquid	III Liquid	I	II	III	I	II	III		
TEST A	8	Forced circ. III 6 ft/sec	10	15	24	46	274	234	179	98	0.9	2.1	130	261	115	76	129	+ 18%
TEST A	9	Control - natural circ.	10	16	25	46	273	232	176	91	0.9	2.2	150	210	98	67	109	Control
TEST B	46	Control - natural circ.	9	13	21	46	275	242	190	94	0.66	1.3	76	318	123	66	128	Control
TEST B	67	Forced circ. III 6 ft/sec	9	13	19	47	274	230	189	96	0.68	1.3	74	271	182	79	150	+ 17%

EFFECT OF FORCED CIRCULATION ON HEAT TRANSFER RATE (SINGLE EFFECT TESTS)

	Run No.	Circulating Velocity ft/sec.	TSS by Refractometer %	Temperature °F				U_a BTU/sq.ft.°F.Hr	Change in U_a due to forced circulation
				Live Steam	Tube Wall	Liquid	ΔT Wall-Liquid		
TEST C	57	Control - natural circ.	50	213	209	121	88	139	Control
TEST C	68	1.5	50	213	209	123	86	118	- 15%
TEST C	69	3.0	50	213	209	123	86	122	- 10%
TEST C	70	4.0	50	213	209	122	87	126	- 7%

for this are uncertain, but are probably related to the unstable balance of U_a values, temperature gradients and solids concentrations which is typical of multi-effect apparatus, and which is largely beyond the control of the experimenter.

The contention that low circulation rates are ineffective was confirmed by the single effect tests (c) shown in Table XI. It was found that for circulation rates within the range of 1.5 ft/sec. to 4.0 ft/sec. the U_a values obtained with forced circulation were from 7% to 15% lower than for the control (natural circulation), but showed a tendency to rise with the liquid velocity.

(d) Air-lift Circulation.

It was discovered that U_a and U_G values could be significantly increased by injecting small quantities of air into the liquid entering the heated tube. A rough preliminary estimate indicated that the improvements in heat transfer achieved in this manner probably involved a considerably lower pumping cost than mechanical forced circulation. As the application of air-lift circulation to evaporation appeared not to have been investigated before, the tests were extended to cover a fairly wide field. Triple effect tests with air-lift circulation applied to the third effect were carried out both with stickwater and with pure water. A series of single effect tests with water as well as stickwater concentrate were performed to estimate the benefit derived from different rates of air injection. A small number of tests were finally made using a Newtonian fluid of high viscosity (solutions of corn syrup) to confirm the favourable results obtained with stickwater concentrate.

Specimen sets of readings and calculations will be found in Appendix III (b) and (c), and the results of all the air-lift tests are summarised in Tables XII to XV.

With from 2.0 to 4.0 litres of air measured at ambient temperature and pressure being injected into the base of the heated tube of the third effect, the overall improvement in heat transfer when operating in triple effect ranged from 17% to 47%. The assessment of the effectiveness of air-lifting in triple effect tests is complicated by the fact that a change in the U_a values of one (e.g. the third) effect brings about a change in the temperature gradients, the solids concentrations and the U_a values in the other effects.

The single effect tests with stickwater concentrate were purposely performed at temperatures and solids concentrations commonly found in the final effects of commercial plant.

The results of these tests (Table XIII) display some features which

may/

TABLE XII

EFFECT OF AIR-LIFT IN THIRD EFFECT ON HEAT TRANSFER RATE

(TRIPLE EFFECT TESTS) (First, Second and Third Effects denoted by I, II and III).

	Run No.	Air admitted Litres/min	TSS by Refractometer %				Temperature °F				Liquid Velocity Ft/Sec			U_a BTU/sq.ft.°F.Hr			U_o	Change in U_o due to Air-Lift
			Raw Stick-water	I	II	III	Live Steam	I Liquid	II Liquid	III Liquid	I	II	III	I	II	III	BTU / Sq.ft.°F.Hr	
TEST A	71	Control	9.5	12	18	46	269	237	206	134		not measured		271	184	69	144	Control
	72	3.0 (0.45 lbs/hr)	9.5	13	19	45	267	238	180	131	"		"	307	117	141	168	+ 17%
TEST B	73	3.0	8.5	12	21	46	260	226	163	108	"		"	310	107	127	159	+ 29%
	74	Control	9.0	12	20	46	257	231	188	107	"		"	305	131	58	123	Control
TEST C	75	Control	9.0	14	23	45	261	236	185	105	"		"	338	100	58	114	Control
	76	4.0 (0.6 lbs/hr)	9.0	11	19	47	261	228	171	105	"		"	324	134	120	168	+ 47%
TEST D	20	Control	9.0	14	24	49	265	241	197	103	"		"	420	153	64	138	Control
	21	2.0 (0.3 lbs/hr)	9.0	13	21	51	265	233	185	118	"		"	336	160	110	174	+ 26%
TEST E	22	Control	9.0	13	24	50	274	246	192	97	2.5	1.4	-	390	117	71	129	Control
	23	2.0	9.0	13	23	52	274	246	192	106	2.0	1.5	1.1	435	152	98	165	+ 28%
TEST F	24	Control	9.0	15	23	48	274	251	200	127	1.3	1.3	0.5	324	93	50	105	Control
	25	2.0	8.5	12	21	50	274	245	203	132	2.3	1.8	1.8	414	195	101	153	+ 46%

may represent genuine trends, but may also be due to freakish behaviour of the concentrate. It was noted for instance that the stickwater samples used for tests A and F appeared to be thixotropic, as they had to be vigorously stirred to render them free-flowing.

The results of five of the tests have been plotted in Fig. 10, and a considerable difference is evident in the shape of the individual curves. For comparison the results of test D (Table XV) with corn syrup have been included in Fig. 10. Note that the tube wall as well as steam jacket temperatures were measured in all runs from Table XIII onwards, but as they are close to and of less interest for comparative purposes than the U_a values, the corresponding h_a values have been omitted except in Table XIV. Where calculated, the h_a values are always based on the average of the temperatures measured by the four thermocouples equidistantly spaced along the length of the tube.

In spite of inconsistencies, the results of the tests showed the following trends:

(i) provided a sufficient volume of air was injected, the U_a values could always be raised by from 100% to 139% above the U_a values measured without air injection;

(ii) the volume of air required to raise a given U_a value by a given amount increases with the solids content of the concentrate, e.g. in test G, using concentrate of 42% TSS, 1.1 litres/min were sufficient to increase the U_a value from 119 BTU/sq.ft. °F. Hr. to 197 BTU/sq.ft. °F. Hr., whereas in test E, using concentrate of 60% TSS, 10 litres/min. were required to raise the U_a value from 137 BTU/sq.ft. °F. Hr. to 172 BTU/sq.ft. °F. Hr.;

(iii) with one exception (test F), the curves of U_a vs. air injection all exhibit one or more changes of slope at some critical air injection rate; and this tendency was also observed in the tests with corn syrup solutions. A fairly steep initial rise is generally followed by a region in which U_a rises steadily but less sharply with the air injection rate, until in a few cases the curves again rise more steeply at injection rates in excess of about 10 litres/min.

Included in Table XIII is a column headed "Pump I.H.P.". The calculation and the significance of this Index of power consumption will be discussed in Chapter VIII.

The results of the tests with water (Table XIV) follow a clearer trend. In order to correlate them, the factor $\dfrac{h_a}{\Delta T \times P^{\frac{1}{3}}}$ was calculated for all runs and plotted against the air injection rate as shown in Fig. 11. It is noteworthy that

(i) the curves for all tests converge towards a single point at

zero/

TABLE XIII

EFFECT OF AIR-LIFT ON HEAT TRANSFER RATE (SINGLE EFFECT TESTS)

Run No.	Air admitted Litres/min	TSS by Refractometer %	Temperature °F			U_a BTU/sq.ft.°F. Hr.	Change in U_a due to Air-Lift	Pump I.H.P.
			Live Steam	Tube Wall	Liquid			
54	Control	57	213	212	124	77	Control	-
77	0.53	59	213	212	124	61	- 21%	0.0015
78	1.0	61	214	212	124	60	- 22%	0.0027
79	3.55	61.5	213	211	123	83	+ 8%	0.0098
80	10.0	57.5	213	207	123	184	+ 139%	0.0267
81	5.1	57.5	213	209	122	178	+ 131%	0.0138
82	3.8	58.5	213	210	124	121	+ 57%	0.0102
83	2.05	59.5	213	211	125	89	+ 16%	0.0054
55	Control	41	213	210	121	129	Control	-
84	2.0	40	213	204	121	189	+ 52%	0.0054
85	3.88	42	212	203	124	207	+ 67%	0.0104
86	1.0	42	213	208	122	171	+ 38%	0.0027
87	0.5	40	213	207	120	163	+ 31%	0.0015
56	Control	40	213	209	120	118	Control	-
88	5.3	68	213	212	126	48	+ 14%	0.0144
89	10.0	68	213	212	124	47	+ 12%	0.0271
90	19.0	66	213	212	120	84	+ 100%	0.0525
91	Control	66	213	213	123	42	Control	-

(continued)

Table XIII (continued)

	Run No.	Air admitted Litres/min	TSS by Refracto-meter	Temperature °F			U_a BTU/sq.ft.°F.Hr.	Change in U_a due to Air-Lift	Pump I.H.P.
				Live Steam	Tube Wall	Liquid			
TEST D	57	Control	50	213	209	121	139	Control	–
	92	5.0	50	213	208	124	189	+ 36%	0.0129
	93	10.0	50	212	205	123	202	+ 45%	0.0267
	94	19.0	50	213	204	127	222	+ 60%	0.0473
TEST E	58	Control	61	213	209	122	137	Control	–
	95	5.0	61	213	209	122	159	+ 17%	0.0148
	96	10.0	59	213	208	122	172	+ 26%	0.0278
	97	15.0	59	213	206	125	196	+44%	0.0400
	98	5.0	60	212	207	118	164	+ 21%	0.0154
	59	Control	60	213	209	118	135	Control	–
	99	1.1	60	213	208	116	148	+ 9%	0.0036
TEST F	100	5.0	63	213	209	117	131	+ 7%	0.0163
	60	Control	63	213	210	118	122	Control	–
	101	19.0	63	213	209	122	149	+ 22%	0.0542
TEST G	102	5.0	42	212	201	122	234	+ 97%	0.0131
	103	2.0	42	213	205	125	203	+ 70%	0.0056
	61	Control	42	213	209	124	119	Control	–
	104	1.1	42	213	207	126	197	+ 65%	0.0029
TEST H	105	1.1	40	213	206	124	200	+ 69%	0.0029
	62	Control	40	213	209	124	118	Control	–

(continued)

Table XIII (continued)

	Run. No.	Air admitted Litres/min	TSS by Refractometer %	Temperature °F Live Steam	Tube Wall	Liquid	U_a BTU/sq.ft.°F.Hr.	Change in U_a due to Air-Lift	Pump I.H.P.
TEST H	106	5.0	62.5	213	208	116	169	+ 32%	0.0131
	63	Control	62.5	213	210	118	128	Control	-
TEST J	107	1.0	47	182	-	99	104	+ 42%	0.0038
	50	Control	46	186	-	100	73	Control	-
TEST K	108	1.25	38	180	-	108	145	+ 123%	0.0044
	51	Control	37	179	-	107	65	Control	-
TEST L	52	Control	37	181	-	108	59	Control	-
	109	1.25	38	180	-	110	128	+ 117%	0.0042
TEST M	110	1.25	46	180	-	106	93	+ 20%	0.0049
	53	Control	45	180	-	105	78	Control	-

TABLE XIV

EFFECT OF AIR-LIFT ON HEAT TRANSFER RATE (SINGLE EFFECT TESTS) WATER

un o.	Air admitted Litres /min	Temperature °F			Liquid Velocity Ft/sec	h_a BTU/sq.ft.°F.Hr.	$P^{\frac{1}{2}}$	$\frac{h_a}{\Delta T \times P^{\frac{1}{2}}}$
		Tube Wall	Liquid	Δ T				
1	Control	207	157	50	1.75	410	2.1	3.9
2	Control	207	157	50	1.75	390	2.1	3.7
3	0.36	208	158	50	2.05	500	2.1	4.8
4	0.96	209	155	54	2.15	540	2.1	4.8
5	0.34	105	96	9	2.28	400	0.9	49.5
6	Control	105	92	13	0.2 (1)	47	0.85	4.25
7	0.53	106	96	10	2.57	440	0.9	49
8	0.33	105	97	8	2.22	370	0.95	48.5
9	0.985	105	97	8	2.90	440	0.95	58
0	0.11	106	97	9	1.88	330	0.95	38.5
1	0.94	123	104	19	3.12	490	1.05	24.5
2	0.29	123	103	20	2.35	480	1.0	24
3	0.57	124	104	20	2.685	510	1.05	24.5
[illegible]	Control	122	101	21	? (2)	80	1.0	3.8
[illegible]	0.11	123	104	19	1.885	370	1.05	18.5
[illegible]	0.28	123	103	20	2.37	460	1.0	23
[illegible]	0.105	153	111	42	1.415	390	1.15	8.1
[illegible]	0.955	153	119	34	2.9	550	1.3	12.5
[illegible]	0.555	153	120	33	2.53	560	1.3	13
[illegible]	0.31	153	120	33	2.07	510	1.3	12
[illegible]	0.10	153	120	33	1.72	430	1.3	10
[illegible]	Control	152	120	32	0.94	210	1.3	5.0

(1) Estimated

(2) Too low to be measured

P 3

TABLE XIV

EFFECT OF AIR-LIFT ON HEAT TRANSFER RATE (SINGLE EFFECT TESTS) WATER

Test	Run No.	Air admitted Litres /min	Temperature °F Tube Wall	Liquid	Δ T	Liquid Velocity Ft/sec	h_a BTU/sq.ft.°F.Hr.	$P^{\frac{1}{2}}$	$\frac{h_a}{\Delta T \times P^{\frac{1}{2}}}$
TEST A	111	Control	207	157	50	1.75	410	2.1	3.9
	112	Control	207	157	50	1.75	390	2.1	3.7
	113	0.36	208	158	50	2.05	500	2.1	4.8
	114	0.96	209	155	54	2.15	540	2.1	4.8
TEST B	115	0.34	105	96	9	2.28	400	0.9	49.5
	116	Control	105	92	13	0.2 (1)	47	0.85	4.25
	117	0.53	106	96	10	2.57	440	0.9	49
	118	0.33	105	97	8	2.22	370	0.95	48.5
	119	0.985	105	97	8	2.90	440	0.95	58
	120	0.11	106	97	9	1.88	330	0.95	38.5
TEST C	121	0.94	123	104	19	3.12	490	1.05	24.5
	122	0.29	123	103	20	2.35	480	1.0	24
	123	0.57	124	104	20	2.685	510	1.05	24.5
	124	Control	122	101	21	? (2)	80	1.0	3.8
	125	0.11	123	104	19	1.885	370	1.05	18.5
	126	0.28	123	103	20	2.37	460	1.0	23
TEST D	127	0.105	153	111	42	1.415	390	1.15	8.1
	128	0.955	153	119	34	2.9	550	1.3	12.5
	129	0.555	153	120	33	2.53	560	1.3	13
	130	0.31	153	120	33	2.07	510	1.3	12
	131	0.10	153	120	33	1.72	430	1.3	10
	132	Control	152	120	32	0.94	210	1.3	5.0

(1) Estimated

(2) Too low to be measured

TABLE XV

EFFECT OF AIR-LIFT ON HEAT TRANSFER RATE (SINGLE EFFECT TESTS) CORN SYRUP

	Run No.	Air admitted Litres/min	TSS by Refractometer (%)	Temperature °F			U_a BTU/sq.ft.°F.Hr.	Change in U_a due to Air-Lift	Approximate Viscosity by U-Tube Viscometer. Centistokes
				Live Steam	Tube Wall	Liquid			
TEST A	43	Control	75	213	210	120	99	Control	250
	133	1.2	77	212	208	120	135	+ 36%	400
TEST B	134	5.0	79	212	207	120	142	+ 51%	750
	135	Control	79	212	209	119	94	Control	750
	136	1.2	80	213	210	120	100	+ 6%	1000
TEST C	137	Control	66.5	213	210	118	84	Control	45
	138	1.0	66.5	213	209	118	147	+ 71%	45
	139	Control	66.5	213	209	117	88	Control	45
	140	0.5	67.5	213	208	119	119	+ 38%	55
TEST D	44	Control	75.5	213	209	119	99	Control	270
	141	1.0	76.0	213	205	118	127	+ 28%	310
	142	5.0	75.5	213	207	120	147	+ 49%	270
	143	10.0	76.0	213	206	121	161	+ 63%	310

TABLE XV

EFFECT OF AIR-LIFT ON HEAT TRANSFER RATE (SINGLE EFFECT TESTS) CORN SYRUP

	Run No.	Air admitted Litres/min	TSS by Refractometer (%)	Temperature °F			U_a BTU/sq.ft.°F.Hr.	Change in U_a due to Air-Lift	Approximate Viscosity by U-Tube Viscometer. Centistokes
				Live Steam	Tube Wall	Liquid			
TEST A	43	Control	75	213	210	120	99	Control	250
	133	1.2	77	212	208	120	135	+ 36%	400
TEST B	134	5.0	79	212	207	120	142	+ 51%	750
	135	Control	79	212	209	119	94	Control	750
	136	1.2	80	213	210	120	100	+ 6%	1000
TEST C	137	Control	66.5	213	210	118	84	Control	45
	138	1.0	66.5	213	209	118	147	+ 71%	45
	139	Control	66.5	213	209	117	88	Control	45
	140	0.5	67.5	213	208	119	119	+ 38%	55
TEST D	44	Control	75.5	213	209	119	99	Control	270
	141	1.0	76.0	213	205	118	127	+ 28%	310
	142	5.0	75.5	213	207	120	147	+ 49%	270
	143	10.0	76.0	213	206	121	161	+ 63%	310

NOTE: Stickwater was used for all tests except one with Corn Syrup for comparison

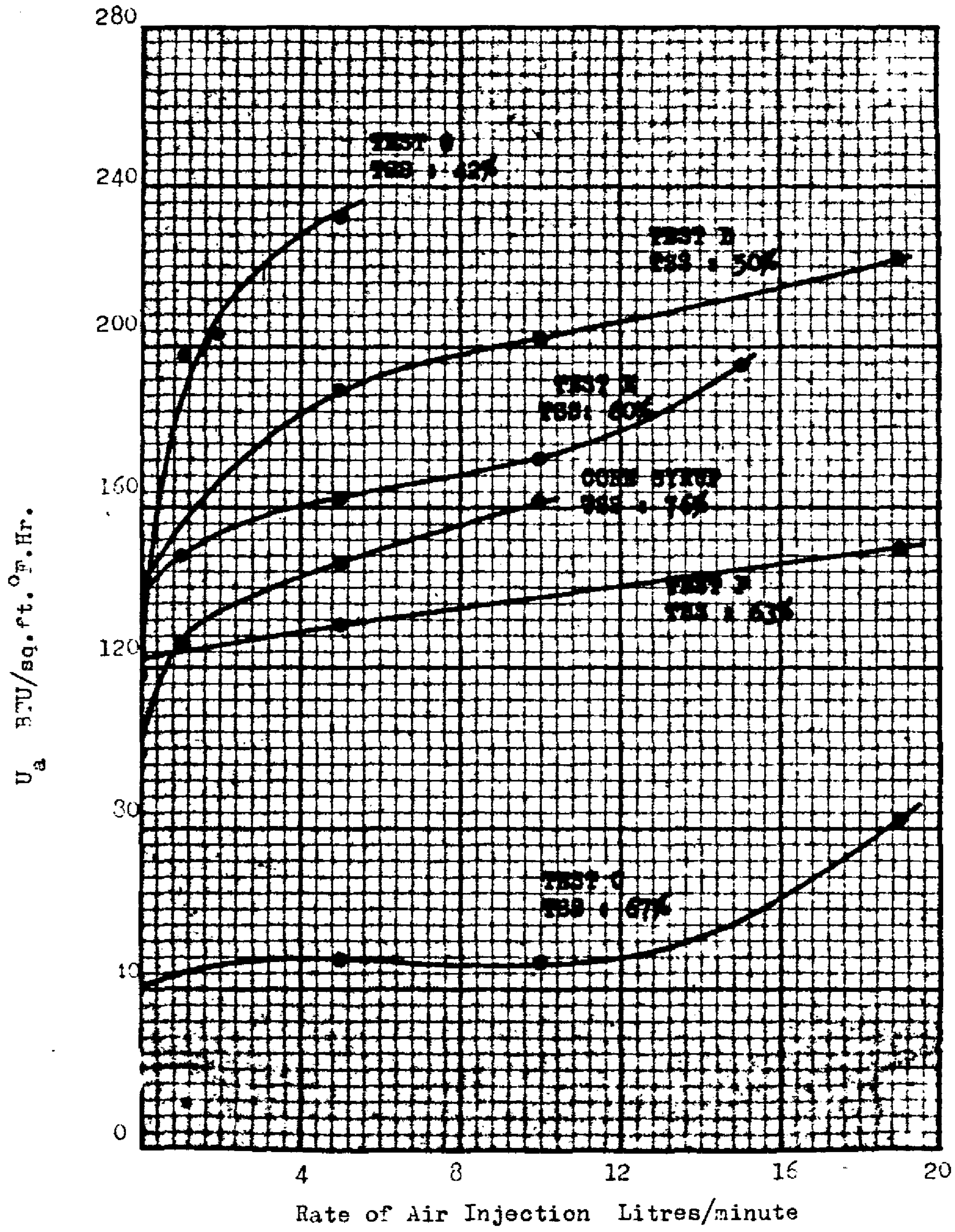

FIG. 10 : EFFECT OF AIR INJECTION ON U_a VALUES

TABLE XVI

EFFECT OF FLOW THROTTLING ON HEAT TRANSFER RATE (SINGLE EFFECT TESTS)

	Run No.	Working Fluid	TSS by Refractometer (%)	Estimated Liquid Velocity Ft/sec	Temperature °F			Approximate Viscosity Centistokes	U_a BTU/sq.ft.°F.Hr.	Change in U_a due to Throttling (1)
					Live Steam	Tube Wall	Liquid			
TEST A	144	S	46	0.42	213	209	116	N F	159	+ 21%
TEST A	145	T	46		213	209	118	O O	134	Control
TEST A	146	I	46	0.22	213	209	119	T R	174	+ 33%
TEST A	147	C C	46	0.11	213	210	117		169	+ 29%
TEST A	148	K O	46.5	0.35	213	210	119	S	159	+ 21%
TEST A	149	W N	47		213	208	119	T	128	Control
TEST B	150	A C	60		213	209	120	M I	139	Control
TEST B	151	T E	60	0.20	213	210	119	E C	137	+ 5%
TEST B	152	E N	60		213	209	121	A K	130	Control
TEST B	153	R T	60	0.32	213	209	120	S W	156	+ 20%
TEST B	154	R	60	0.42	214	210	120	U A	140	+ 8%
TEST B	155	A	60	0.38	213	209	120	R T	146	+ 12%
TEST B	156	T	60	0.27	213	210	120	E E	117	- 10%
TEST B	157	E	60		213	210	120	D R	122	Control

(1) Based on Average Control U_a value in each test.

(continued)

Table XVI (contd.)

	Run No.	Working Fluid	TSS by Refractometer (%)	Estimated Liquid Velocity Ft/sec	Temperature °F Live Steam	Temperature °F Tube Wall	Temperature °F Liquid	Approximate Viscosity Centistokes	U_a BTU/sq.ft.°F.Hr.	Change in U_a due to Throttling (1)
	158	Stickwater concentrate	20		213	213	176	not measured	67	Control
TEST C	159	- do -	20	0.1	214	213	173	- do -	123	+ 84%
	160	- do -	20	0.22	213	213	174	- do -	87	+ 30%
	39	Water	0		214	208	175	0.37	356	Control
TEST D	161	Water	0	0.1	214	208	174	0.37	451	+ 26%
	40	Water	0		214	209	174	0.37	362	Control
	162	Sucrose	65.5		213	208	120	25	83	Control
	163	Sucrose	66.5	0.057	213	208	119	28	107	+ 29%
	164	Sucrose	67.5	0.113	213	205	119	34	147	+ 77%
TEST E	165	Sucrose	67	0.23	213	208	117	32	166	+ 100%
	166	Sucrose	67.5	0.45	213	207	121	34	152	+ 83%
	167	Sucrose	67	0.68	213	207	116	32	128	+ 54%

(1) Based on Average Control U_a value in each test

following trends can be observed:

(i) under any given set of operating conditions the increase in U_a values achieved by throttling passes through a peak at some intermediate liquid velocity. This is brought out clearly in test E using a 65.5–67.5% sucrose solution. The tests with stickwater concentrate suffer from the habitual irregularity in the behaviour of the working fluid;

(ii) the maximum improvements in U_a values observed in the tests with stickwater decrease with solids contents and range from 84% (for concentrate of 20% TSS) to 20% (for concentrate of 60% TSS).

Compared with accelerated circulation (either by mechanical or by air-lift pumping), throttling appears to have two advantages :

(a) it can be applied to all effects instead of only to the final effect as air-lifting; and

(b) theoretically it acts as a source rather than a consumer of mechanically energy.

CHAPTER VII

THE MECHANISM OF HEAT TRANSFER IN NATURAL CIRCULATION EVAPORATORS

Natural circulation evaporators are amongst the most useful elements of process engineering. Requiring no circulating pumps or special controls, they represent the most economical means of evaporating large volumes of water from a wide range of mother liquids. But while their study covers an extremely wide field, all published information appears to be largely empirical, or related to specific products.

A searching investigation into the heat transfer to liquids flowing upwards in vertical tubes was carried out by W.L. Badger and collaborators (17; 18; 22). The existence of two fundamentally different types of heat transfer zones was suggested; viz. a "non-boiling" lower, and a "boiling" upper tube region. Empirical expressions for heat transfer coefficients applying to each of the two regions were derived experimentally, but no method was suggested for estimating an overall or average coefficient for the tube as a whole.

A more recent and systematic study was made by E. Kirschbaum and collaborators (20) of the most important factors which control heat transfer in natural circulation evaporators. The variables which were investigated in detail were :

the temperature gradient between tube wall and bulk of liquid;
the bulk temperature of the liquid;
the "head", or apparent liquid level in the heated tube;
the tube length and diameter; and
the viscosity of the working fluid.

The outstanding achievement of this programme was the evolution of a dimensionless correlation linking the average tube wall heat transfer coefficient and the most important properties of the working fluid and operating conditions:

$$N_v = \underline{a} \times R_v \times S^{\frac{1}{2}} \times Z^{\frac{1}{4}}$$

or

$$\frac{h_a \times \sigma}{k \times P} = \underline{a} \times \frac{\Delta T \times C}{\lambda} \times \left(\frac{\rho_L}{1000\,\rho_G}\right)^{\frac{1}{2}} \times \left(\frac{\mu_w}{\mu}\right)^{\frac{1}{4}}$$

where h_a = Liquid film coefficient of Heat Transfer
σ = Surface tension
P = Pressure in plenum
λ = Latent Heat
k = Thermal conductivity of working fluid
ρ_L = Density of Liquid
ρ_G = Density of Vapour
μ_w = Viscosity of Water
μ = Viscosity of Working Fluid
C = Specific heat of Working Fluid
ΔT = Temperature gradient between Tube wall and Liquid

a/

<u>a</u> - a coefficient determined by the apparent level of the liquid in the heated tube. The numerical value of <u>a</u> can only be determined experimentally.

The correlation by this expression of the results of a number of tests with distilled water and sugar solutions, using a tube with an internal diameter of 40 mm. and 4 m long, was reasonably good. The expression does not reflect a distinction between a boiling and non-boiling section. Its application to circumstances other than those for which it was developed can only be confirmed by further experiment, and referring to the tests described in Chapters V and VI, it fails to explain the trend of heat transfer coefficients at high viscosities.

In discussing the performance of commercial plant (Chapter V), it was noted that when the stickwater exceeds roughly 30% TSS, the U_a values appear strangely insensitive to further increases in solids concentration even up to 60%, and may even show a slight rise. A similar phenomenon was observed in some of the tests with corn syrup; in run No. 44 (Table XV), for instance, the solution at a TSS of 75.5% and an estimated viscosity of 270 centistokes gave a U_a value of 99 BTU/sq.ft. °F. Hr., whilst at a TSS of 66.5% and an estimated viscosity of 45 centistokes the U_a value was only 86 BTU/sq.ft. °F. Hr.; a <u>decrease</u> of 15%, whereas by Kirschbaum's formula one would have expected an <u>increase</u> of at least

$$\left\{\left(\frac{270}{45}\right)^{\frac{1}{4}} - 1\right\} \times 100 = 56\%$$

The highest viscosity of any of the working fluids tested by Kirschbaum was about 4.0 cp. The circulation velocity was not measured in Kirschbaum's tests, but judging by the results of the tests reported in Table XIV, a figure of say 1.0 ft/sec. appears a conservative estimate.

The Reynolds Number in the non-boiling region of a tube of 40 mm I.D. would thus be of the order of $\frac{1 \times 0.131 \times 75 \times 10^4}{4 \times 6.72}$ or say 3,600. In view of the unsettled conditions in the non-boiling section, the flow would almost certainly be turbulent. When, however, the working fluid reaches a viscosity of 270 centistokes as in Run No. 44, with corn syrup, the flow - even without allowing for a lower circulating velocity - can only be laminar. When working with a pseudo plastic and thixotropic fluid like stickwater concentrate, laminar flow (or its rheological equivalent) is likely to be well established at much lower apparent viscosities.

An association apparently exists between the inconsistent trend of U_a values at high viscosities and the development of laminar flow in the non-boiling section of the heated tube. It is possible to derive such correlation in terms of the proved and accepted expressions for fluid flow and heat transfer in tubes.

In the simplest form of natural circulation evaporator, liquid flows down an unheated "return tube" and enters the heated tube at approximately the same temperature as the bulk in the header drum. Circulation is maintained by the hydraulic imbalance between the liquid in the return tube and the liquid and vapour mixture in the heated tube.

Evaporation cannot commence at the point at which the liquid enters the heated tube, as it must first be heated to the saturation temperature equivalent to the pressure in the plenum space in the header drum <u>plus</u> the pressure due to the hydraulic head in the heated tube. As the liquid rises in the heated tube, its temperature rises and the hydraulic head diminishes until at a certain level the appropriate saturation temperature is reached. Boiling commences with the practically instantaneous "flashing" of part of the superheated liquid on entering a region of lower pressure. Only the upper part of the tube is given to boiling in its narrower sense.

It has been reported by investigators such as Boarts et al (17) and Groothuis and Hendal (23) that heat transfer in the lower (non-boiling) section of the heated tube follows the relationships for forced convection:

$$Nu = 0.027\, Re^{0.8} \times Pr^{0.4}\ ^{*} \qquad \ldots\ldots\ldots\ldots \qquad (i)$$

and

$$Nu = 2.0 \left(\frac{wc}{KL}\right)^{\frac{1}{3}} \qquad \ldots\ldots\ldots\ldots \qquad (ii)$$

depending on whether the flow is turbulent or laminar.

Heat transfer in the upper (boiling) section has been less clearly defined. Most correlations for "Boiling On Submerged Surfaces" regard the movement of the liquid relative to the heated surface as due to a combination of free convection and the pumping action brought about by the formation and the collapse of the nucleate vapour bubbles.

When boiling occurs inside a vertical tube, the uni-directional flow of the liquid-vapour mixture provides a third mixing force, which may obscure the effect of the other two. As a result, the quantitative prediction of heat transfer coefficients has not yet been perfected.

Fortunately, the present purpose requires no more than a knowledge of the broad, qualitative correlations which have been confirmed by experts in the field. It is generally agreed that the heat transfer coefficients to liquids boiling in tubes are :

a) considerably higher than the corresponding non-boiling coefficients. Handbooks such as Perry's (24) quote U_a values for condensing steam to liquids of 50 to 200 BTU/sq.ft. °F. Hr., but 300 to 800 BTU/sq.ft. °F. Hr. for condensing steam to boiling liquids;
b) not subject to radical change from one level to the next within the boiling region. This is exemplified by Kirschbaum's correlation ($N_v = \underline{a} \times R_v \times S^{\frac{1}{2}} \times Z^{\frac{1}{4}}$), which contains no factor which would be expected to change by more than say 20% between tube inlet and outlet; and
c) relatively independent of viscosity. Insinger and Bliss (25)

found/

* See Note (1) on p. 49

found that boiling on the <u>outside</u> of tubes was completely unaffected by viscosity. Even Kirschbaum's viscosity correction $\left(\frac{\mu_w}{\mu}\right)^{\frac{1}{4}}$ may be on the high side, as it reflects the effect of viscosity not only on boiling, but also on the relative extent of the boiling regions. A good compromise appears to be $\left(\frac{\mu_w}{\mu}\right)^{0.125}$.

There appears to be no agreement about the effect of the liquid temperature and the temperature gradient on the heat transfer coefficients to liquids boiling in tubes (as opposed to "submerged surfaces"). The liquid temperature is often represented by the corresponding vapour pressure, or specific vapour volume, and according to Kirschbaum (20) $h_a \propto \frac{\Delta T}{P} \times \left(\frac{V_G}{V_L}\right)^{\frac{1}{3}}$

However, Strosbe et al (22) suggest $h_a \propto \frac{V_G^{0.1}}{T^{0.13}}$

In industrial evaporators ΔT and V_G tend to move in the same direction, i.e. large temperature gradients in the final effects are generally associated with low plenum pressures or large specific volumes. As moreover variations of ΔT, V_G and P within the practical range are of a smaller order than variations of μ, the heat transfer coefficient in the boiling section will be for the present purpose represented by :

$$(\text{a constant}) \times \left(\frac{\mu_w}{\mu}\right)^{0.125}$$

An average heat transfer coefficient in terms of the individual coefficients for the boiling and the non-boiling sections is defined by :

$$h_a = \frac{h_1 \times L + h_2 (L_o - L)}{L_o} \qquad \ldots\ldots \qquad \text{(iii)}$$

where : h_a = average coefficient for whole tube — BTU/sq.ft. °F. Hr.
h_1 = coefficient for non-boiling section — BTU/sq.ft. °F. Hr.
h_2 = coefficient for boiling section — BTU/sq.ft. °F. Hr.
L = length of non-boiling section — feet
L_o = overall length of heated tube — feet

In this expression only L is unrelated to some familiar pattern of flow or heat transfer, yet under conditions where h_1 and h_2 differ widely, it is the ratio of L to L_o which determines h_a. Reference is therefore made to Fig. 12 which illustrates diagrammatically the temperature rise and fall of the fluid passing up the heated tube. The line AB represents the steady heating of the liquid in the non-boiling section until it/

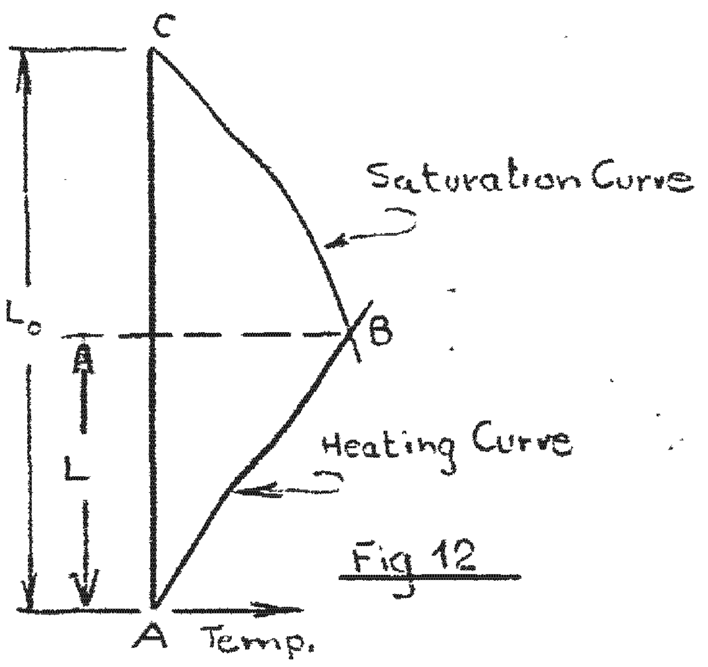

Fig 12

reaches/

reaches the appropriate saturation temperature at B – the point at which "flashing" occurs. BC is the saturation line corresponding to the diminishing hydraulic head on the liquid-vapour mixture rising in the tube. The shape of this curve has been confirmed experimentally by Boarts et al (17).

Calling q_1 : the heat transferred in the non-boiling section (BTU/sec)

D : the tube diameter (feet)

v : the average velocity of the liquid in the non-boiling section (feet/sec)

ρ_L : the density of the liquid (lbs/cubic foot)

C : the specific heat of the liquid (BTU/lb)

ΔT : the average temperature difference between the tube wall and the bulk of the liquid (large enough to be considered constant in relation to T) (°F)

δT : the temperature rise of the liquid passing through L feet of tube (°F)

then $q_1 = h_1 \times \Delta T \times L \times D \times \pi$

The heat gained by the liquid in passing through L feet of tube

$= v \times \frac{\pi D^2}{4} \times \rho_L \times C \times \delta T$, and this must also equal q_1.

Solving for h_1 yields $h_1 = \frac{\pi D^2 \times v \times \rho_L \times C}{4 \times L \times D \times} \times \frac{\delta T}{\Delta T}$

This may be expressed in dimensionless form as :

$$Nu = Re \times Pr \times \frac{D}{4L} \times \frac{\delta T}{\Delta T} \qquad \text{..........} \qquad (iv)$$

As the purpose of this calculation is the determination of L, Nu can be eliminated from (iv) by equating it with the fundamental heat transfer relations (i) or (ii) (depending on the type of flow).

In the case of turbulent flow this leads to:

$$Re^{0.2} = 0.108 \times \frac{L}{D} \times Pr^{-\frac{3}{5}} \times \frac{\Delta T}{\delta T} \qquad \text{..........} \qquad (v)$$

and for laminar flow:

$$Re^{1/3} = 7.4 \times \left(\frac{L}{D}\right)^{\frac{2}{3}} \times Pr^{-\frac{2}{3}} \times \frac{\Delta T}{\delta T} \qquad \text{..........} \qquad (vi)$$

The problem thus resolves itself into an estimation of Re, or [illegible] the liquid velocity v.

The flow in the non-boiling section must also obey the laws of fluid friction.

$$\text{i.e.} \quad H = 4f \frac{Lv^2}{2gD} \qquad \text{..........} \qquad (vii)$$

where H = Loss of head due to friction (feet)

f = Fanning friction factor.

For turbulent flow within the range Re = 5,000 to 200,000, and for smooth pipe surfaces McAdams (27) recommends the substitution of

$$f = \frac{0.046}{Re^{0.2}}$$

whilst for laminar flow $f = \frac{16}{Re}$ by definition.

Thus/

Thus solving (vii) for v yields

$$v^2 = \frac{Re^{0.2} \times g \times D}{0.092} \times \frac{H}{L} \qquad \text{..........} \qquad \text{(viii)}$$

for turbulent flow,

$$\text{and} \quad v^2 = \frac{Re \times g \times D}{32} \times \frac{H}{L} \qquad \text{..........} \qquad \text{(ix)}$$

for laminar flow.

We are left with the problem of assigning a value to H. Under equilibrium conditions the pressure drops in both legs of the system must balance. In practice the hydraulic head in the return tube is usually some constant fraction "a_1" of the length of the heated tube. If the friction loss in the (larger diameter) unheated section is ignored, the pressure at the base of the return tube can therefore be written

$$a_1 \times L_o \times \rho_L \quad ^*$$

Ignoring also other minor losses due to contraction, expansion and fittings, this pressure must equal the sum of :

(a) the hydraulic pressure of the non-boiling section ($= L\rho_L$)
(b) the friction pressure loss in the non-boiling section ($= H\rho_L$)
(c) the hydraulic pressure drop in the boiling section (say P_H)
(d) the friction pressure loss in the boiling section (say P_F)
(e) the pressure required to accelerate the liquid-vapour mixture in the boiling section (say P_A)

i.e. $L\rho_L + H\rho_L + P_H + P_F + P_A = a_1 L_o \rho_L$

or
$$H = a_1 L_o - L - \frac{P_H + P_F + P_A}{\rho_L}$$

The following analysis of the pressure drop of gas-liquid mixtures flowing upwards in vertical tubes is due to Govier (28). The terms P_H, P_F and P_A are defined by writing the mechanical energy balance for each phase in the stream:

$$w_L \, dt \left(V_L \, dP + dL + \frac{v_L \, dv_L}{g} + dH_{FL} \right) = 0 \qquad \text{..........} \qquad \text{(x)}$$

$$w_G \, dt \left(V_G \, dP + dL + \frac{v_G \, dv_G}{g} + dH_{FG} \right) = 0 \qquad \text{..........} \qquad \text{(xi)}$$

where
w_L = Flow rate of liquid (lbs/sec)
w_G = Flow rate of vapour (lbs/sec)
V_L = Specific volume of liquid (cubic feet/lb)
V_G = Specific volume of vapour (cubic feet/lb)
dL = Differential length of tube
v_L = Linear velocity of liquid (ft/sec)
v_G = Linear velocity of vapour (ft/sec)
dH_{FL} = Frictional Head loss of liquid (feet liquid)
dH_{FG} = Frictional Head loss of vapour (feet vapour)
dt = differential time
dP = Differential pressure drop

Adding/

* See Note (2) on p. 49

Adding (x) and (xi)

$$V_L\, dP + dL + \frac{v_L dv_L}{g} + dH_{FL} + \frac{w_G}{w_L} \times V_G dP + \frac{w_G}{w_L} dL + \frac{w_G}{w_L} \cdot \frac{v_G dv_G}{g} + \frac{w_G}{w_L} dH_{FG} = 0$$

Putting $(dH)_{FL} = dH_{FL} + \frac{w_G}{w_L} \times dH_{FG}$, and re-arranging :

$$-V_L \frac{dP}{dL} = \frac{1 + Rm}{1 + RV} + \frac{1}{1 + RV}\left(\frac{dH}{dL}\right)_{FL} + \left(\frac{v_L dv_L}{g} + \frac{w_G}{w_L} \cdot \frac{v_G dv_G}{g}\right)\frac{1}{(1 + RV)dL} \quad \ldots\ldots \quad \text{(xii)}$$

where Rm = Vapour liquid mass ratio $\frac{w_G}{w_L}$

RV = Vapour liquid volume ratio $\frac{w_G}{w_L} \times \frac{V_G}{V_L}$

If Rm and RV can be considered constant over a finite length L, expression (xii) becomes :

$$-V_L \frac{\Delta P}{\Delta L} = \frac{1 + Rm}{1 + RV} + \frac{1}{1 + RV}\left(\frac{\Delta H}{\Delta L}\right)_{FL} + \left(\frac{v_L \Delta v_L}{g} + Rm\frac{v_G \Delta v_G}{g}\right)\frac{1}{(1 + RV)dL} \quad \ldots\ldots \quad \text{(xiii)}$$

In this expression the three pressure drops P_H, P_F and P_A are represented by the terms $\frac{1 + Rm}{1 + RV}$, $\frac{1}{1 + RV}\left(\frac{\Delta H}{\Delta L}\right)_{FL}$ and $\left(\frac{v_L \Delta v_L}{g} + Rm\frac{v_G \Delta v_G}{g}\right)\frac{1}{(1 + RV)dL}$ respectively. Owing to vapour generation the ratios Rm and RV are not constant along the length of the boiling section of the heated tube, and $V_L\frac{\Delta P}{L\Delta L}$ can therefore not be evaluated for the section as a whole. It is possible, however, to estimate the magnitude of each of the three components at selected points along the length of the tube, and to derive graphically an average figure for their sum.

Full details of this calculation are given in Appendix V, and it is shown that the term

$$\frac{P_H + P_F + P_A}{\rho L}$$

can be expressed as a fraction "a_2" of a column of liquid equal to the length of the boiling section,

$$H = a_1 L_o - L - a_2(L_o - L)$$

$$= (a_1 - a_2) L_o - (1 - a_2) L \quad \ldots\ldots \quad \text{(xiv)}$$

Expression (viii) for turbulent flow thus becomes

$$v^2 = \frac{Re^{0.2} \times g \times D}{0.092} \times \frac{(a_1 - a_2)L_o - (1 - a_2)L}{L}$$

To convert to a dimensionless form multiply both sides by $\frac{\rho_L{}^2 D^2}{\mu^2}$, whence, on separating the groups,

$$Re^{1.8} = \frac{gD^3\rho^2}{0.092\mu^2} \times \frac{(a_1 - a_2)L_o - (1 - a_2)L}{L} \quad \ldots\ldots \quad \text{(xv)}$$

Similarly for laminar flow

$$Re = \frac{gD^3\rho^2}{32\mu^2} \times \frac{(a_1 - a_2)L_o - (1 - a_2)L}{L} \quad \ldots\ldots \quad \text{(xvi)}$$

By combining (xv) with (v) and (xvi) with (vi), expressions are obtained

which/

which – given the dimensions of the heated tube and the physical constants of the working fluid – permit L to be calculated for turbulent as well as for laminar flow. To complete the calculation L is inserted in (iii) and h_a is obtained.

Worked Examples

The object is to determine the variation of the average heat transfer coefficient h_a over a practical range of viscosities. In a typical stick-water concentrator the apparent viscosity of the liquid may range from say 1 cp in the first to 500 cp in the final effect. Compared with this order of variation, the other physical properties of the liquid may be considered constant.

Suppose therefore that the (idealised) fluid has a density of 70 lbs/cubic foot; a specific heat of 0.5 BTU/lb; and a thermal conductivity of 0.2 ft. BTU/sq.ft.°F.Hr; also that the heated tube is 8.0 ft. long, and has an effective diameter of 35 mm (0.115 ft.) As final-effect conditions are of particular interest, assume that the bulk temperature of the liquid is 120°F; and that the nominal temperature gradient between the tube wall and the liquid is 100°F. Let the boiling point elevation of the liquid at levels below the tube outlet follow the saturation curve for a 70% corn syrup solution, as determined experimentally (Appendix I) and illustrated in Fig. 13.

Assume also that the plant is operated (as is common practice) with the liquid level in the return tube at the height of the upper tube plate in the calandria (constant a_1 in (xiv) equal to unity). As shown in Appendix V, the maximum and minimum values of the factor "a_2" can be taken as 0.1 and 0.9, and the limits for H are therefore

$$L_o - L - 0.1(L_o - L) = 0.9\,(L_o - L)$$

and

$$L_o - L - 0.9(L_o - L) = 0.1\,(L_o - L)$$

It is possible to calculate the length of the non-boiling section L from the dimensionless relations (xv), (v), (xvi) and (vi), but it is more convenient to select a number of arbitrary points on the saturation curve (δT vs L) and to determine the corresponding values of μ, v, Re and Pr and h_1, which by combination with L and h_2 leads to the unknown h_a.

In the case of turbulent flow, expression (v) then becomes:

$$v = \frac{1}{(0.000672\mu)^2} \cdot \underbrace{\left| \frac{1}{D\rho}\left\{0.108\left(\frac{k}{3{,}600C}\right)^{3/5} \cdot \frac{L}{D} \cdot \frac{\Delta T}{\delta T}\right\}^5 \right|}_{\text{A (say)}} \qquad \text{..........} \qquad \text{(xvii)}$$

and (viii) becomes

$$v = \frac{1}{(0.000672\mu)^{0.111}} \underbrace{\left| \left\{ \frac{\rho^{0.2} \times D^{1.2} \times g}{0.092} \times \frac{H}{L} \right\}^{0.555} \right|}_{\text{B (say)}} \qquad \text{..........} \qquad \text{(xviii)}$$

Hence $\mu = 1490\left(\frac{A}{B}\right)^{0.53}$

Also/

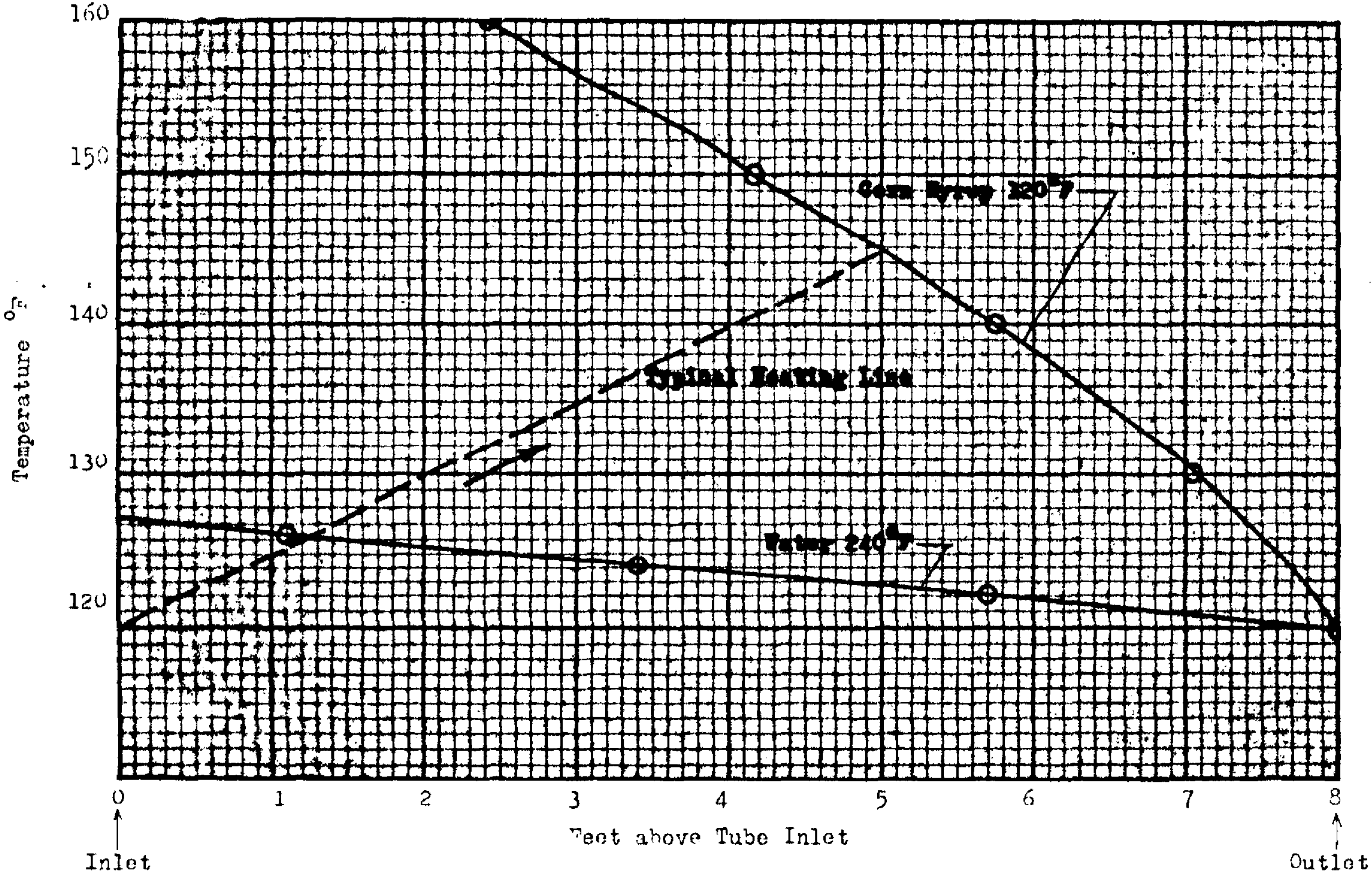

FIG. 13 : SATURATION LINES FOR WATER (240°F) AND A 70% CORN SYRUP SOLUTION (120°F)

Also $\quad Re = 12{,}000 \frac{v}{\mu}$

$\quad Pr = 6.05\mu$

$\quad h_1 = 0.047 \ Re^{0.8} \ Pr^{0.4}$

The corresponding relations for laminar flow, starting with (vi) are :

$$w = \frac{1}{3{,}600} \cdot \frac{kL}{C}\left(2\pi \cdot \frac{\Delta T}{\delta T}\right)^{3/2}$$

$$\mu = \frac{D^2 \cdot \rho g}{32} \cdot \frac{H}{v_L}$$

$$v = \frac{4}{\pi D^2 \rho} \cdot w$$

$$h_1 = 2.0 \times \frac{k}{D}\left(\frac{3{,}600C}{k}\right)^{\frac{1}{3}}\left(\frac{w}{L}\right)^{\frac{1}{3}}$$

To calculate h_a, values have to be assigned to h_2, the heat transfer coefficient in the boiling section.

The highest heat transfer coefficient observed in any of the tests with the experimental apparatus was measured in run 161 (Table XVI, throttled flow with water), under conditions in which boiling must have occurred along practically the whole length of the heated tube. It is a legitimate assumption that under these circumstances h_a must have been very nearly equal to h_2, or in figures 502 BTU/sq.ft.°F.Hr. The bulk temperature of the water in this test was 174°F, and its viscosity 0.37 centistokes (or 0.35 cp.)

Applying the viscosity correction (p.42), h_2 can for practical purposes be expressed as

$$500\left(\frac{0.35}{\mu}\right)^{0.125}$$

While this is no more than an approximation, it will presently be shown that for the purpose of the argument the numerical value of h_2 is not critical, since it does not affect the fundamental difference between turbulent and laminar flow and heat transfer.

Having derived h_1, h_2 and L as described, the average heat transfer coefficient for the whole tube can be calculated for a representative range of viscosities from

$$h_a = \frac{h_1 \times L + h_2\,(L_o - L)}{L_o}$$

Details of the calculations are shown in Table XVII and the corresponding curves have been plotted in Fig. 14.

To check that the discontinuity in the $h_a - \mu$ relation occurs under all operating conditions encountered in stickwater concentrators, a similar set of calculations was performed with the key parameters changed to represent a typical first effect. It was assumed that the liquid temperature was 240°F, the boiling point elevation curve (shown in Fig. 13) was taken as for pure water, and the temperature gradient was assumed to be 30°F.

Details/

TABLE XVII

VARIATION OF AVERAGE FILM COEFFICIENT h_a WITH VISCOSITY

Final Effect Conditions : $T = 120^{\circ}F$; $\Delta T = 100^{\circ}F$. Saturation curve : As for 70% Corn Syrup Solution

(A) TURBULENT FLOW

Head Def^n	δT $^{\circ}F$	L ft	H ft	μ c.p.	v ft/sec	Re	Pr	h_1 BTU/sq.ft.$^{\circ}$F.Hr	h_2 BTU/sq.ft.$^{\circ}$F.Hr.	h_a BTU/sq.ft.$^{\circ}$F.Hr
$0.1(L_o-L)$	15	6.4	0.16	3.96	2.48	7,500	24	212	370	244
	20	5.75	0.225	1.22	3.62	35,600	7.4	459	428	450
	25	5.0	0.30	0.41	5.2	154,000	2.48	957	490	782
$0.9(L_o-L)$	10	7.05	0.855	9.3	5.5	7,100	56.2	283	332	290
	15	6.4	1.44	2.08	9.0	52,000	12.6	768	400	694
	20	5.75	2.02	0.65	12.7	235,000	3.93	1610	462	1285

(B) LAMINAR FLOW

Head Def^n	δT $^{\circ}F$	L ft	H ft	v ft/sec	μ c.p.	Re	h_1 BTU/sq.ft.$^{\circ}$F.Hr.	h_2 BTU/sq.ft.$^{\circ}$F.Hr.	h_a BTU/sq.ft.$^{\circ}$F.Hr.
$0.1(L_o-L)$	5	7.6	0.04	1.65	4.4	4,500	39	360	55
	10	7.05	0.095	0.536	34.7	185	27	281	57
	15	6.4	0.16	0.265	130	24.5	22	239	65
	20	5.75	0.225	0.154	350	5.3	19.5	210	72
	25	5.0	0.30	0.096	860	1.34	17.5	188	82
$0.9(L_o-L)$	5	7.6	0.36	1.65	39.6	500	39	277	51
	7.5	7.35	0.585	0.86	128	81	31.8	239	49
	10	7.05	0.855	0.536	312	20.6	27	214	49
	15	6.4	1.44	0.265	1,170	2.7	22	181	54

TABLE XVIII

VARIATION OF AVERAGE FILM COEFFICIENT h_a WITH VISCOSITY

First Effect Conditions : $T = 240^oF$; $\Delta T = 30^oF$.   Saturation curve : As for Water

(A) TURBULENT FLOW

Head Def^n	δT ^oF	L ft	H ft	μ c.p.	v ft/sec	Re	Pr	h_1 BTU/sq.ft.^{o}F.Hr.	h_2 BTU/sq.ft.^{o}F.Hr.	h_a BTU/sq.ft.^{o}F.Hr.
$0.1(L_o-L)$	2.0	6.0	0.2	26	2.41	1,100	157	61	339	143
	2.9	5.0	0.3	5	3.98	9,600	30.2	281	358	310
	3.9	4.0	0.4	1.1	6.19	67,500	6.65	730	432	581
$0.9(L_o-L)$	2.0	6.0	1.8	13.75	8.53	7,450	83.1	344	315	336
	2.9	5.0	2.7	2.65	14.3	64,700	16.0	992	388	765
	3.9	4.0	3.6	0.575	22.8	476,000	3.48	2,710	470	1,590

(B) LAMINAR FLOW

Head Def^n	δT ^oF	L ft	H ft	v ft/sec	μ c.p.	Re	h_1 BTU/sq.ft.^{o}F.Hr.	h_2 BTU/sq.ft.^{o}F.Hr.	h_a BTU/sq.ft.^{o}F.Hr.
$0.1(L_o-L)$	1	7.0	0.1	2.78	7.1	4,700	47	340	84
	2	6.0	0.2	0.84	55	183	34	266	92
	2.9	5.0	0.3	0.402	206	23.4	28	225	102
	3.9	4.0	0.4	0.206	670	3.7	24	194	109
	4.8	3.0	0.5	0.113	2,030	0.67	22	169	114
$0.9(L_o-L)$	1	7.0	0.9	2.78	64	520	47	260	74
	2	6.0	1.8	0.84	493	20.5	34	218	80
	2.9	5.0	2.7	0.402	1,850	2.6	28	171	82

Details of the calculations are shown in Table XVIII, and the relevant curves have been included in Fig. 14.

Inspection of these figures raises the question how h_a is affected by forced - or throttled - circulation. The answer to this is largely provided by the information in Tables XVII and XVIII. Using the same δT and L values read from the saturation curves (Fig. 13) as before, the corresponding liquid velocity was, in the case of turbulent flow, calculated from expression (xvii). The high temperature operating conditions were applied, as being the more appropriate, and h_a was calculated for viscosities of 0.5 cp, 2.0 cp and 5.0 cp.

For the laminar region, the values of v and h_1 (being independent of viscosity) were extracted from Table XVII for the low temperature operating conditions. The boiling zone coefficient h_2, and hence h_a, were calculated for viscosities of 20 cp, 100 cp and 500 cp.

All velocity : h_a figures are summarised in Table XIX and the corresponding curves have been plotted in Fig. 15.

DISCUSSION

Some of the assumptions that have been made in order to illustrate the dual-zone theory have necessitated over-simplification.

Doubts exist, for instance, about the sharpness of the division between the boiling and the non-boiling regions, since nucleate boiling can occur on sub-cooled surfaces. Furthermore, Kirschbaum (20) has suggested that the inner tube surface near the outlet may become blanketed with vapour, thus presenting a greater resistance to the passage of heat than the rest of the boiling zone.

It is also obviously incorrect to assume that the physical properties of the fluid are insensitive to changes in temperature and solids concentration.

However, the limits of temperature, viscosity and propelling head (H) have purposely been chosen sufficiently wide apart to embrace the most drastic changes in the secondary variables that could reasonably be expected. In view of this generous allowance for error and uncertainty, the stability of the pattern of h_a variation with viscosity is remarkable.

The soundness of the arguments that have been advanced is confirmed by a comparison of the theoretical curves in Fig. 14 with the summary of the experimental runs in Fig. 9. It is noteworthy that the change in the direction of the curve of $\dfrac{U_a}{\Delta T \times P^{\frac{1}{2}}}$ vs. Total Solids Concentration occurs roughly in the position that would be expected from an inspection of the h_a vs. Viscosity curves.

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VARIATION OF AVERAGE FILM COEFFICIENT h_a WITH VELOCITY

(A) TURBULENT FLOW Liquid Bulk Temperature = 240°F Saturation curve: As for Water

δT °F	L ft	μ = 0.5 cp				μ = 2.0 cp				μ = 5.0 cp			
		Pr = 3.025 h_2 = 480				Pr = 12.1 h_2 = 400				Pr = 30.25 h_2 = 358			
		v ft/sec	Re	h_1	h_a	v ft/sec	Re	h_1	h_a	v ft/sec	Re	h_1	h_a
2	6									64.2	154,000	2,590	2,032
2.9	5					25.7	154,000	1,800	1,275	4.0	9,600	282	310
3.9	4	30	720,000	3,540	2,010	1.88	11,300	223	312				
4.8	3	2.52	60,500	489	484								

(B) LAMINAR FLOW Liquid Bulk Temperature = 120°F Saturation curve : As for 70% Corn Syrup Solution

δT °F	L ft	v ft/sec	h_1 BTU/sq.ft.°F.Hr	μ = 20 cp		μ = 100 cp		μ = 500 cp	
				h_2	h_a	h_2	h_a	h_2	h_a
5	7.6	1.65	39	300	52	246	49	201	47
10	7.05	0.536	27	"	59	"	53	"	47.5
15	6.4	0.265	22	"	78	"	67	"	58
20	5.75	0.154	19.5	"	98	"	83	"	70
25	5.0	0.096	17.5	"	123	"	103	"	86
30	4.1	0.06	15.5	"	154	"	128	"	106

TABLE XIX

VARIATION OF AVERAGE FILM COEFFICIENT h_a WITH VELOCITY

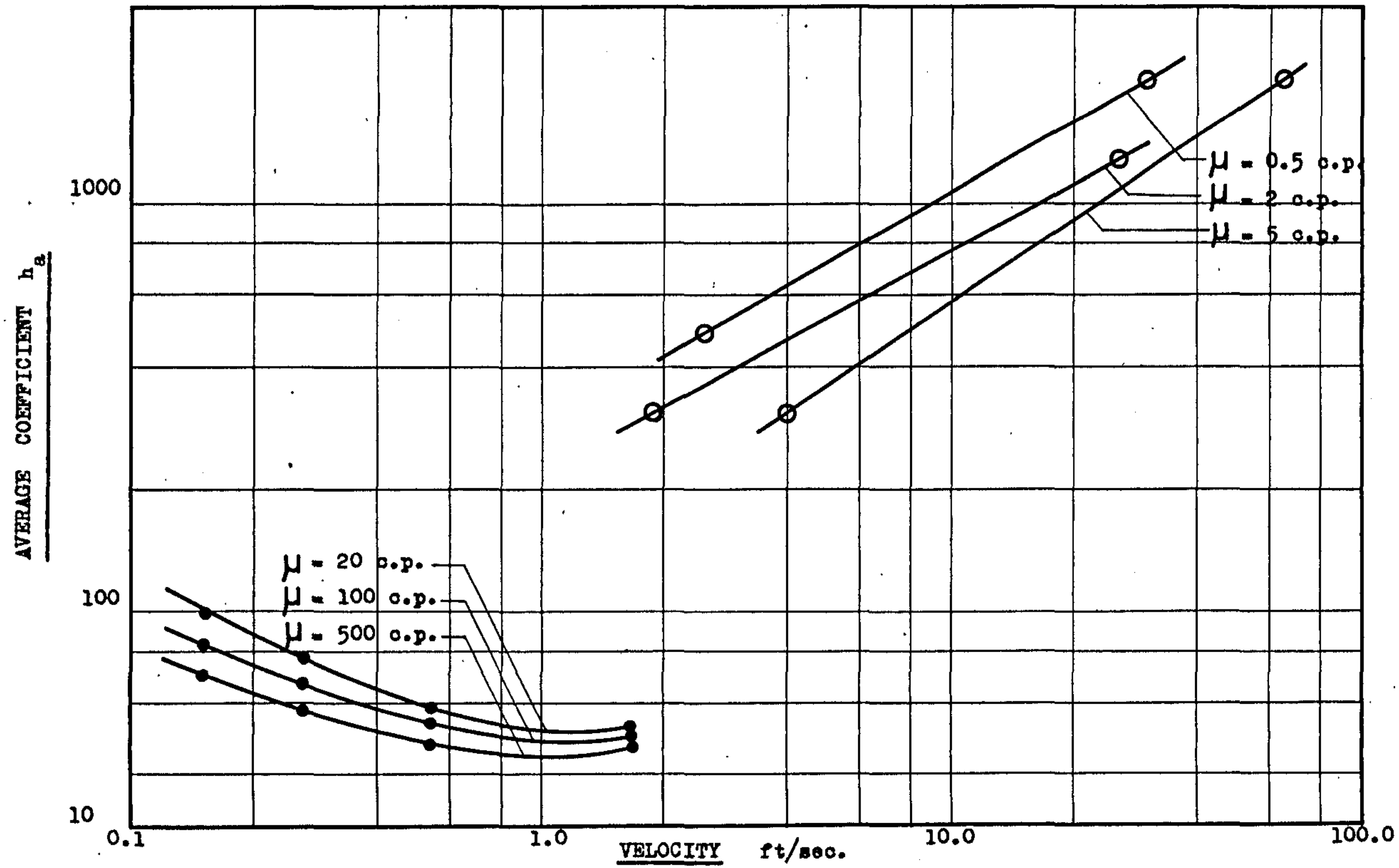
(A) TURBULENT FLOW   Liquid Bulk Temperature = 240°F   Saturation curve: As for Water

δT °F	L ft	μ = 0.5 cp, Pr = 3.025, h_2 = 480				μ = 2.0 cp, Pr = 12.1, h_2 = 400				μ = 5.0 cp, Pr = 30.25, h_2 = 358			
		v ft/sec	Re	h_1	h_a	v ft/sec	Re	h_1	h_a	v ft/sec	Re	h_1	h_a
2	6									64.2	154,000	2,590	2,032
2.9	5					25.7	154,000	1,800	1,275	4.0	9,600	282	310
3.9	4	30	720,000	3,540	2,010	1.88	11,300	223	312				
4.8	3	2.52	60,500	489	484								

(B) LAMINAR FLOW   Liquid Bulk Temperature = 120°F   Saturation curve : As for 70% Corn Syrup Solution

δT °F	L ft	v ft/sec	h_1 BTU/sq.ft.°F.Hr	μ = 20 cp		μ = 100 cp		μ = 500 cp	
				h_2	h_a	h_2	h_a	h_2	h_a
5	7.6	1.65	39	300	52	246	49	201	47
10	7.05	0.536	27	"	59	"	53	"	47.5
15	6.4	0.265	22	"	78	"	67	"	58
20	5.75	0.154	19.5	"	98	"	83	"	70
25	5.0	0.096	17.5	"	123	"	103	"	86
30	4.1	0.06	15.5	"	154	"	128	"	106

FIG. 15 : VARIATION OF FILM COEFFICIENT h_a WITH VELOCITY OF LIQUID IN NON-BOILING SECTION.



An attempt has been made to show that the discontinuity in these curves is most readily explained by presuming a transition from turbulent to laminar flow in the non-boiling zone, and that the viscosity of the liquid is by far the most important factor in bringing this transition about.

Note (1) : The numerical values of the coefficients reported by different investigators vary slightly, but not sufficiently to affect the argument.

Note (2) : The friction loss in the unheated section can be included by increasing L in (viii) and (ix) by a constant amount. This is equivalent to slightly changing H which, as shown in Fig. 14 does not alter the slope of the curves.

CHAPTER VIII

THE THEORETICAL AND PRACTICAL SIGNIFICANCE OF THE EXPERIMENTS

It was shown in the last chapter that the two distinguishing features of heat transfer in natural circulation evaporators are :

(a) the interdependence of the primary variables;

(b) the reversal of trends following the transition from turbulent to laminar flow in the non-boiling section of the heated tube.

Consider, for example, the effect of changes in the viscosity of the liquid.

If the viscosity is initially low enough to ensure turbulent flow in the non-boiling section, increasing it will tend to :

(i) reduce the heat transfer coefficient in the non-boiling section roughly in proportion to $\mu^{-0.4}$;

(ii) reduce the circulating velocity, which further reduces the heat transfer coefficient in the non-boiling section roughly in proportion to $v^{0.8}$;

(iii) reduce the length of the boiling section. While the flow in the non-boiling section is turbulent, the heat transfer coefficients in the boiling and non-boiling sections are roughly of a similar order of magnitude, and the relative length of the two zones is not critical.

When the viscosity has increased and the circulation velocity has dropped sufficiently for laminar flow to predominate in the non-boiling section, the heat transfer coefficient in this section drops to about one-tenth of the figure for turbulent flow, and becomes independent of viscosity. Further increases in viscosity therefore tend to :

(i) decrease the circulating velocity; but as the heat transfer coefficient in the non-boiling section remains virtually constant, the rate of temperature rise of the liquid ascending the tube increases, and it attains the critical saturation temperature at a lower level.

This in turn tends to

(ii) increase the length of the boiling section.

As the boiling heat transfer coefficient, though reduced at high viscosities, remains of a higher order of magnitude than the laminar film coefficient, the average coefficient for the heated tube as a whole will rise slightly, or remain virtually constant. This is illustrated by Fig. 14 in Chapter VII.

Consider next the effect of artificially changing the circulation velocity at constant viscosity.

If the velocity is high enough for the flow in the non-boiling section to be turbulent, the corresponding heat transfer coefficient is

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of a similar order as in the boiling zone; the average coefficient for the tube as a whole will be virtually independent of the relative lengths of the boiling and non-boiling sections, and will tend to decrease with velocity as a function of the Reynolds Number. This has been confirmed experimentally by Boarts et al (17).

When, however, the velocity drops sufficiently for laminar flow to predominate in the non-boiling section, further decreases in velocity will cause the length of the boiling zone to increase and the average heat transfer coefficient to rise accordingly. This is illustrated by Fig. 15, and it is this line of thought which prompted the exploratory tests with throttled flow reported in Chapter VI.

Consider finally the effect of changing the bulk temperature of the liquid. Fig. A.I(8) in Appendix I illustrates the increasing steepness of the saturation temperature curves at reduced pressures. This means that the boiling point elevation due to a given hydraulic head is greater at low temperatures than at high temperatures. Hence, assuming other conditions in the non-boiling section to be constant, the ascending liquid has to travel a longer distance to reach its equilibrium boiling temperature when it is cold than when it is hot. This results in a corresponding shortening or lengthening of the boiling section, and consequent reduction or increase of the average heat transfer coefficient.

In practice, changing the temperature of the liquid also changes its physical properties, particularly its viscosity, and it is difficult to separate the contribution of each variable to the final change in the average heat transfer coefficient.

A test was, however, devised to illustrate the dependence of U_a on the liquid bulk temperature. Using a single effect of the experimental apparatus, a run was made with water at 117°F. This was followed by a run using a 22% sucrose solution at 178°F, the sucrose concentration having purposely been chosen to have the same viscosity as water at 117°F.

The result of this test was as follows :

	Liquid Bulk Temperature	ΔT °F	U_a BTU/sq.ft.°F.Hr.
Water	117	48	213
22% Sucrose Solution	178	41	357

In spite of a slightly lower temperature gradient (which favours the test with water), the U_a value observed with the sucrose solution was about 67% higher. It is difficult to see how an increase of this order could have been caused by any factor other than the rise in the bulk temperature of the liquid, and its effect on the extension of the boiling zone.

The/

The practical implication of the "Dual Zone" theory is, that all efforts to increase the coefficients of heat transfer in natural circulation evaporators should aim at achieving one, or both of the following :

a) the extension of the length of the boiling zone relative to the non-boiling zone;

b) the promotion of turbulence in the non-boiling zone.

The tests described in Chapter VI are reviewed on the basis of this principle, taking into account economic and operational considerations.

A. REVERSE FEEDING

Reverse feeding, as opposed to forward feeding, results in a reversal of the physical properties of the liquid in the first and third effects. The solids concentration of the liquid in the second effect remains virtually unchanged. The gross difference between the viscosities of the liquids in the first and the final effects is narrowed by the rise in the temperature of the thick liquor and the fall in the temperature of the thin liquor. Consider, for instance, the TSS figures for runs 26 and 27 of Test "B", Table IX. If the viscosities at the appropriate temperatures are estimated by reference to Appendix I, the following picture emerges:

Run No.	TSS (%)			Viscosity (Centistokes)		
	I	II	III	I	II	III
26 (test)	47	14	8	30	1.5	1
27 (control)	11	16	49	0.5	1.5	100

Superficially it may appear promising that a six-fold (from $\frac{100}{0.5}$ to $\frac{30}{1}$) reduction in the ratio of final to first effect viscosities has been achieved. But it has been shown that heat transfer coefficients benefit from changes in viscosity only if:

a) Laminar flow changes to turbulent flow;

b) Turbulent flow becomes more turbulent;

c) (in exceptional cases) Laminar flow becomes more laminar.

The approximate Reynolds Numbers for these runs are :

Run No.	I	II	III
26 (test)	700	11,000	14,500
27 (control)	60,000	11,000	50

It/

It appears that while reverse feeding causes the flow in the third effect to become turbulent, it cannot prevent the opposite from happening in the first effect, and judging by the change in Reynolds Numbers a drop rather than a rise in heat transfer coefficient would be expected. This is confirmed by Tests "A" and "C" (Table IX), where reverse feeding resulted in a reduction of the gross U value by 5% and 3% respectively.

This reduction is small, and may even be absent as in Test "B". This may be due to the transfer of laminar flow conditions from the low temperature third effect to the high temperature effect. It was explained that the relative length of the boiling section is more critical when the flow is laminar than when it is turbulent, and also that its length increases with the bulk temperature of the liquid. The gain in heat transfer due to a given temperature rise is therefore greater if the flow is laminar than the loss due to a similar reduction in temperature if the flow is turbulent.

It appears that reverse feeding results in a fairly even balance of the forces that tend to improve and those that tend to depress the gross heat transfer coefficients. It does not appear to be the most promising means of stepping up the performance of existing plant.

B. ENZYME TREATMENT

It is shown in Appendix IV that the treatment of stickwater with certain enzymes (such as are for instance present in pilchard viscera) may reduce the viscosity of the concentrate by as much as 90%.

The change of heat transfer coefficients due to viscosity reduction would not be expected to be the same in successive effects of industrial stickwater concentrators. In the effects in which the liquid flow is normally turbulent, a viscosity reduction should improve the U values roughly in proportion to $\Delta\mu^{0.45}$ (see Fig. 14). In the final effect, in which the flow is normally laminar, a viscosity reduction will double or treble the U value, provided it is large enough to create turbulent conditions. Failing this - i.e. if the flow remains laminar - no significant change in U value can be expected.

Referring to Test "C" (Table X), the Reynolds Number for the flow in the third effect of the control run - assuming a viscosity of the untreated concentrate of say 100 cp and a density of 65 lbs/cubic ft -

would have been about $\dfrac{0.115 \times 0.6 \times 65 \times 10^4}{100 \times 6.72} = 65$. Assuming a maximum reduction in viscosity of 1 : 10 and a doubling of the velocity, the Reynolds Number after treatment would have been about 1,300.

Allowing for disturbing influences such as nucleate boiling, it is

probable/

probable that some turbulence would occur in the non-boiling section even at this low Reynolds Number, and a slight rise in heat transfer would be expected.

In practice a 20:1 increase in the Reynolds Number as a result of enzyme treatment would rarely occur, laminar flow would persist and the U_a value would hardly change.

This is the trend observed in the experiments. For instance, in Test "A" (Table X) treatment of stickwater with 10% pilchard gut increased the U_a values of the first, second and third effects by 13%, 32% and 7% respectively. The greater increase of U_a values in the second effect as compared with the first (high temperature) effect observed in all tests with enzyme treatment is indicative of the greater length of the non-boiling section in the former. In Test "B" (Table X), the U_a value in the first effect decreased with enzyme treatment, but this may be due to differences in the terminal temperatures, which was a weakness of the earlier triple effect tests.

The enzyme treatment of stickwater has the following advantages and disadvantages:

1) under optimum conditions it is capable of increasing the gross U value of existing stickwater concentrators by at least 30%; <u>but</u> the effect is inconsistent, and may at times be insignificant. It would, therefore, be unwise to base a design on the highest of the observed coefficients;
2) for maximum effectiveness, the treated stickwater should be kept at a temperature between 100°F and 150°F for at least half an hour. This may involve cooling and storing the raw stickwater, which normally enters the first effect at about 180°F. On the other hand, the enzyme carrier may also be introduced into the final effects of vacuum concentrators (which operate at or near the optimum reaction temperature). In such case the earlier effects would not benefit from the treatment;
3) the viscosity reduction effected by the enzymes not only aids heat transfer, but also facilitates mixing the concentrate with the presscake, or its conversion to drum-dried solubles. The concentration of maasbanker stickwater is virtually impossible without the use of enzymes to prevent tube blockage as described in Chapter III;
4) in most pilchard canneries the disposal of the viscera presents a problem, and its use as a stickwater additive is likely to be welcomed regardless of the effectiveness of the treatment.

The advantages offered by the enzyme treatment of stickwater are largely intangible and difficult to express in economic terms. If, however, the enzymes are available at nominal cost as in pilchard viscera, the factories appear to have much to gain from its use.

C. FORCED/

C. FORCED CIRCULATION

All experimental evidence points to the final effect as being the bottleneck to heat transfer in conventional concentrators. To produce concentrate of 50% TSS or higher, and with liquid velocities with natural circulation of 0.5 ft/sec or lower, it is inevitable that the flow in the non-boiling section of the tubes is laminar. Forced circulation, to be effective, should theoretically be applied at a sufficient rate to cause turbulence in the non-boiling section. Failing this, i.e. if the accelerated flow remains laminar, the reduction in the length of the boiling zone may lower instead of raise the average U_a values, as was found in Test "C" (Table XI).

In view of the naturally unsteady conditions in the non-boiling section, it may be permissible to take a Reynolds Number as low as 2,000 as the minimum required to ensure adequate turbulence. Then, assuming that the concentrate has a viscosity of 100 cp, the lowest desirable circulation velocity in a typical plant would be

$$\frac{2{,}000 \times 100 \times 6.72}{0.115 \times 65 \times 10^4} = 18 \text{ ft/sec.}$$

Reference to Table XIX and to the literature (31) indicates that at this liquid velocity boiling in the tube is practically eliminated, and the forced convection relationship may be used to estimate the heat transfer coefficients, i.e.

$$Nu = 0.027\ Re^{0.8} \times Pr^{0.4}$$

Assuming for example k = 0.2 BTU/sq.ft.°F.Hr. and C = 0.5 BTU/lb,

$$h_a = \frac{0.027 \times 0.2}{0.115} \times 2{,}000^{0.8} \times 605^{0.4} = 266 \text{ BTU/sq.ft.°F.Hr.}$$

To convert to the overall coefficient, assume steam film and tube wall coefficients of 1,000 and 2,000 BTU/sq. ft. °F. Hr. respectively, whence

$$U_a = \frac{1}{\frac{1}{266} + \frac{1}{2{,}000} + \frac{1}{1{,}000}} = 190 \text{ BTU/sq.ft. °F. Hr.}$$

Even if allowance is made for the Sieder and Tate viscosity correction, and for some nucleate boiling, it would be unwise to accept a practical U_a value in excess of <u>say 200 BTU/sq.ft. °F. Hr.</u> This is roughly twice the average U_a value for all the control tests with natural circulation shown in Table XIII and represents an improvement which would be warmly welcomed by the operators.

To estimate the horse-power required to maintain this circulation rate, a conservative estimate of the friction factor would be about 0.01. Assuming contraction and enlargement coefficients of 0.5 and 1.0 respectively (32), the Head loss would be :

$$\left(4 \times 0.01 \times \frac{8}{0.115} + 0.5 + 1.0\right)\frac{18^2}{2g} = 4.28 \times \frac{18^2}{2g} = 22 \text{ feet.}$$

Assuming the concentrate has a density of 70 lbs/cubic ft., this is equivalent to a nett power requirement per tube of

$$\frac{22 \times 70 \times 18}{550} \times \frac{\pi \times 0.115^2}{4} = 0.525 \text{ HP.}$$

For a typical third effect comprising two calandria with 230 tubes each, the nett power requirement would thus be 460 x 0.525 = 250 HP. The brake horse-power requirement would be considerably higher. The capital and the operating cost of a pump of this size far exceeds the maximum sum which could economically be justified even by doubling the heat transfer in existing plant.

Forced circulation at less than the theoretical minimum rate (18 ft/sec in the example just discussed) need not be without merit. The experiments (Table XI) suggest that provided the circulation velocity is higher than about 5 ft/sec., it is unlikely to cause the U_a value to decrease by more than a nominal amount.

Forced circulation at such low rates may take the place of enzyme treatment as a means of preventing tube blockages by maasbanker or similar gelatinous types of concentrate.

The improvement in the average heat transfer coefficients that can be achieved at intermediate circulation velocities (say 10 ft/sec.) depends on the degree to which the reduction in length of the boiling zone is compensated by the correspondingly greater coefficient in the non-boiling zone. An investigation into the economic justification of forced circulation in this region would require more extensive tests than could be undertaken as part of the present programme.

D. AIR-LIFT CIRCULATION

The essential feature of injecting air into the base of a heated tube in a natural circulation calandria is that it changes the non-boiling zone from a single to a two-phase system. Heat transfer to non-boiling two-phase systems has been extensively investigated and reviewed by McAdams (33).

A useful analysis was made by Groothuis and Hendal (34), who assembled their experimental data in terms of a two-phase Reynolds Number defined as the sum of the liquid and gas Reynolds Numbers, both based on superficial velocities.

None of the attempts to correlate the heat transfer coefficients observed with air-lift circulation either by means of known two-phase theorems, or by employing modified forms of Reynolds' Analogy (based on the energy potential of the injected air) has proved entirely successful. It appears that the presence and the effect of the boiling zone - with its entirely different mode of heat transfer - was never completely eliminated at the comparatively moderate rates of air injection that were used in these experiments.[1] This was particularly noticeable in the tests with concentrate and corn syrup solution, where the observed U_a

(1) Air injection rates in excess of about 2 litres per min. per tube would require outsize vacuum pumps.

values/

values were always considerably higher than those calculated by Groothuis and Hendal's method (34). Although the quantitative analysis of heat transfer with air-lift circulation is still a matter for speculation, it is possible to explain its action qualitatively.

Air injection invariably increases the circulation rate as shown in Tables XII and XIV. This taken by itself will tend to raise h_a if the flow in the non-boiling section is turbulent, or leave it virtually unchanged if the flow is laminar. Superimposed on this is the effect of two-phase instead of single phase heat transfer in the non-boiling section. According to Groothuis and Hendal (34) two-phase heat transfer changes from laminar to turbulent over a critical range of two-phase Reynolds Numbers in exactly the same way as single phase heat transfer. (They found that "laminar" two-phase Nusselt Numbers could best be correlated by $Re^{0.39}$ and "turbulent" Nusselt Numbers by $Re^{0.87}$).

A similar trend is evident when comparing Table XIV (tests with water) with Table XV (tests with corn syrup). Air injection appears to be much more effective as a promoter of heat transfer when the working fluid is water than when it is corn syrup. In a typical test with water (Table XIV, "C"), injection of 0.555 litres of air per minute raised the h_a value from 210 BTU/sq.ft. °F. Hr. to 560 BTU/sq.ft. °F. Hr.

With corn syrup solutions of estimated viscosities ranging from 45 to 1,000 centistokes, air injection rates of from 1 litre/min to 10 litres/min had to be used to achieve improvements in the U values of between 6% and 71%. Both groups of tests show a rapid reduction of the rate of increase of heat transfer coefficients with increasing air injection rates.

The tests with water (Table XIV and Fig. 11) also illustrated the greater effectiveness of air injection at low temperature gradients (ΔT). This is to be expected, since a higher temperature gradient results in an increase in the rate of temperature rise of the liquid in the non-boiling zone. Hence the length of the non-boiling zone is shortened, and the benefit of its conversion to a two-phase system diminishes.

The feature which renders air-lift circulation attractive as an aid to heat transfer in spite of uncertainty regarding its operating mechanism, is its low power requirement.

The nett power required to achieve the observed improvement in heat transfer has been calculated for each of the test runs in Table XIII.

The air-lift cycle in its simplest form is illustrated in Fig. 16 (a) and the corresponding P-V diagram in Fig. 16 (b).

Putting/

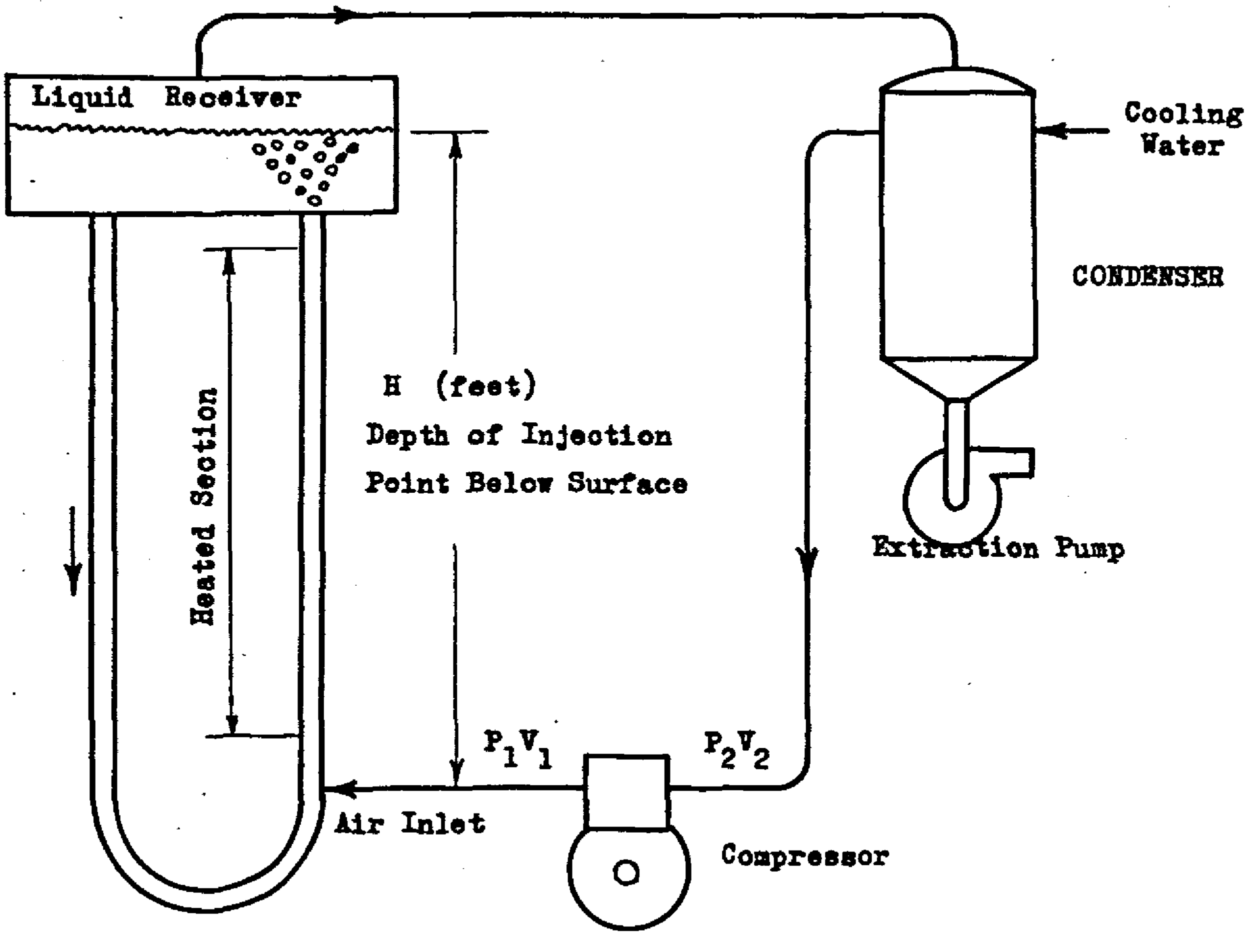

FIG. 16 (a) : THE AIR-LIFT CYCLE

$$W = \frac{P_1V_1 - P_2V_2}{n-1} = \frac{P_2V_2}{n-1}\left\{\left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}} - 1\right\}$$

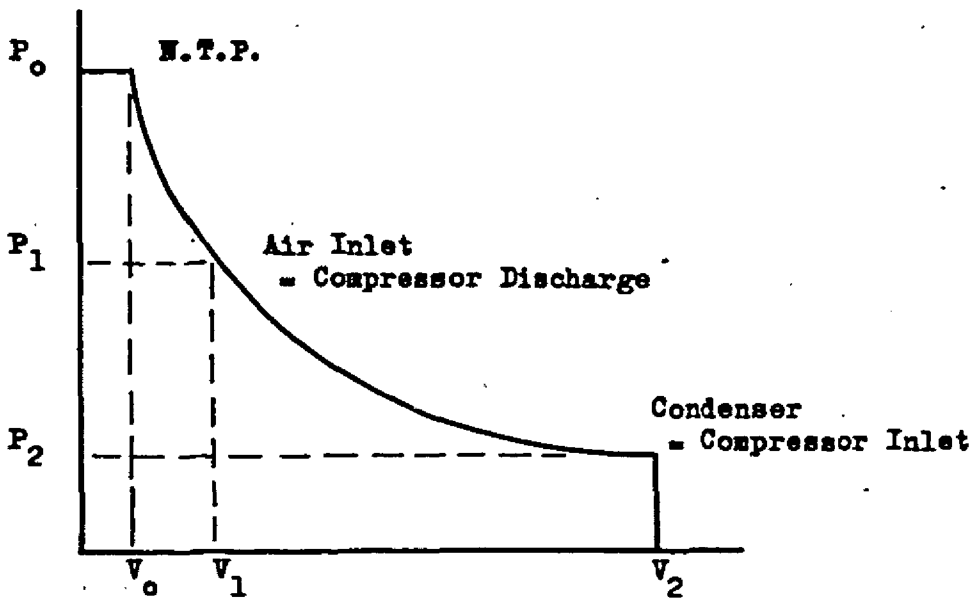

FIG. 16 (b) : AIR-LIFT P-V DIAGRAM

Putting W = Work performed. Compressing air. ft.lbs/lb.
P_o = Atmospheric pressure lbs/sq.ft.
P_1 = Air pressure at injection point "
P_2 = Air pressure in plenum "
V_1 = Specific volume of air at injection pressure cubic ft./lb.
V_2 = Specific volume of air in plenum "
w = Weight of air injected lbs/sec.

From first principles, the work represented by any PV diagram is given by

$$W = \frac{P_1V_1 - P_2V_2}{n-1} = \frac{P_2V_2}{n-1}\left[\left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}} - 1\right]$$

It was assumed that

a) Compression is adiabatic, i.e. $n = 1.4$
b) The pressure of the air at the compressor inlet is the same as the plenum pressure
c) The pressure at the point of air injection is equal to the plenum pressure plus the equivalent of a 10 ft. head of water.
d) The specific volume V_2 of the air entering the compressor is equal to the specific volume of dry air at 100°F (the assumed inlet temperature to the compressor) multiplied by $\frac{P_o}{P_2}$

The weight of air injected per second was calculated by converting the observed injection rate from Litres/min to cubic ft./sec. and dividing by 13.5, the nominal specific volume of the ambient air. Hence the I.H.P. was calculated as $\frac{w \times W}{550}$ and the results included in Table XIII.

It will be observed that the I.H.P. varies considerably from run to run; e.g. from 0.0015 HP to achieve a 31% improvement in U_a with stickwater of 40% TSS (run 87), to 0.0525 HP to achieve a 100% improvement in U_a with stickwater of 66% TSS (run 90). It is fundamentally impossible due to variations of the physical properties of stickwater to deduce a correlation which permits the horse power required to achieve a given improvement in U_a to be presented graphically in a comprehensible manner.

However, the following is a list extracted from Table XIII of the I.H.P. required to achieve U_a values of about 200 BTU/sq.ft. °F. Hr. by air-lift circulation :

Run No.	TSS %	U_a (Control) BTU/sq.ft.°F.Hr.	U_a (test) BTU/sq.ft.°F.Hr.	I.H.P.
85	42	129	207	0.0104
93	50	139	202	0.0267
94	50	139	222	0.0473
103	42	119	203	0.0056
102	42	119	234	0.0132
80	57.5	77	184	0.0267
97	59	137	196	0.040

Comparing/

Comparing these figures with the 0.525 HP estimated to be required to achieve a U_a value of 200 BTU/sq.ft. °F. Hr. by forced circulation suggests that the same could be achieved by air-lift circulation using less than a tenth of the power.

The application of air-lift circulation to existing plant involves a number of secondary problems which are difficult to study on a laboratory scale. The more important of these are :-

1) the uniform distribution of air to individual tubes in the calandria;
2) the effect of the injected air on the condenser capacity; and
3) the possibility of accelerated corrosion of the mild steel components owing to the presence of excess air in the vapour.

Nevertheless, the experiments with air-lift circulation appear sufficiently promising to warrant further tests on a semi-industrial scale.

E. THROTTLED CIRCULATION

Technically, throttled circulation is a type of forced circulation, but its action is quite different. Forced circulation is a "brute force" method of raising the heat transfer coefficient of the non-boiling section to a similar level as that of the boiling zone (which is generally eliminated in the process).

Throttled circulation, requiring no force, is aimed at maintaining the boiling zone at its maximum length under all conditions of temperature and viscosity.

This can be achieved by :

a) reducing the hydraulic head on the liquid in the heated tube, thus permitting boiling to commence at a lower temperature; or by
b) reducing the velocity of the liquid in the non-boiling section to ensure that it attains its boiling temperature at the lowest possible level; or by
c) a combination of (a) and (b).

The head on the liquid in the heated tube is most conveniently reduced by means of a throttling valve in the return line, the degree of throttling being measured by the liquid level in a manometer tube connected to a point close to the inlet to the heated tube.

The velocity of the liquid into the heated tube is best regulated by means of a calibrated pump interposed in the return line.

Although the two throttling methods are based on different lines of reasoning and differ in principle, their action cannot be separated in practice. When, for instance, the head on the liquid entering the heated tube is reduced, it is inevitable that the velocity is reduced as well.

Similarly/

Similarly, controlling the velocity of the liquid flowing into the heated tube involves a break in the return line and a reduction of head.

The effect of throttling by valve has been studied. Kirschbaum (20), for instance, found that when evaporating water in a vertical tube 4 m. long, 40 mm. I.D., the h_a value rose by an average of 60% when the apparent liquid level in the heated tube was reduced from 75% to 40% of its length.

The U_a value observed by Kerr (35) in an industrial evaporator, with the apparent water level at 33% of the height of the heated tube (54 in.), was about 50% higher than when the liquid was level with the tube outlets.

No reference has been found to throttling at constant velocity, or to throttling applied to the concentration of viscous liquids.

The results of the experiments summarised in Table XVI are therefore of interest, firstly because they confirm the feasibility of throttling by pump, and secondly because they suggest that throttling is effective when concentrating stickwater to Total Solids contents even as high as 60%.

The peak in the range of U_a values observed at certain intermediate pumping rates (e.g. in Test "E" with sucrose solution) is analogous to the peak in h_a values found by Kirschbaum (20) for certain intermediate liquid heads, and is an indication of a reduction in heat transfer in the upper end of the heated tube. When evaporating water, this is caused by formation of dry spots in the boiling zone, but when concentrating a solution like sucrose, parts of the tube may become coated with overdried matter.

Consider, for instance, run 163 (Test "E"). The observed rate of distillation during this run was 27.4 lbs water per hour. The weight of liquid entering the tube was 0.0104 x 0.057 x 3,600 x 62 = say 135 lbs/hr, of which $135 \times \frac{66.5}{100} = 90$ lbs. was dry matter. The evaporation of 27.4 lbs. of water from the product implies that it must have left the tube with a solids content of $\frac{90 \times 100}{(135-27.4)} = 84\%$.

At such concentrations sucrose solutions as well as stickwater concentrate are semi-solid, and would rapidly foul the tube surfaces.

The liquid velocity at which the maximum U_a value was observed in Test "E" was 0.23 ft/sec., and the distillation rate was 43.7 lbs/hr. By the same calculation as before, the solids content of the liquid leaving the tube would in this case have been about 72%, a rise of 5% per pass, which evidently did not have a serious effect on heat transfer in the boiling zone.

It/

It must be concluded that when restricting the circulation of liquids at high solids concentrations, the measurement and control of the circulating velocity is more critical than the head of liquid in the heated tube. In such cases, throttling by means of a calibrated pump or similar positive displacement device would be preferable to throttling by valve.

It is apparent from Table XVI that the highest liquid velocity likely to be required for throttled flow would be about 0.25 ft/sec. Hence, for a typical effect comprising two calandria of 230 tubes each, the throttling pump should have a maximum capacity of

$$0.25 \times 0.0104 \times 460 \times 6.2 \times 60 = 450 \text{ gallons/min.}$$

In theory, throttling acts as a source rather than a consumer of mechanical energy. Assuming, for example, that the throttling action just considered is equivalent to maintaining a difference of say two feet head within the system, this for a circulation velocity of 0.25 ft/sec. is equivalent to a power output of

$$\frac{0.25 \times 0.0104 \times 460 \times 2 \times 62}{550} = \underline{0.27 \text{ H.P.}}$$

(A positive drive would in practice be fitted to overcome the fluid friction in the system).

Thus, although the displacement of a throttling pump is likely to be high, its power consumption would only be nominal.

In all but the final effects of existing plant throttling by valve should prove satisfactory, as the circulation rate is not critical. Alternatively, it may be possible to devise a control mechanism whereby the throttling valve could be operated to maintain a measured, pre-set rate of flow.

The aim should be to maintain the lowest rate of flow consistent with avoiding excessive thickening of the product on its passage through the heated tube. The optimum liquid velocity for this purpose can only be determined by experiment, but, once established, it should be controlled directly and not by means of secondary variables such as the hydraulic head.

CHAPTER IX

HEAT TRANSFER AND THE ECONOMICS OF STICKWATER CONCENTRATION

However obvious the advantage of increasing the efficiency of a process may appear, the appropriate measures will only be taken if they promise to raise the profits. Any proposal to boost the capacity of a given stickwater concentrator must therefore be related to the economics of the process.

The profitability of a process is defined as the difference between the market value of the product and its production cost. The market value of stickwater concentrate is a function of its chemical composition and its nutritional properties.

Tressler (36) quotes the following analysis as being typical of condensed fish solubles prepared from pilchards on the basis of 50% total solids:

	%
Protein (Minimum)	32.00
Fat (Minimum)	4.00
Fibre	0.00
Ash	9.00

Minerals

	%		%
Calcium (Ca)	0.87	Magnesium (Mg)	0.016
Phosphorus (P)	0.85	Sodium Oxide (Na_2O)	1.87
Iron (Fe)	0.025	Potassium Oxide (K_2O)	1.93

Amino Acids

	% of Protein		% of Protein
Arginine	4.34	Tryptophane	0.35
Histidine	5.79	Methionine	1.51
Lysine	4.87	Threonine	2.35
Leucine	4.67	Cystine	0.58
Isoleucine	2.73	Glycine	10.90
Valine	2.98	Glutamic Acid	8.44
Phenylalanine	2.33		

Vitamins

	Mg per lb		Mg per lb
Riboflavin	8 to 9	Pantothenic Acid	18.00
Thiamin	1.80	Nicotinic Acid	150.00
Choline	1500 to 1800	Pyridoxin	5.00

The nutritional properties of stickwater concentrate cannot be expressed in terms of a single, well-defined component. At

(1) F.I.R.I. analyses, though incomplete for certain minerals and vitamins, generally agree with these figures.

different/

different times particular importance has been attached to its content of crude protein, specific amino acids, vitamins or "growth factors". The problem has been extensively investigated e.g. by Laksesvela (37) and Wiechers and Laubscher (38).

In practice the concentrate is usually incorporated with the presscake to produce full meal. As no difference is made in the selling price of full meal as compared with ordinary meal, it is permissible to assign the same price to the stickwater solids.

The fixed price of fish meal sold on the local market is R77 per ton "at nearest station", or about R75 per ton at the factory. For fish meal with an average moisture content of 9%, this is equivalent to $75 \times \frac{100}{91}$ = R82.50 per ton on a dry basis. To arrive at the hypothetical selling price of stickwater concentrate, an allowance must be made for drying it from the moisture content at which it leaves the concentrator (say 50%) to the moisture content of the full meal (9%). A member of the Fishing Industry with considerable experience in this field has suggested a figure between R7 and R8 as being reasonable (39), in which case the market value of the dry solids in commercial stickwater can be taken as R75 per ton.

The cost of processing stickwater concentrate can be estimated by way of a worked example. The following calculation is based on the plant specified in Chapter III.

The salient features quoted for this plant were :

Evaporation :	38,000 pounds water per hour
Steam consumption :	16,000 pounds per hour

Specific steam consumption $= \frac{16,000}{38,000} = 0.42$ lbs/lb.

(to allow for starting and stopping, cleaning and breakdowns, the overall specific consumption is taken as <u>0.475</u> lbs/lb.)

Power requirement :	60 Horse Power
Cost :	R58,000 ex works.

A factory installing a plant of this size would expect to process an average of 50,000 tons of raw fish per annum. The average length of a fishing season can be taken as eight months.

The individual charges can be broken down as follows:

1) <u>Plant.</u> Allowing 15% for transport and erection, the total cost of the plant in operating condition will be

$$R58,000 \times \frac{115}{100} = R66,700.$$

Assume the life of the shell to be six years, and allow no credit for the tubes at the end of this period. Allowing 10% for interest and maintenance, the annual charge will be

$$66,700 \frac{(16.7 + 10)}{100} = \underline{R17,800}$$

2) Building/

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2) Building/

2) Building. Assume the floor space required is 20 ft x 50 ft and the cost of construction R4 per sq. ft. Then the cost of the building will be : R4,000 and the annual charge at 15% : R600.

3) Labour. The plant will require one attendant earning say R180 per month, and two labourers earning about R10 per week each to operate it. Two shifts must be provided for. Hence the total labour charge per season of eight months will be R3,600. Allowing for overtime and off-season employment, the annual charge will be say R5,000.

4) Steam. Assume that an average of 45% of the weight of the raw fish is evaporated during stickwater concentration. Hence the annual evaporation will be $50{,}000 \times \frac{45}{100} = 22{,}500$ tons, requiring $22{,}500 \times 0.475 = 10{,}700$ tons of steam. The cost of steam in a locality such as Walvis Bay is about R2.50 per ton, hence the annual charge will be R26,800.

5) Power. The average cost of electricity, including KVA charges in a locality such as Walvis Bay, is about 2 cents per unit. The nett operating time of the plant will be $\frac{22{,}500 \times 2{,}000}{38{,}000}$ = 1,180 hours; or allowing for starting, stopping and breakdowns say 1,500 hours. The annual charge will thus be $60 \times 0.746 \times 1{,}500 \times \frac{2}{100}$ = say R1,500.

The processing cost is the sum of items 1 - 5 :

Plant	R17,800
Building	600
Labour	5,000
Steam	26,800
Power	1,500
	R51,700 per annum

which is equivalent to

$$\frac{51{,}700}{22{,}500} = \text{R2.30 per ton of evaporation}$$

To arrive at the production cost, the cost of the raw material (stickwater) must be added to the processing cost.

There is no unequivocal way of calculating the charge for stickwater. If looked upon as a waste product, which creates a nuisance when it is returned to the sea, its value is nil. It can also be argued that because the fish has to be cooked and pressed, and the press-liquor centrifuged, the stickwater should bear the cost of pre-processing. Yet another approach is to estimate the maximum price which an independent operator would be prepared to pay for stickwater, intending to sell the concentrate at a reasonable profit.

The/

The most practical course is to base the value of the solids in the stickwater on the price which the factory pays for the raw fish.

Raw fish contains an average of 30% dry matter (22% solids and 8% fat), and its average purchase price is R10.00 per ton. Stickwater containing 9% TSS should then be charged out at $\frac{10 \times 9}{30} = $ R3.00 per ton.

A cost of R3.00 per ton of stickwater is equivalent to $3.00 \times \frac{100}{91}$ = R3.30 per ton of evaporation, which in turn is equivalent to a yield of $\frac{2{,}000 \times 9}{91}$ = 198 lbs. of dry solids.

At R75.00 per ton, the value of 198 lbs. of dry solids is $\frac{75 \times 198}{2{,}000}$ = R7.40.

The average working profit operating this concentrator is therefore of the order of

R7.40 - (2.30 + 3.30) = <u>R1.80</u> per ton of evaporation.

<u>Note:</u> This figure has been calculated only as an example. It is based on hypothetical values of raw material and finished product, and it can therefore not be used for comparative purposes.

It was shown in Chapter III that the U_a values for the first, second and third effects of this plant are about 260, 160 and 63 BTU/sq.ft.°F.Hr. respectively. The gross U_a value was calculated as $\frac{38{,}000 \times 1000}{1{,}800 \times 183}$ = 115 BTU.

Suppose that by applying air-lift circulation the U_a value for the third effect were increased to 150 BTU/sq.ft.°F.Hr., and that the U_a values of the other effects remain the same.

The new gross U value is given by

$$\frac{3}{\frac{1}{260} + \frac{1}{160} + \frac{1}{150}} = 179 \text{ BTU/sq.ft.°F.Hr.}$$

The most direct way of turning a higher Gross U value into profit is to reduce the size of the plant (or its replacement). Assume that the cost of components such as pumps and condensers which remain unaffected by an increase in the U_G value is about 25% of that of the whole plant. For estimating the cost of scaled-down plant, Zimmerman and Lavine (40) recommend using the "six-tenths" factor.

The ex works cost of the smaller plant is therefore estimated as

$$\frac{58{,}000 \times 25}{100} + \frac{58{,}000 \times 75}{100}\left(\frac{115}{179}\right)^{0.6} = 14{,}500 + 43{,}500 \ \frac{17.2}{22.5} = \text{R}47{,}800$$

The erected cost of the smaller plant will be $47{,}800 \times \frac{115}{100}$ = R55,000,

and the revised annual charge = $55{,}000 \times \frac{26.7}{100}$ = <u>R14,700</u> per annum.

Reference to Table XIII indicates that the power required to achieve this increase in U_a value by means of air-lift circulation would not exceed 0.02 I.H.P. per tube, or 0.02 x 460 = say 10 H.P. for the whole effect. Suppose this is equivalent to a real power

requirement/

requirement of 25 H.P., and capital equipment costing say R2,000. If the latter is treated as part of the main plant, the annual charge would be $2,000 \times \frac{26.7}{100}$ = R534 plus

$$25 \times 0.746 \times 1,500 \times \frac{2}{100} = \text{R560 for power.}$$

The nett annual saving would thus be (17,800 - 14,700) - (534 + 560) = R2,000 equivalent to $\frac{2,000 \times 100}{22,500}$ = 8.9 cents per ton evaporation.

This represents a $\frac{8.9 \times 100}{230}$ = 3.9% reduction in processing cost.

If it is assumed that the improvement in gross U value were achieved at no extra cost, for instance by throttling or enzyme treatment, the total annual saving would be R3,100 or $\frac{3,100}{51,700}$ = 6.0% of the processing cost.

Since the cost of steam is the major processing expense, an improvement in plant capacity should preferably be applied to reduce the steam consumption. A convenient way of achieving this is by operating one extra effect.

During normal operation, the temperature gradients across the first, second and third effects of this plant are 27°F, 33°F and 112°F respectively. The imbalance of the temperature gradients is a result of the inequality of the U_a values. Operating a fourth effect under these conditions has in practice been found to result in a reduction of plant capacity, owing presumably to a further deterioration of the final effect U_a value. Operators have sometimes attempted to use the "floating" calandria of this type of plant as a fourth effect, but have then been unable to take in all the stickwater.

Let it be assumed that by employing air-lift circulation, throttling or enzyme treatment, the capacity of a four effect plant could be maintained at least at the same level as that of a similar triple effect plant operated without aid to heat transfer.

The specific steam consumption of a typical plant converted to four effect operation would roughly be $0.42 \times \frac{3}{4}$ = 0.315, or for practical purposes say 0.375 lbs/lb.

The annual saving would thus be 0.1 x 22,500 x 2.50 = R5,600.

Using the "six-tenths" factor, the cost of a plant with one extra effect is estimated to be

$$\frac{58,000 \times 25}{100} + \frac{58,000 \times 75}{100}\left(\frac{5}{4}\right)^{0.6} = 14,500 + 43,500 \times 1.14 = \text{R64,100}$$

Hence/

Hence the revised annual charge will be

$$\frac{64,100 \times 115 \times 26.7}{100 \times 100} = R19,700$$

an increase of 19,700 - 17,800 = R1,900.

Allowing R1,100 p.a. for air-lift circulation as before, the nett annual saving becomes 5,600 - (1,900 + 1,100) = R2,600

$$\text{or } \frac{2,600 \times 100}{51,700} = 5\% \text{ of the processing cost.}$$

If the capacity of the plant can be maintained at no extra cost, e.g. by throttling or enzyme treatment, the expected saving due to operating a fourth effect would be

$$\frac{3,700 \times 100}{51,700} = 7\% \text{ of the processing cost.}$$

Two other methods that suggest themselves for capitalising a higher gross value are :

a) decreasing the overall temperature gradient by lowering the live steam temperature. This might permit the operation of a back pressure turbine to supply power to the rest of the reduction plant; or

b) decreasing the overall temperature gradient by raising the concentrate discharge temperature. It is a bacteriological requirement, which is not always observed in practice, that the temperature of the presscake - concentrate mixture entering the drier should exceed 180°F. Raising the temperature in the final effect would result in a moderate saving of live steam normally used to preheat the concentrate prior to mixing, and might also have the benefit, important though impossible to express in figures, of eliminating a potential source of bacterial contamination.

Although one of these systems might in practice result in a higher saving in processing cost than has been estimated for the two worked examples, yet potentially, the most important advantage of boosting the heat transfer in existing concentrators is that it provides reserve capacity.

The history of fish meal production in South Africa has been one of sustained expansion. Most fish meal plants are regularly replaced with units of larger capacity. But the enlarged stickwater plant to match the meal plant is generally only installed when the original plant has to be scrapped on account of wear and tear. Under present conditions, it is thus sometimes unavoidable for factories to have to run a proportion of their stickwater to waste for one or more seasons.

It also happens that a concentrator is just capable of handling all the stickwater when it is clean, but suffers a critical loss in

capacity/

capacity due to fouling between cleaning cycles. Again a considerable part of the stickwater that is produced has to be run to waste.

It can be seen from Fig 2 that a factory processing 50,000 tons of raw fish will produce about $\frac{55 \times 50,000}{100}$ = 27,500 tons of stickwater, containing roughly 9% solids. If only 10% of this is run to waste during a season, this is equivalent to a loss of revenue of $\frac{27,500 \times 9 \times 10 \times 75}{100 \times 100}$ = R18,500. The concentration of 2,700 tons of stickwater calls for an evaporation of about 2,700 ($\frac{41}{50}$) = say 2,200 tons of water. At R2.30 per ton (see p.64) this will cost 2.30 x 2,200 = R4,900. Assuming that boosting the capacity of the plant by air lifting involves an annual expenditure of R1,100 (as discussed on p.66), then the net profit to be expected from utilising the stickwater otherwise run to waste will be R12,500. No progressive fishmeal producer would ignore potential savings of this order.

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APPENDIX I

THE PHYSICAL PROPERTIES OF STICKWATER AT DIFFERENT SOLIDS CONCENTRATIONS, AND AQUEOUS SOLUTIONS OF CORN SYRUP AND CANE SUGAR

A. STICKWATER

Note: Most of the data on stickwater were first published by the writer as Progress Report No. 54 of the Fishing Industry Research Institute, University of Cape Town.

Stickwater is neither chemically pure nor physically homogeneous. Its composition and its physical properties have been shown to vary for different species of fish, and also depend on the condition of the fish at the time when it is processed. The tests that follow have confirmed this, particularly in the case of the apparent viscosity of stickwater at high total solids concentrations. It has nevertheless been possible to achieve the principal purpose of this investigation, which was to estimate the order of magnitude of the physical properties concerned, and the variations to be expected from sample to sample.

1. Viscosity

The object was to obtain sets of curves showing the variation of viscosity with temperature and with solids content of the concentrate. Furthermore, in order to utilise the data in the relevant flow and heat transfer calculations, it was essential that the viscosities should be determined in absolute units (centistokes or centipoises) and not in the relative units (seconds or degrees) used in most industrial viscometers.

After a few preliminary tests it was established that the standard U-tube method, as described in B.S.S. 188 of 1937, could be used satisfactorily with most stickwaters of moderate solids concentrations. As has been pointed out by Thomson et al (1), the term "viscosity" becomes vague as the solids content of any given sample is increased to high values, as it is affected by thixotropy, gelation during test, etc.; and indeed, reliable readings at viscosities above about 100 centistokes were often obtained with difficulty when using standard U-tubes. These difficulties generally disappeared when the viscosity of the sample was reduced, by lowering its solids content by dilution with distilled water. Wherever readings at high viscosities were therefore difficult or impossible to obtain, it was considered permissible to extrapolate to a limited extent from the more exact readings obtained at lower solids contents.

The solids concentrations of the samples under test were determined

by/

by refractometer. A few checks confirmed previous findings by Stander and Lewis (2) that the so-called "Total Soluble Solids" content of stickwater concentrate as determined by refractometer is actually closer to the "Total Solids" content (i.e. includes most of the Insoluble Solids). For convenience, refractometer readings reported as "% TSS by refractometer" were employed as standard.

The U-tubes were calibrated, and the whole test procedure laid down in B.S.S. 188 was followed in detail. The samples of concentrate were prepared in the laboratory from commercial stickwater made from fish which on the average had been processed not more than 24 hours after catch. The stickwater was concentrated in a laboratory triple effect evaporator, which in operation (e.g. in regard to temperatures in individual effects) closely followed industrial practice. Where stickwater or concentrate had to be kept for any length of time, the samples were stored at 32°F. In no case was the material stored for so long that signs of putrefaction had become evident.

A total of eight untreated pilchard concentrates were examined, and the results are summarised in Fig. A.I (1). This illustrates the variation of viscosity with % TSS by refractometer. As some of the curves were practically coincident, only the weighted average curve for all the samples, as well as the curves for the most viscous and the most fluid samples, have been plotted. The portions of the curves drawn unbroken are within the region for which good clear readings were obtained by direct measurement. The dotted upper portions have been extrapolated as the readings obtained by U-tube viscometers were indefinite.

Also shown dotted in Fig. A.I (1) is a curve representing the average of the viscosity determinations for two samples of maasbanker stickwater. The difference in viscosity between these particular samples and the average pilchard stickwater is striking, and bears out the contention of factory managers that maasbanker stickwater is much more difficult to process than pilchard stickwater.

These curves can best be interpreted in conjunction with Fig. A.I (2), which shows the variation of viscosity with temperature for a typical sample of concentrate. When plotted on a semi-log scale, all curves remain virtually parallel to the pure water base-line, even up to the highest solids concentrations, for which reliable readings were obtained.

As U-tubes could no longer be used at apparent viscosities above about 100 centistokes, attempts were made to obtain a picture of the fluid properties of thick concentrate by using a rotational viscometer.

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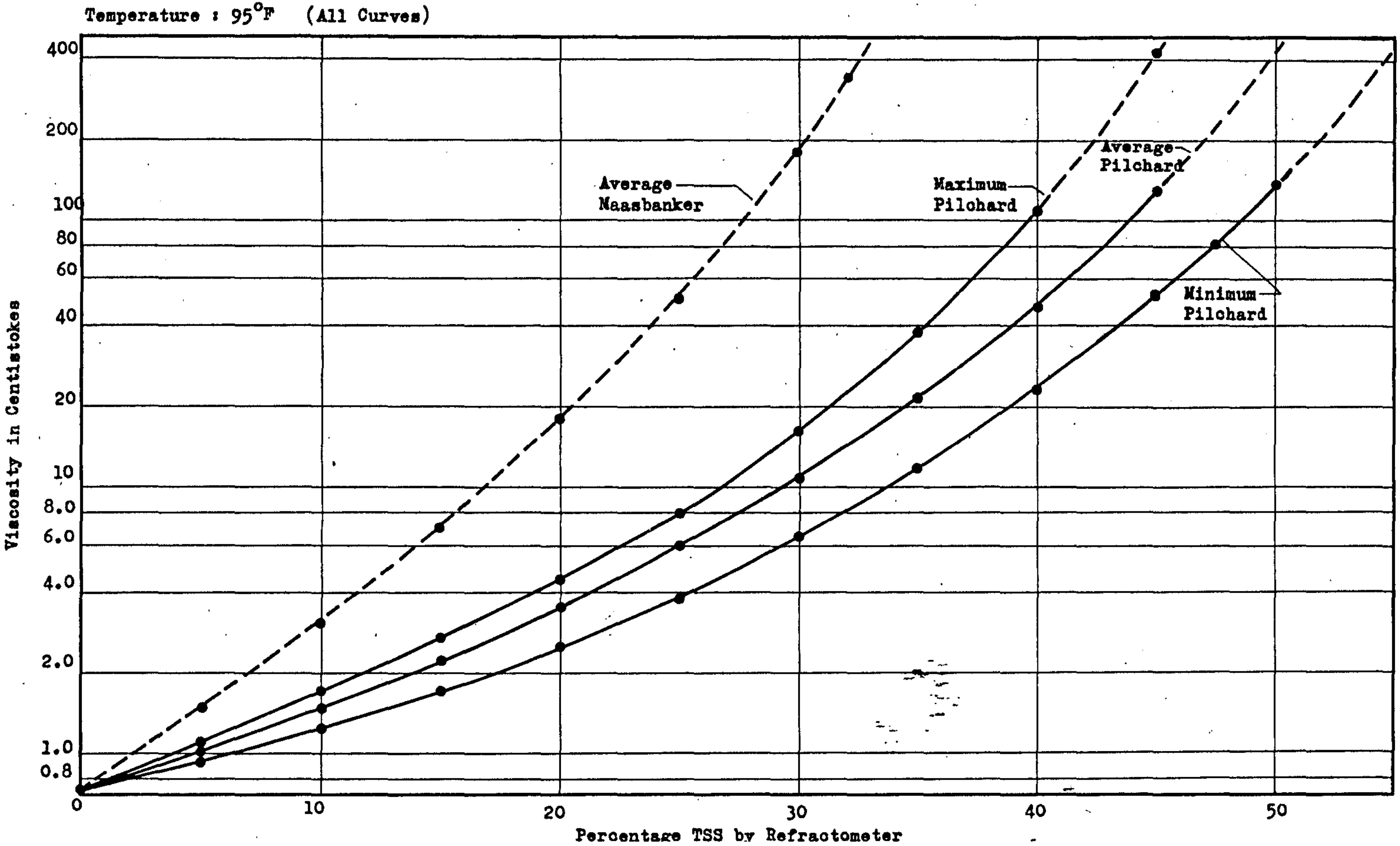
FIG. A.I (1) : VARIATION OF VISCOSITY WITH PERCENTAGE SOLUBLE SOLIDS FOR PILCHARD AND MAASBANKER STICKWATER



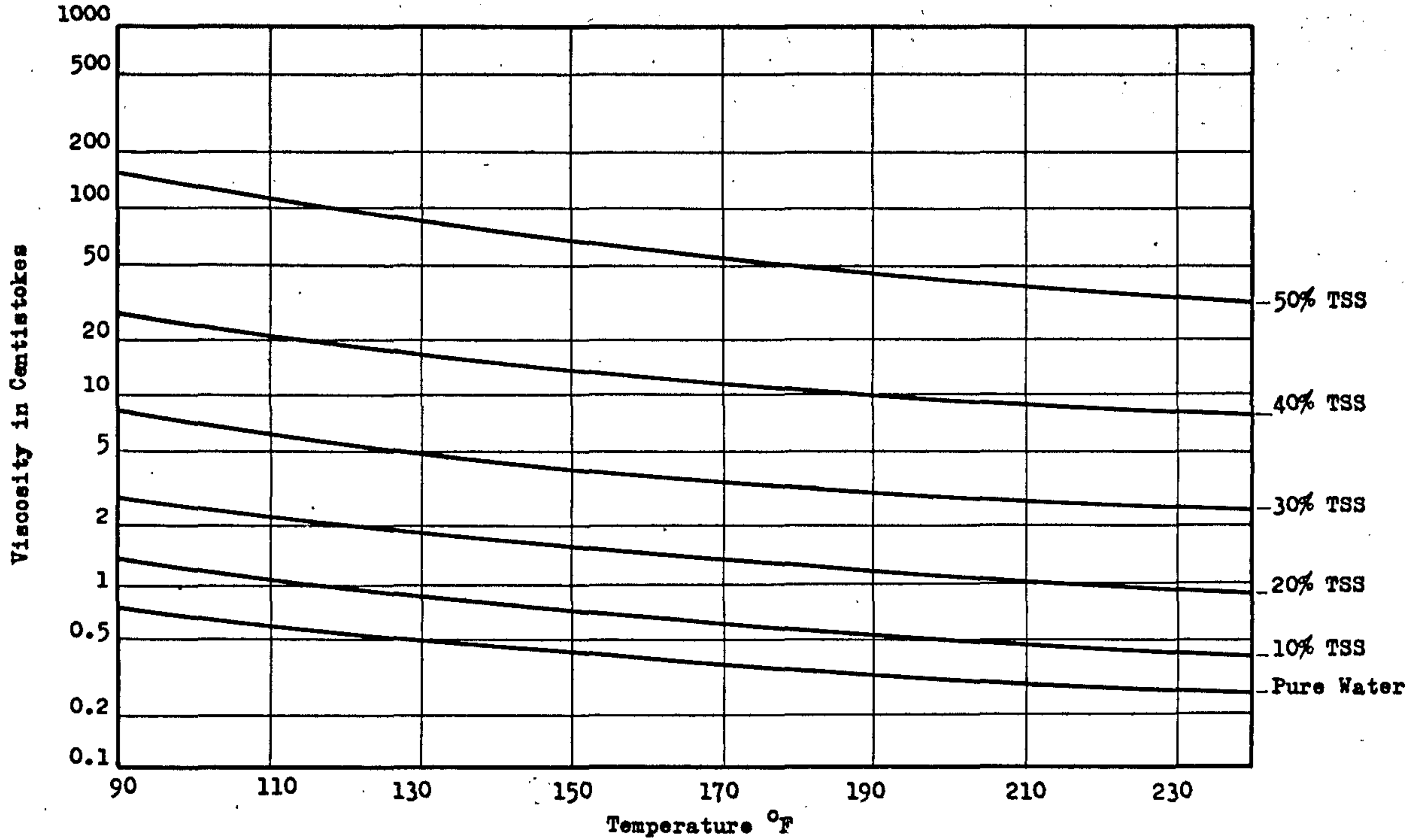


FIG. A.I (2) : VARIATION OF VISCOSITY WITH TEMPERATURE AND PERCENTAGE SOLUBLE SOLIDS FOR TYPICAL PILCHARD STICKWATER

The instrument used was the Brookfield Model LVT, which spins selected spindles in the test fluid, and measures the resulting drag on a suitably graduated scale. By taking readings at different spinning velocities, which can be selected via an eight-speed gearbox, a curve of shear rate (or r.p.m.) versus shear stress (or deflection on the arbitrary 0 - 100 scale) can be plotted. If the flow characteristics of a given fluid comply with the Newtonian definition of viscosity, all the points should lie on a straight line through the origin. This was confirmed by calibrating the instrument with a special type of mineral oil supplied for the purpose by a local Oil Company. The calibration curve is indicated in Fig. A.I (3). Also shown is the speed of rotation versus deflection curve for a typical sample of stickwater concentrate at 46% TSS and 121°F. The instrument was started at its lowest practical speed (1.5 r.p.m.) and a reading taken after one minute. It was then switched to the next higher speed (3 r.p.m.) and readings taken at the beginning and at the end of the one minute dwell period. This procedure was repeated for the intermediate velocities (6 r.p.m., 12 r.p.m. and 30 r.p.m.) up to the maximum velocity of 60 r.p.m. Without stopping the instrument, the velocities were then reduced and readings taken at one minute intervals as during the "up" run. The shape of the curve and the change of deflection at any given speed are typical of all concentrate samples tested.

These tests suggest two reasons for the irregular behaviour of thick concentrate: it is both pseudoplastic and thixotropic. Studies such as by Wilkinson (3) of the fluid mechanics of non-Newtonian fluids have shown that both effects can be classified mathematically, but the application of the relevant expressions to problems of flow andheat transfer have not yet had sufficient experimental verification. It appears particularly dangerous in the case of a loosely defined liquid such as stickwater concentrate, to over-simplify a limited number of experimental results in an effort to fit them to theories which have been developed for fluids of entirely different character.

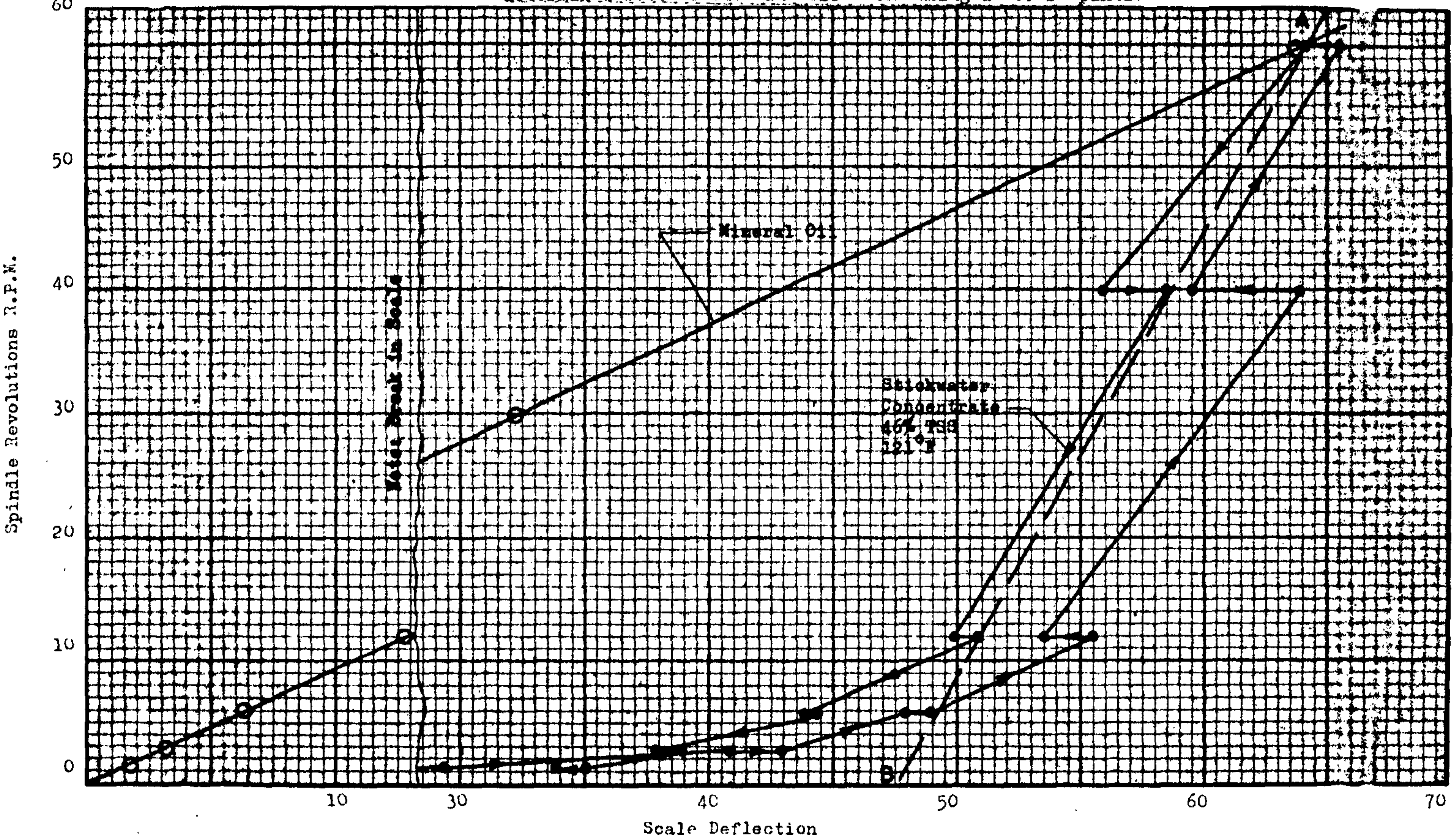
In such cases it is more satisfactory to select the parameter which most closely approaches the definition of Newtonian viscosity, with the understanding that the <u>apparent</u> viscosity thus determined may be used for comparative or qualitative purposes only.

Referring to Fig. A.I (3), it seems that the shape of the line A - B is the most easily defined feature of the stress - strain curve. By using the conversion chart supplied with the instrument, it is found that for this particular sample the apparent viscosity - as determined by the Brookfield Viscometer - is about 85 centipoises. The true viscosity of the sample of mineral oil was 330 centipoises, determined by the same method.

The/

FIG. A.I (3) : STRESS-STRAIN CURVES OF SAMPLES OF MINERAL OIL AND PILCHARD STICKWATER

determined with Brookfield Viscometer using a No. 2 Spindle



The apparent yield value (scale deflection "B" for zero spindle revolutions), and the thixotropic behaviour of the product are important in situations where, for instance, the liquid has to be pumped starting from rest. The starting-up power required would in practice be several times higher than indicated by the slope of the apparent viscosity line.

In other cases, notably when considering heat transfer in concentrators, it is possible that the apparent viscosity describes the flow characteristic of the fluid quite closely. Pseudoplasticity and thixotropy are of major importance only when the fluid is undisturbed for prolonged periods. In a concentrator this condition would be approached only in the non-boiling tube section, and then only in the immediate vicinity of the tube wall.

Nevertheless, it would be unwise to attach too much meaning to the apparent viscosity so defined, and it has consequently never been assumed to provide more than an indication of the order of magnitude of the fluid friction of any sample of concentrate which was too thick to give reliable readings by U-tube.

2. Thermal Conductivity

The apparatus used for determining the thermal conductivity of nine different concentrate samples at twelve different solids contents by refractometer is diagrammatically illustrated in Fig. A.I (4). The liquid under test was filled into the space marked "liquid disc". The upper brass plate was then brought to a constant temperature by means of an electric element, and the power input measured with a wattmeter. The lower brass disc was maintained at the temperature of melting ice by circulating ice water through the "cold tank".

Several preliminary runs were carried out with distilled water, and the upper brass disc maintained at a temperature between 90° and 100°F. From the known power input (about 20 w.) and the known thermal conductivity of water under the test conditions, the heat leakage through the insulation was determined (as BTU/Hr/°F temperature difference between "hot plate" and the ambient air). This figure for heat loss (approximately 0.2 BTU/°F.Hr.) was subsequently applied as a correction factor to the power input as determined during the runs with stickwater concentrate. (Note: the operating temperature was selected, to keep the heat loss to about 10% of the input). The absence of significant convection effects was confirmed by comparing the heat transfer through distilled water with that through a solution of 3% gelatine in distilled water(set firmly in the test cell at the time of the test). The rates of heat conduction were practically identical, thus ruling out major convection errors.

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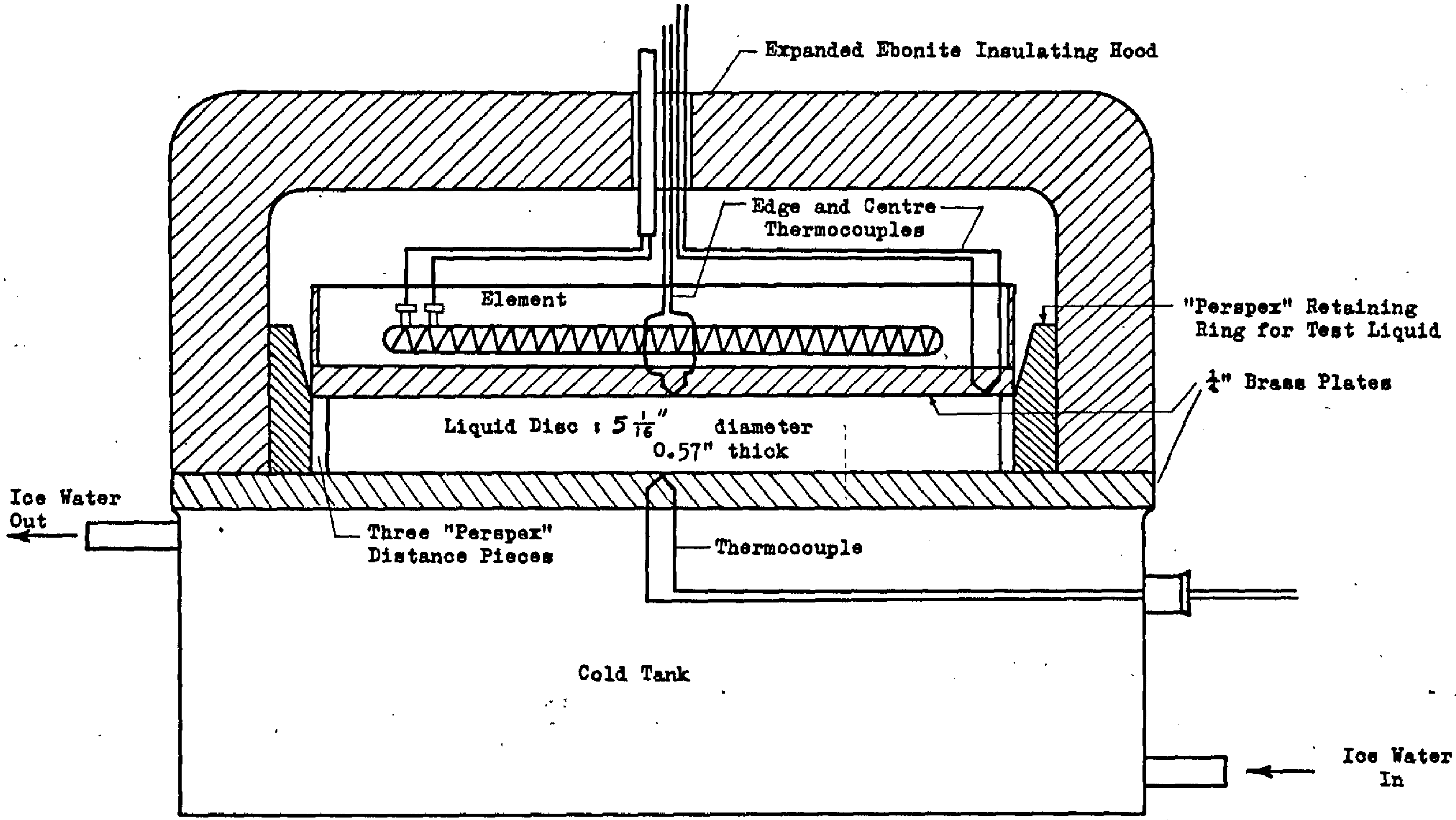
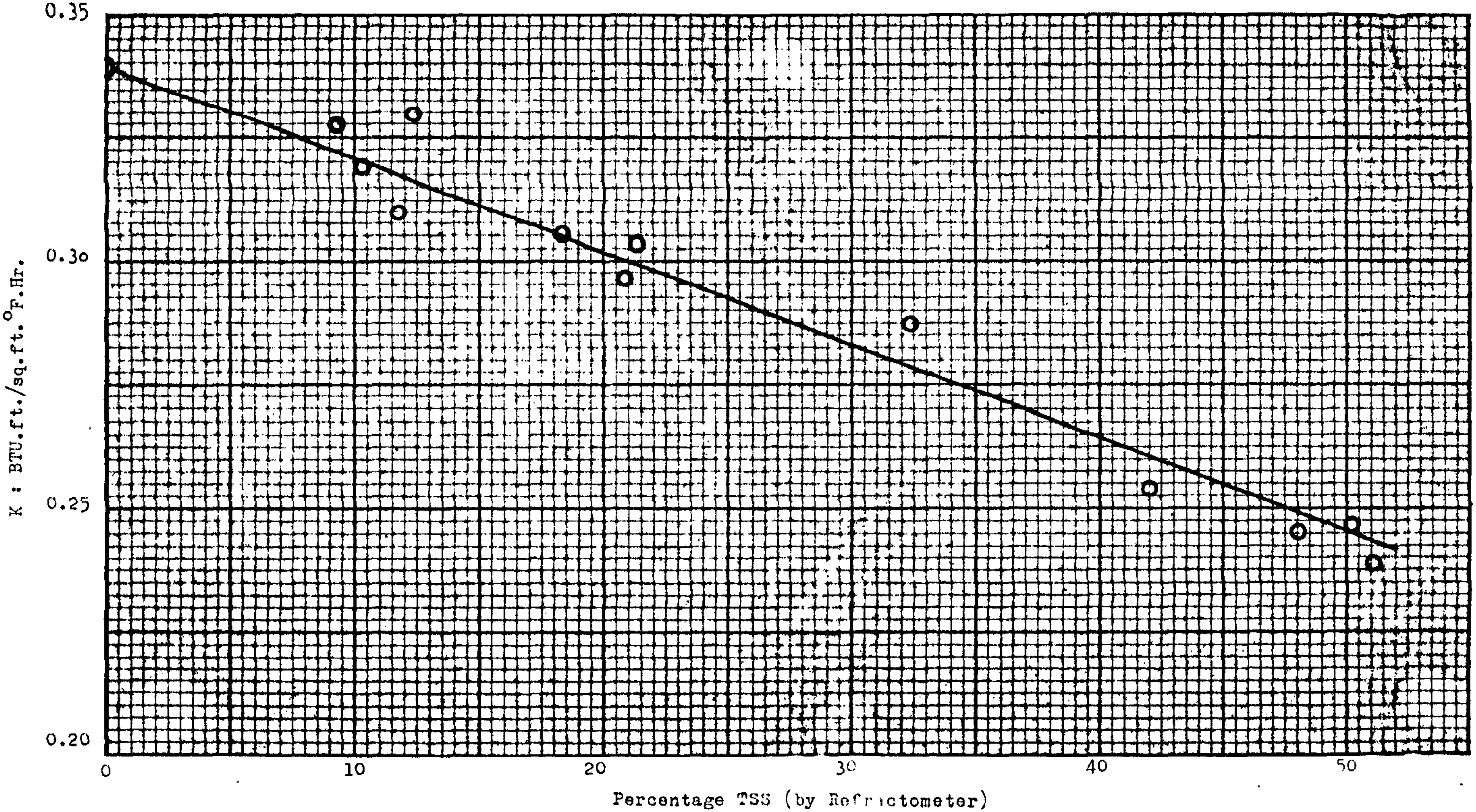


FIG. A.I (4) : APPARATUS FOR DETERMINING THERMAL CONDUCTIVITY OF STICKWATER CONCENTRATE

FIG. A.I (5) : THE THERMAL CONDUCTIVITY OF STICKWATER CONCENTRATE



The results of the tests with stickwater concentrate are shown in Fig. A.I (5). In view of the variable composition of the material, the agreement can be considered satisfactory. A relation of sufficient accuracy for most calculations can be expressed as k (concentrate) = k (water) (1 - 0.0056 x % TSS by refractometer).

3. Density

The density of seven samples of stickwater concentrate at different solids content, as measured by refractometer, was determined, using standard 25 c.c. specific gravity bottles. The determinations were carried out at room temperature (68°F), and are plotted in Fig. A.I (6).

Curve "A" represents the decrease of density of a given sample ("S") as it was thinned down by the addition of distilled water from an original 55.7% TSS by refractometer. The linearity of this curve confirms the soundness of the method.

Curve "B" is the best straight line that could be drawn through the origin and the points determined for the seven random samples. It can be expressed approximately as Density (STICKWATER) = Density WATER (1 + 0.004 x % TSS by refractometer).

This suggests that the solids in the concentrate have a density of 62.4 (1 + 0.4) = 87.5 lbs./cubic foot (at room temperature).

This agrees well with the average particle density for pilchard meal of 88.9 lbs./cubic foot previously reported by Nachenius and Pieterse (4). (It must be remembered that fish meal - as opposed to the solids dissolved in the concentrate - contains a proportion of bony material of greater density; Kaye and Laby (5) quote the density of bones as 1.8 to 2.0).

4. Specific Heat

The method followed was as described in the Dictionary of Applied Physics (6). A total of ten random samples of concentrate were tested at different % TSS (by refractometer), and the results are plotted in Fig. A.I (7). All determinations were carried out between 70° and 80°F.

The curve approximately follows the relation : Specific Heat (concentrate) = Specific Heat (water) (1 - 0.065 x % TSS of concentrate by refractometer).

This agrees reasonably well with the specific heat of about 0.25 BTU/lb. which has at various times been determined for ordinary fish meal.

5. Boiling Point/

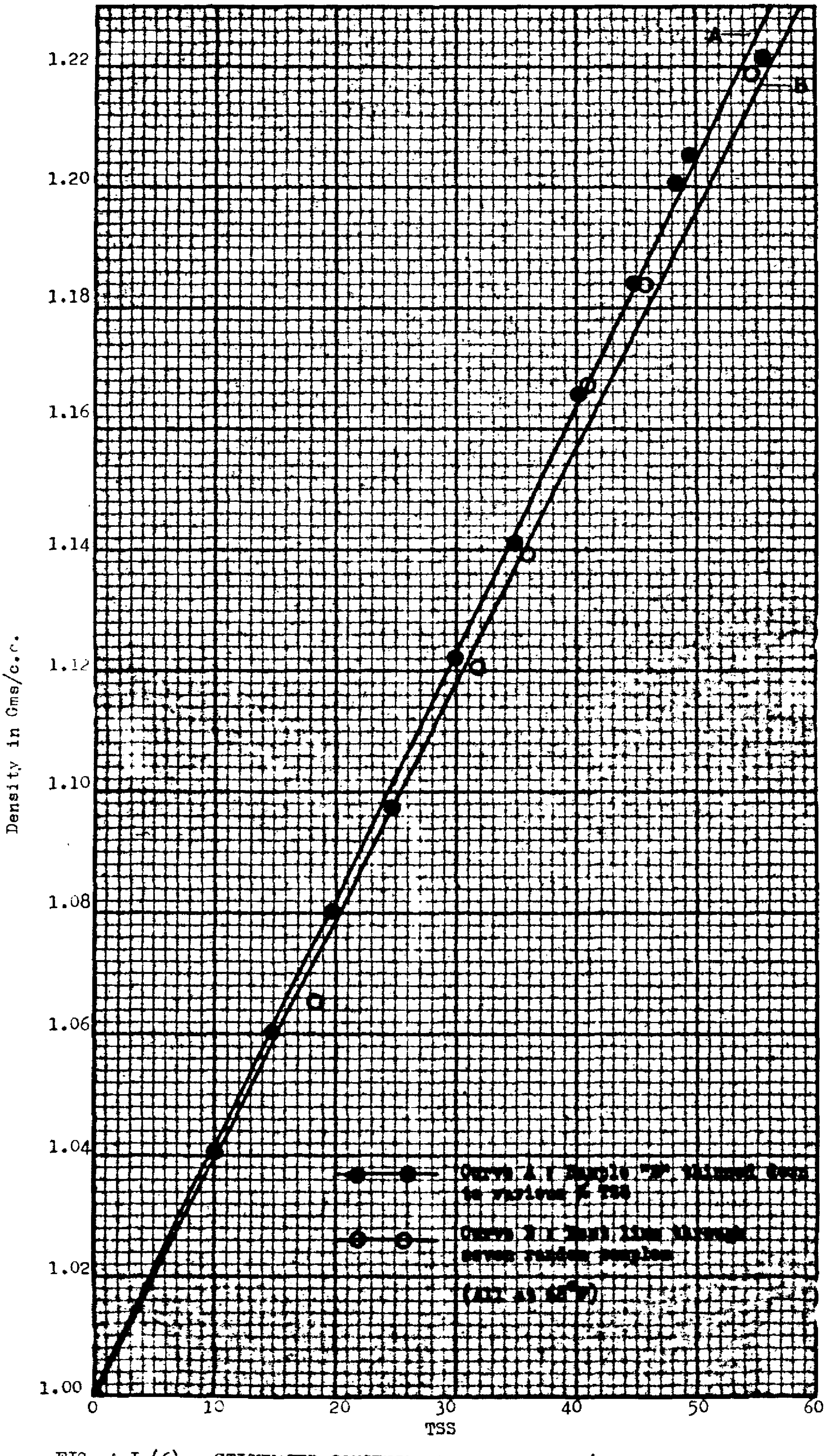


FIG. A.I (6) : STICKWATER CONCENTRATE DENSITY vs. % TSS BY REFRACTOMETER

5. Boiling Point

The object was to estimate the extent to which the boiling point of stickwater concentrate departs from that of water, measured at various sub-atmospheric pressures. About 500 c.c.'s of a typical sample of concentrate of known solids concentration was boiled under reflux in a 1 litre flask held in an electrically heated Isomantle. The pressure was maintained at any selected sub-atmospheric level by applying a constant vacuum. The onset of boiling was observed visually, and the steady temperature of the liquid was measured with a mercury in glass thermometer.

In Fig. A.I (8), the temperature versus pressure curves obtained for two different samples of concentrate at two different percentages TSS have been plotted, together with the saturation line for water as obtained from steam tables. It will be observed that at high solids concentrations and at absolute pressures of from 2 in. to 4 in. of mercury, i.e. under conditions met with in the final effects of commercial plant, the boiling point elevation may be of the order of $10^{\circ} - 15^{\circ}$F. This can have a significant effect on the leval at which boiling commences in the evaporator tubes, as discussed in the text.

B. CORN SYRUP

Solutions of corn syrup in water were in some cases used as working fluids, on account of their Newtonian viscosity. Mention is made in the literature of the variation of the viscosity of these solutions depending on the chemical composition of the corn syrup. It was, therefore, considered safer to measure the viscosity of the syrup supplied by a local manufacturer, rather than to extrapolate Washburn's (7) figures which are usually quoted, but which do not cover the range of temperatures and solids concentrations dealt with in these tests.

The identical method using B.S.S. U-tubes was employed as described for stickwater in the preceding section. The viscosity versus % TSS curve at 120°F for this material is shown in Fig. A.I (9).

The Boiling Point Elevation of this sample, measured at two different concentrations, was also determined as described for stickwater, and is shown in Fig. A.I (10).

C. SUCROSE SOLUTIONS

Solutions of sucrose for use as working fluids were made up by dissolving commercial cane sugar in water. In view of the reputed consistency of the composition of cane sugar, the relevant critical physical data were all taken from the International Critical Tables (8)

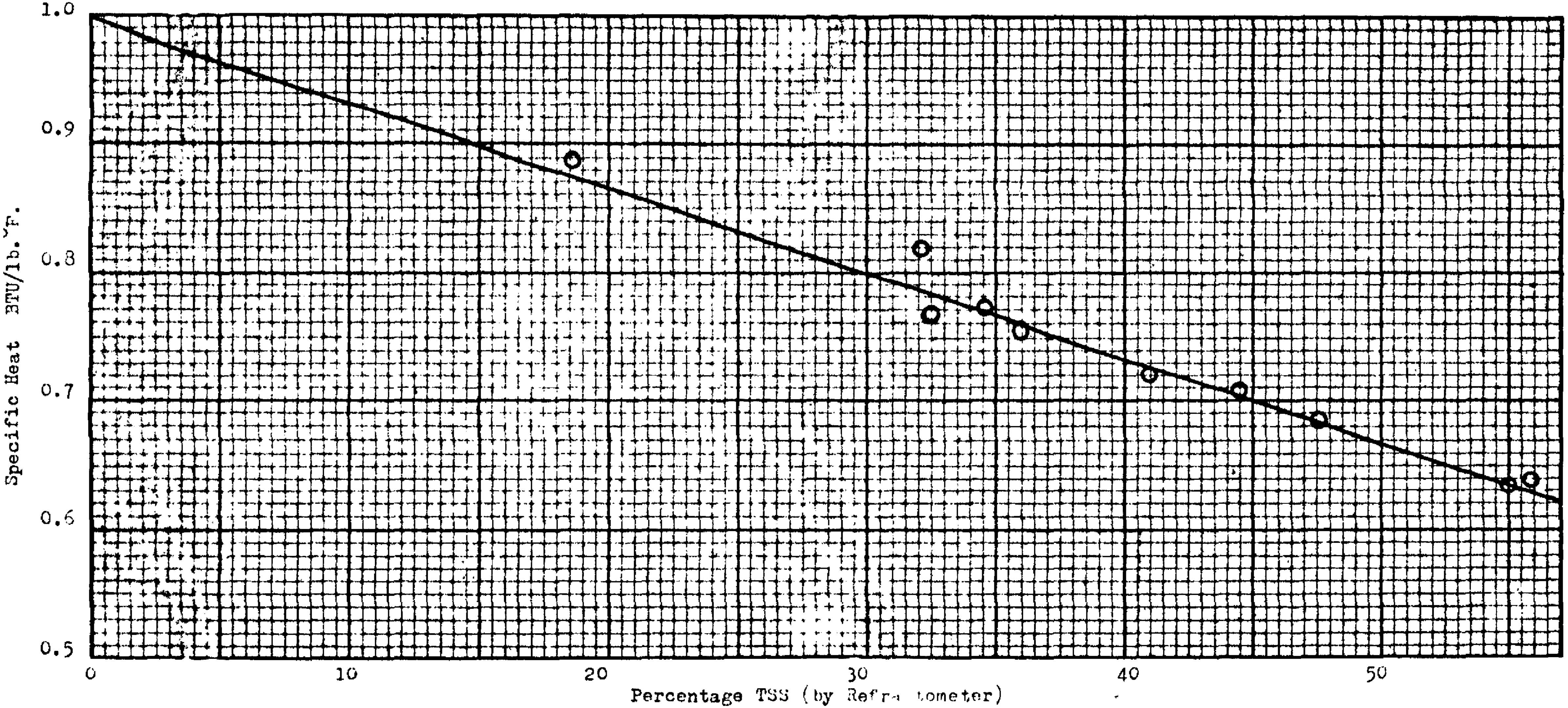


FIG. A.I (7) : THE SPECIFIC HEAT OF STICKWATER CONCENTRATE

(temperature range 70° to 80°F)

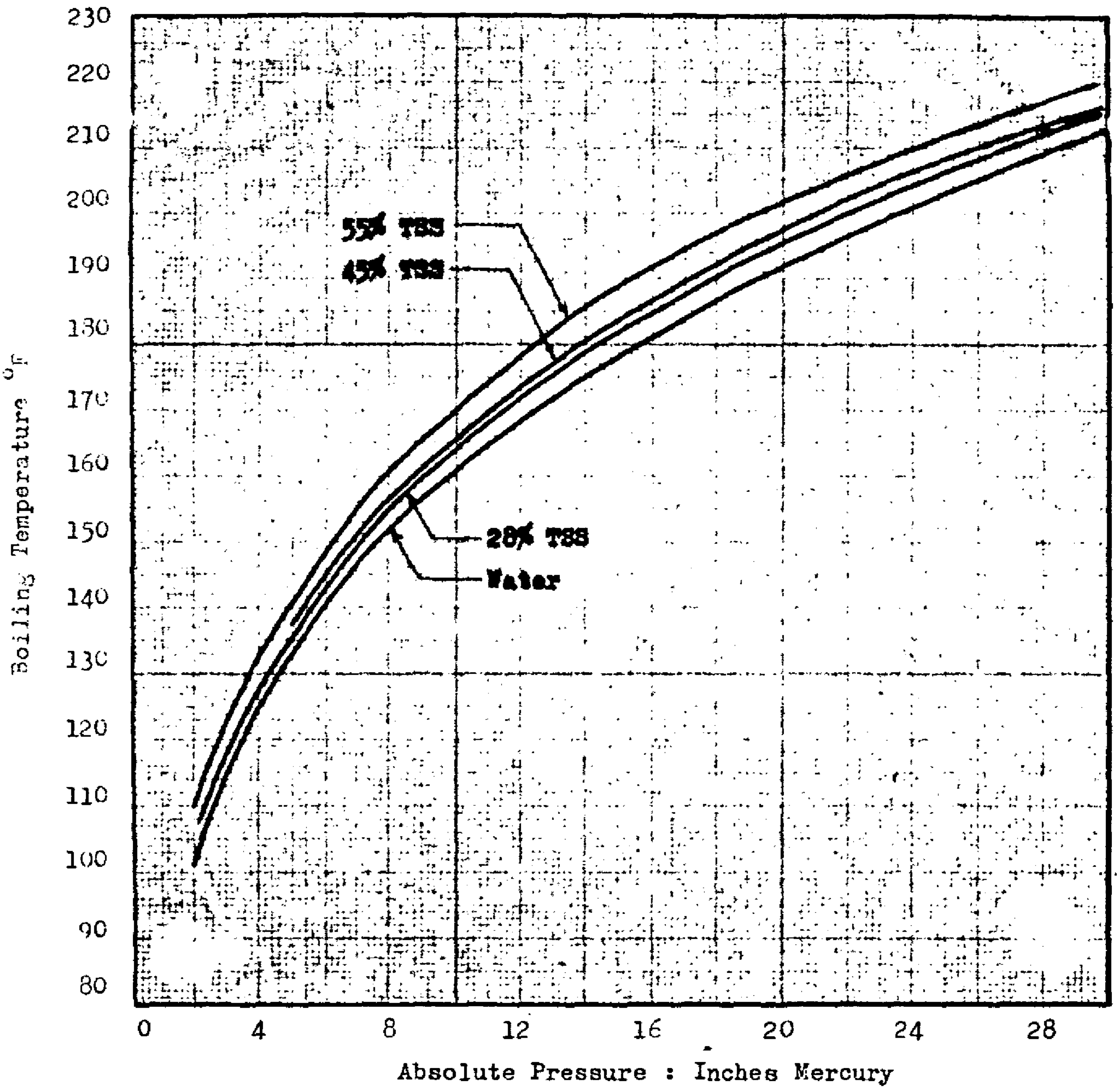


FIG. A.I (8) : SATURATION LINES OF STICKWATER CONCENTRATE

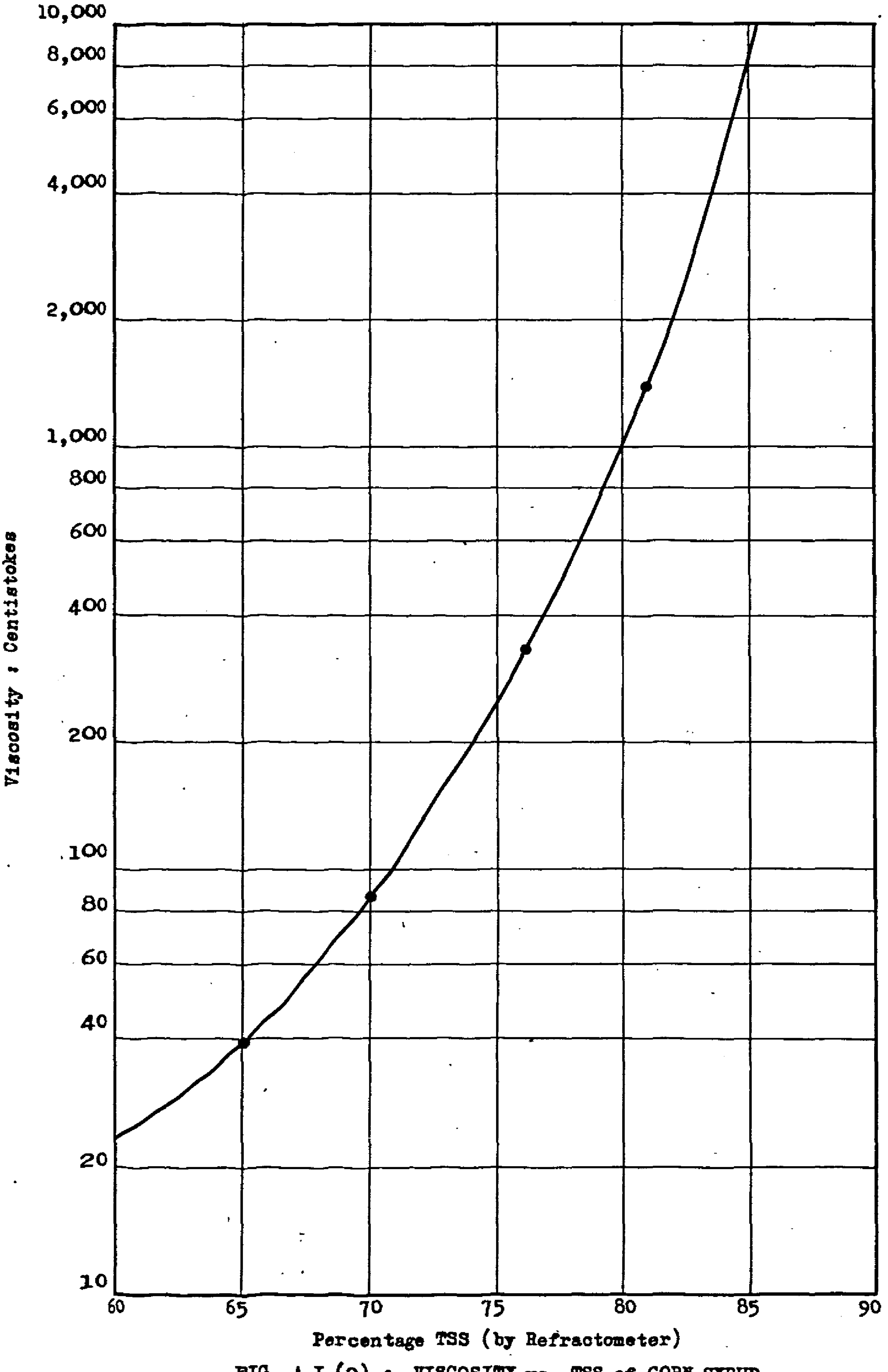


FIG. A.I (9) : VISCOSITY vs. TSS of CORN SYRUP as determined experimentally (at 120°F).

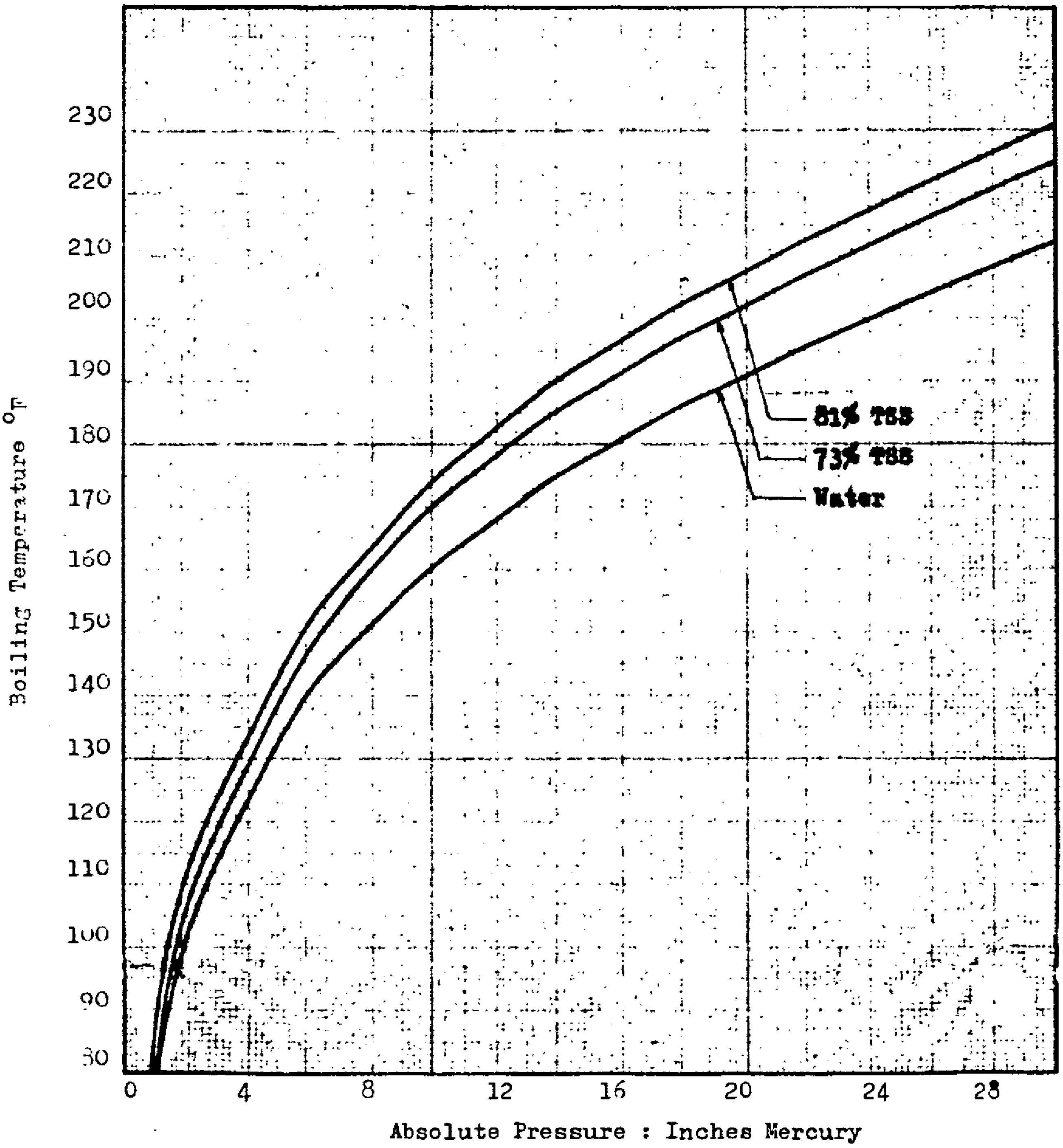


FIG. A.I (10) : SATURATION LINES OF CORN SYRUP

Appendix I

REFERENCES

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5. Kaye, G.W.C., and Laby, T.H., "Tables of Physical and Chemical Constants", Longmans, Green & Co., London (1948) p. 30

6. Glazebrook, Sir Richard, "A Dictionary of Applied Physics", Vol. I, (MacMillan & Co., London (1922) p. 35

7. Washburn, E.W., and Shelton, G.R., "The Viscosities and Surface Tensions of Soda-Lime-Silica Glasses at High Temperatures", University of Illinois Engineering Experiment Station, Bulletin No. 140 (1924) 24.

8. International Critical Tables (McGraw-Hill, New York. (1926)

APPENDIX II

DESCRIPTION OF FLOW METERS

In the few flow measurements on natural circulation evaporators that have been made, for instance by Foust, Baker and Badger (1), the working fluid was always water, and the minimum velocity about 1 ft/sec.

In these cases the velocity could be measured by standard devices such as Pitot Tubes or Orifice Plates. The use of obstruction meters under conditions prevailing in the final effect of a stickwater concentrator must be considered questionable for the following reasons:

1. the maximum dynamic "head" of the stream is too small, particularly at the low circulating velocities occurring in practice. At a liquid velocity of say 0.1 ft/sec., it is $\frac{0.1^2}{32 \times 2} = 0.000156$ feet, which, for a concentrate density of say 70 lbs/cubic foot, is equal to a pressure of 0.011 lbs/sq.ft. The measurement of pressures of this order not only requires the use of an advanced and costly (Bell-type) micromanometer, but the differential pressure readings due to the flow of the fluid would probably be obscured by variations in the static hydraulic head on the pressure tappings;
2. uncertainty about the rheological behaviour of stickwater concentrate (see Appendix I) would necessitate the re-calibration of the orifice plate or similar device for each new sample of concentrate used;
3. the gumminess of the thick concentrate and the presence of suspended solids must be expected to lead to obstruction of the pressure tappings and other fittings; and
4. the pressure drops due to the obstructing device are undesirable in view of the low propelling force which maintains the flow in natural circulation evaporators.

The only reliable method of measuring the velocities of liquids such as stickwater concentrate without interfering with their flow appears to be by magnetic induction. Suitably sized flow meters of this type were offered by several well-known instrument makers, but their cost is of the order of R1,000, which is more than could reasonably be spent for the tests under discussion.

However, rather than abandon the idea of measuring, or at least estimating, the circulation velocity, a number of other methods were tested.

A. Cold/

A) Cold water injection.

This is illustrated diagrammatically in Fig. A.II (1). The technique consists of injecting about 10 to 15 cubic centimetres of ice cooled water into the return tube at point "B". The chilled liquid moves with the main stream of the liquid, and produces an observable deflection of the needle of the potentiometer via suitably located thermocouples. The timed interval between the injection of the liquid at point "B" and the deflection of the galvanometer in response to temperature changes at points "C" or "D" is an indication of the velocity of the liquid in the tube.

An analysis was made by the writer (2) of the correlation of the interval between injection and deflection, and the known liquid velocity being maintained by pumping at a measured rate, but the error could not be reduced to less than about ± 20%. The errors are chiefly due to the operators' reaction lag, and turbulence generated by the act of injection.

To circumvent all theoretical complications a series of tests was carried out with water and stickwater concentrate of 40% TSS, in which the injection interval was timed, while the true circulating velocity was being measured simultaneously by collecting and weighing the discharge from the circulating pump. A representative set of readings is listed in Tables A.II (I) and A.II (II).

The calculated liquid velocities were plotted against the reciprocal of the timed interval, and the best straight line drawn through all the points, as shown in Fig. A.II (2). In converting the weight of collected liquid to volume, the densities of the water and the concentrate were taken as 62.4 and 70 lbs./cubic foot respectively. It will be observed that the maximum deviation of any two readings from the straight line is about 0.2 ft. per sec., which at the corresponding true velocity of 3 ft. per sec. involves an error of about 7 per cent. It is also noticeable that even with concentrate of 40% TSS the observed intervals are extremely close to those for water.

Selected readings taken on the straight line in Fig. A.II (2) were corrected for the true density of water at 160°F (61.0 lbs/cubic foot instead of 62.4 lbs./cubic foot) and replotted as shown in Fig. A.II (3). This curve was used whenever liquid velocity determinations were made up to Run No. 110.

Tests to automate the timing of the interval between water injection and deflection of the galvanometer needle were only moderately successful, and the improvement in accuracy as compared with manual timing did not warrant its perfection.

Velocity measurement by water injection proved useful for estimating the circulating velocity over a wide range of conditions, but it had the

following/

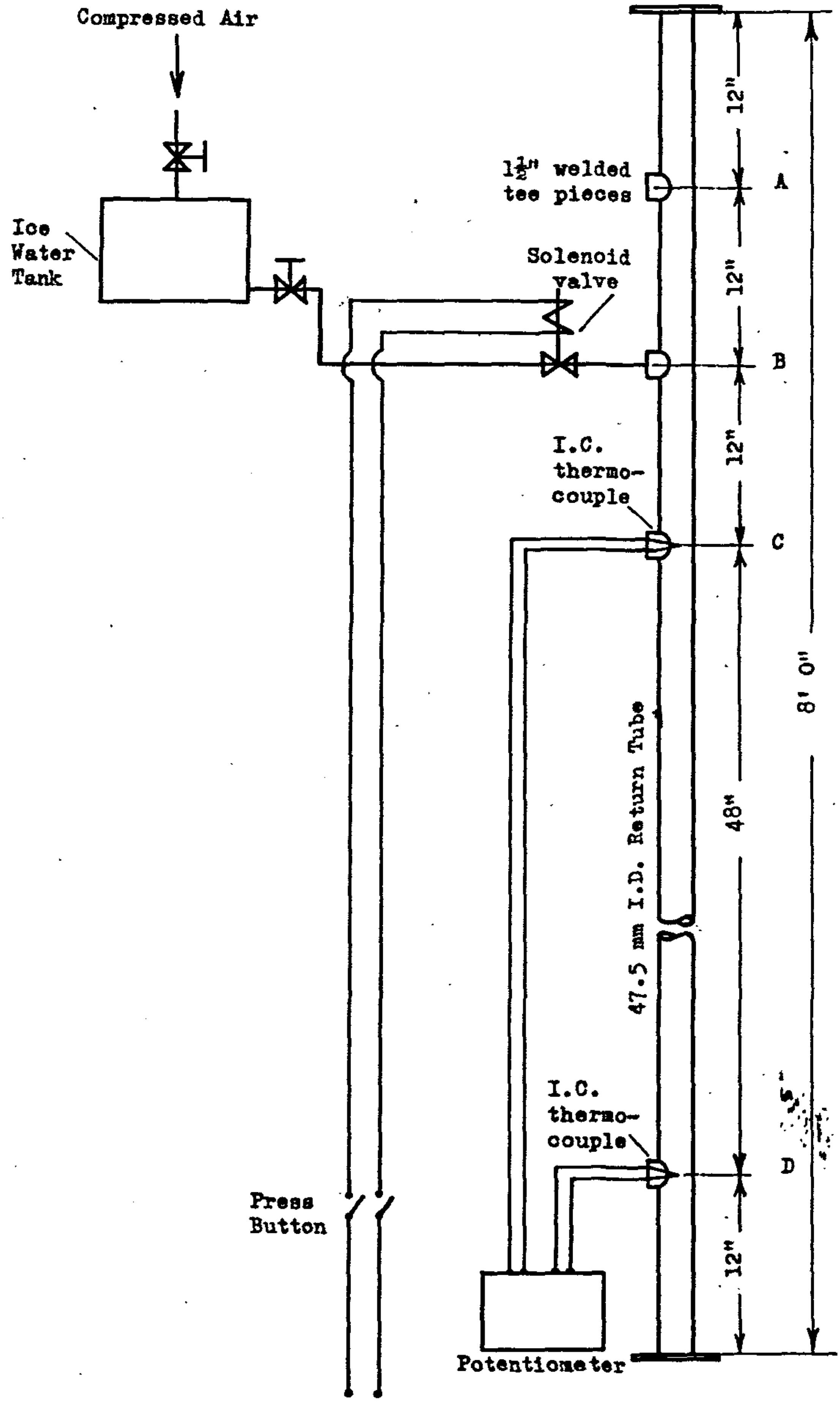


FIG. A.II (1). DIAGRAMMATIC ARRANGEMENT FOR VELOCITY MEASUREMENT BY COLD WATER INJECTION

TABLE A.II (I)

CALIBRATION OF COLD WATER INJECTION METHOD FOR LIQUID VELOCITY MEASUREMENT IN EXPERIMENTAL APPARATUS

Cold water pressure : 10–15 psi. Liquid temperature approx. 160°F.

W A T E R

Test No	Measured Pump Discharge	Calculated Equivalent Liquid Velocity in 35 mm. tube		Timed Interval to travel 5 ft. in 47.5 mm. tube	
	lbs/sec.	ft/sec.	Ave.ft/sec.	Secs.	Ave. secs.
1	0.976 0.963	1.55 1.53	1.54	4.7 4.7 4.8 4.6	4.7
2	1.44 1.39 1.39	2.29 2.22 2.22	2.24	3.5 3.5 3.5 3.4	3.5
3	1.71 1.67	2.72 2.66	2.69	2.8 2.8 2.9 2.9	2.85
4	1.77 1.69 1.78 1.77	2.81 2.76 2.83 2.81	2.80	2.7 2.7 2.7 2.7	2.7
5	1.95 1.95	3.10 3.10	3.10	2.5 2.5 2.4 2.5	2.5
6	2.13 2.20	3.40 3.51	3.45	2.3 2.4 2.4 2.3	2.35
7	2.46 2.37 2.40	3.94 3.76 3.81	3.84	2.0 2.0 2.1 2.1	2.05
8	3.08 3.00	4.81 4.77	4.79	1.6 1.7 1.7 1.7	1.7
9	3.41 3.46	5.41 5.49	5.45	1.5 1.5 1.5 1.4	1.5
10	0.45 0.46	0.71 0.73	0.72	9.9 9.7 9.8 9.7	9.8

TABLE A.II (II)

CALIBRATION OF COLD WATER INJECTION METHOD FOR LIQUID VELOCITY MEASUREMENT IN EXPERIMENTAL APPARATUS

36-44% TSS (By Refractometer) Temperature approx 160°F. Density approx. 70 lbs/cu.ft.

CONCENTRATE

Test No	Measured Pump Discharge	Calculated Equivalent Liquid Velocity in 35 mm. tube		Timed Interval to travel 5 ft. in 47.5 mm. tube	
	lbs/sec.	ft/sec.	Ave. ft/sec.	Secs.	Ave. secs.
1	1.11 1.13	1.54 1.56	1.55	4.7 4.5 4.5 4.6 4.5	4.55
2	1.70 1.67 1.66	2.35 2.30 2.28	2.31	3.4 3.3 3.3 3.3	3.3
3	2.05 2.05	2.82 2.82	2.82	2.9 2.8 2.8 2.8	2.8
4	2.15 2.08	2.96 2.86	2.91	2.7 2.7 2.7 2.8	2.7
5	2.28 2.24	3.15 3.09	3.12	2.6 2.5 2.5	2.5
6	2.56 2.55	3.54 3.53	3.53	2.2 2.3 2.3	2.3
7	2.88 2.82	3.98 3.88	3.93	2.0 2.1 2.0 2.0	2.0
8	3.37 3.31	4.66 4.58	4.62	1.9 1.8 1.8 1.8	1.8
9	3.88 3.90	5.34 5.36	5.35	1.5 1.7 1.5 1.5	1.5
10	0.34 0.35 0.33	0.47 0.48 0.46	0.47	11.6 11.4 11.7 12.4 12.0 15.4 14.6 14.4	12.9

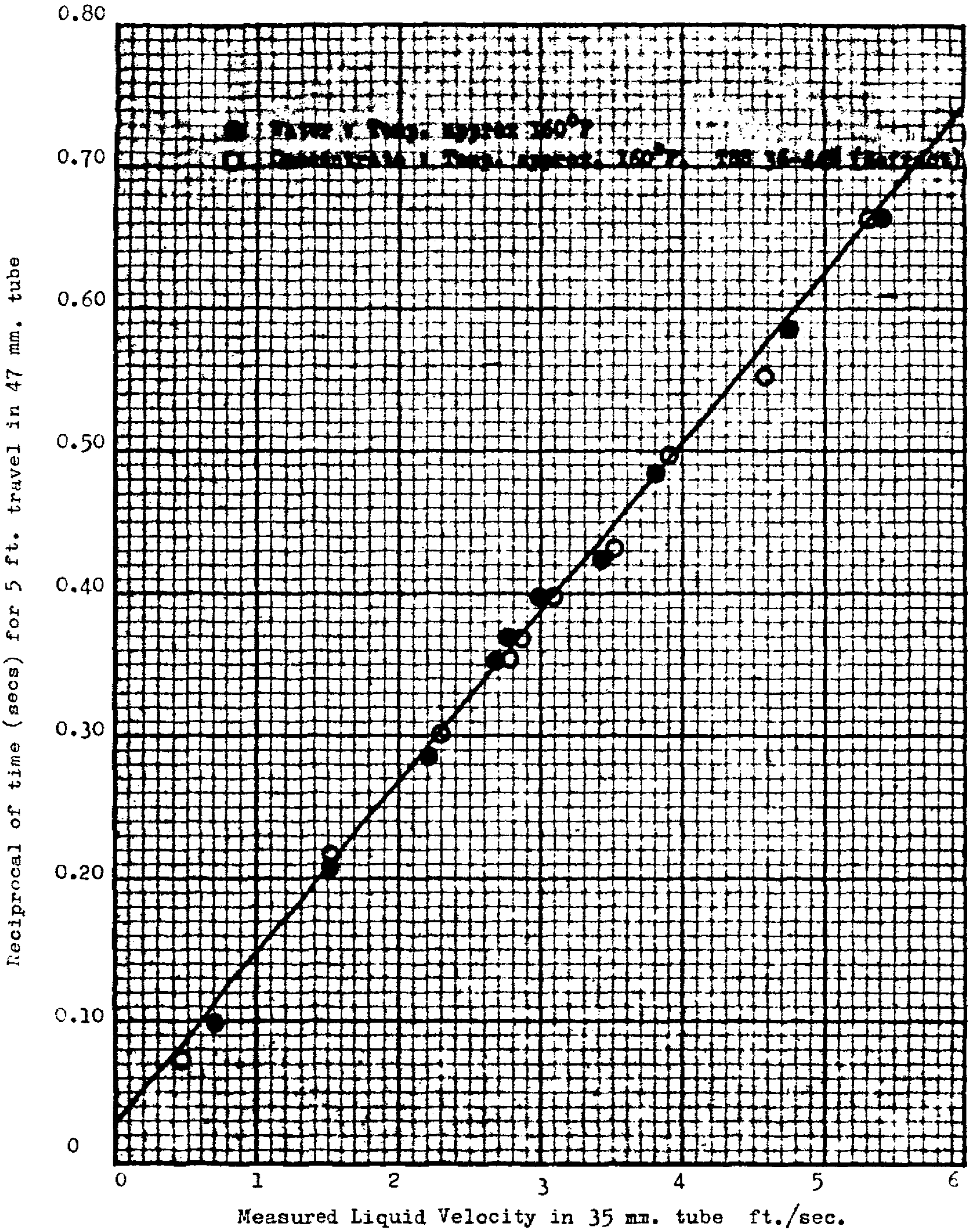


FIG. A.II (2) : VELOCITY MEASUREMENT BY WATER INJECTION

MEASURED VELOCITY vs. RECIPROCAL OF TIMED INTERVAL.

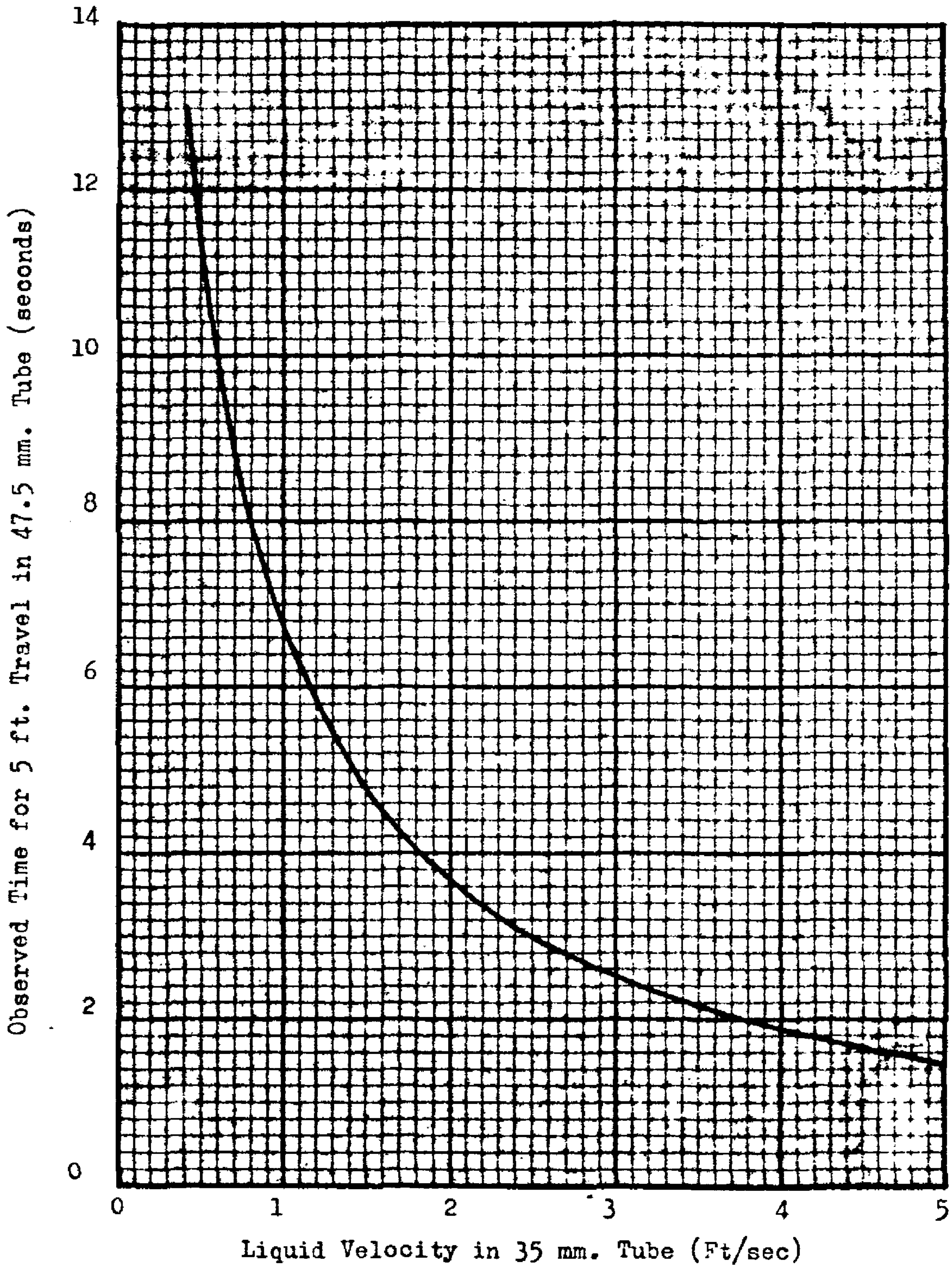


FIG. A.II (3) : VELOCITY MEASUREMENT BY WATER INJECTION

CALIBRATION CURVE

following limitations:

1. it was unreliable when the timed interval exceeded about 12 seconds (i.e. when the liquid velocity in the heated tube dropped to below 0.5 ft. per sec.). The galvanometer needle deflection then became sluggish and clear readings were difficult to obtain. Sometimes better readings could be obtained using the thermocouple at point "C" - 12 inches below the injection point instead of 60 inches as "D" - but even this did not extend the range of liquid velocities that could be measured with confidence to below about 0.3 ft. per sec.

   The drop in sensitivity with decreasing flow rates was probably due to incomplete mixing of the injected water with the concentrate and convection of the cold liquid within the (slowly moving) bulk. Both effects would be difficulty to remedy;
2. it can be used only for taking spot readings, instead of continuous indication. This is particularly bothersome if the flow is intermittent, as for instance when "surging" occurs in the heated tube. It is then impossible to arrive at an acceptable average velocity reading or to estimate the extent of the velocity variations without taking a large number of readings, which is impractical;
3. the injection even of small volumes of cold water is undesirable while a heat transfer test is in progress. Although the dilution and cooling due to a single injection are negligible (about 0.2% TSS and 4 BTU per injection when operating at 50% TSS), the error becomes appreciable as the number of readings increases. For this reason, velocity readings were generally only taken before and after a particular run was completed.

In view of these difficulties another method of flow measurement was devised and tested.

B) Torsion Vane.

Details of this device are shown in Fig. A.II (4). The shaped vane is suspended under tension by a steel wire, which is anchored at both ends of the return tube. Tension is maintained by a fairly stiff spring. Rotation of the vane is observed through the viewing windows at "A" and "B". The viewing windows consist of short glass rods with polished ends passing through rubber stoppers. The rotation of the coarse indicator at "A" is one fifth of that of the vane itself at "B", as the ratio of their distances from the upper suspension point is 1 : 5. The coarse indicator was useful when the vane completed more than one revolution.

The device was calibrated against the weighed discharge from a circulating pump as described in the preceding section, and the test data are shown in Table A.II (III). As the device is a form of obstruction meter,

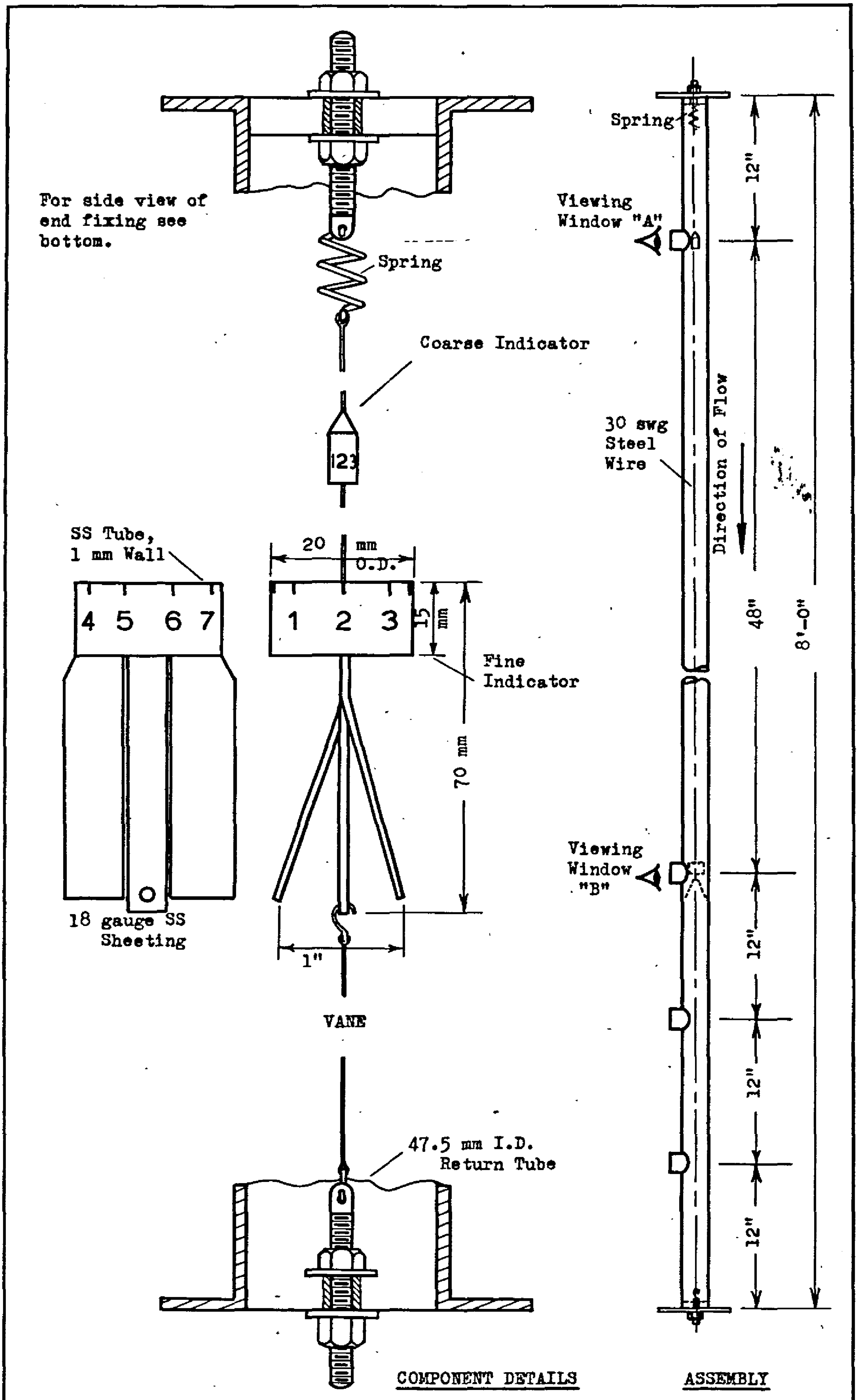


FIG. A.II (4). ROTATING VANE FLOW INDICATOR

TABLE A.II (III)

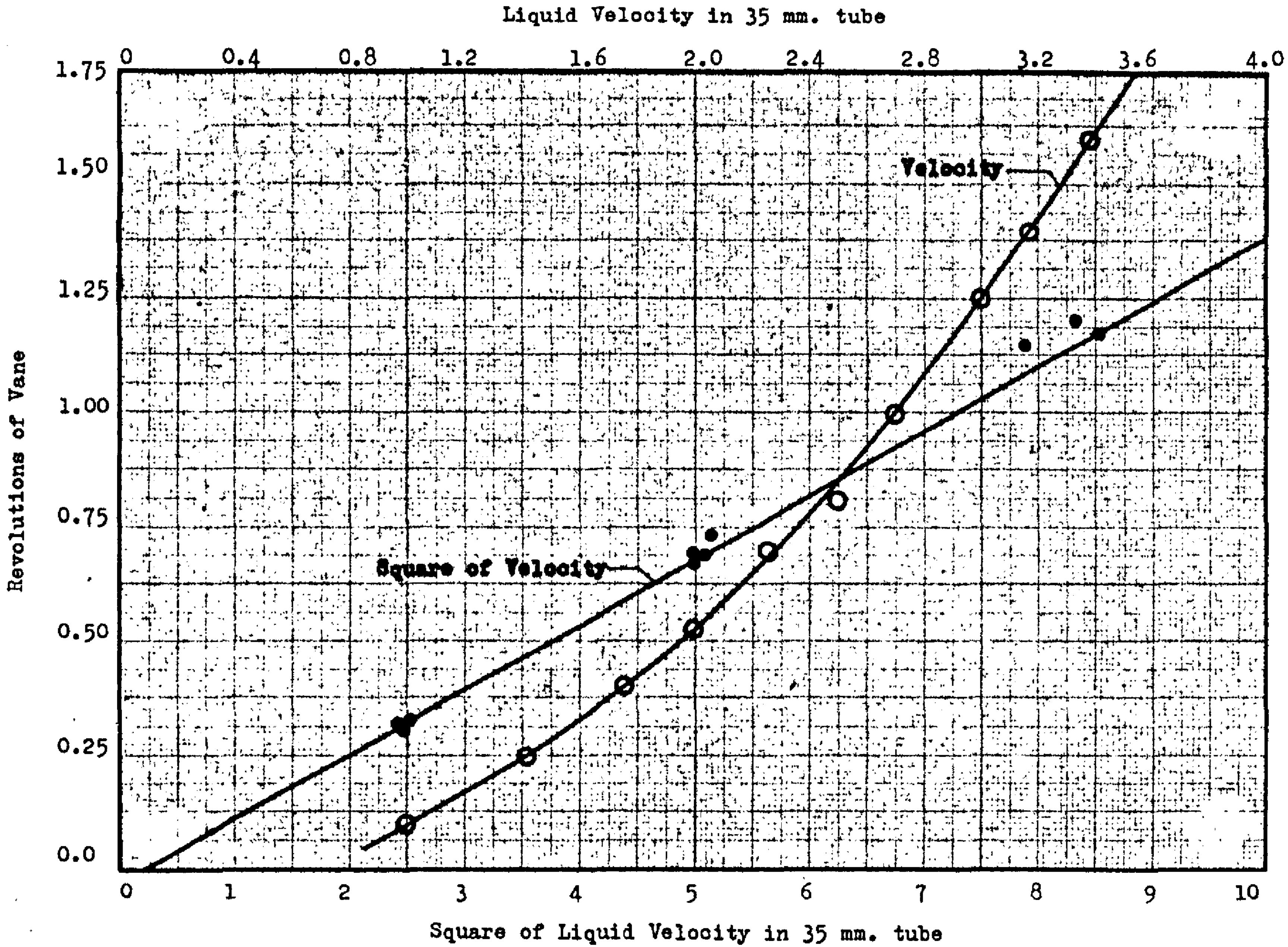
CALIBRATION OF TORSION VANE USED FOR

LIQUID VELOCITY MEASUREMENTS IN EXPERIMENTAL APPARATUS

WATER

TEST	WATER TEMP. °F	MEASURED DISCHARGE FROM PUMP Ave. lbs/sec.	CALCULATED EQUIVALENT LIQUID VELOCITY IN 35 mm. TUBE ft/sec.	OBSERVED VANE DEFLECTION Revs.
1	75	3.71	5.75	3.85
2	75	3.17	4.92	2.45
3	75	2.31	3.58	1.80
4	75	1.88	2.92	1.18
5	75	1.44	2.23	0.69
6	75	1.01	1.56	0.31
7	84	2.27	3.54	1.78
8	84	1.85	2.89	1.21
9	84	1.44	2.25	0.69
10	84	1.01	1.57	0.31
11	194	2.30	3.68	1.85
12	194	1.42	2.27	0.73
13	194	0.98	1.57	0.33
14	73	2.26	3.52	1.72
15	73	1.80	2.81	1.15
16	73	1.44	2.24	0.68
17	73	0.995	1.55	0.33

FIG. A.II (5) : VELOCITY MEASUREMENT BY TORSION VANE : CALIBRATION CURVE



The deflection of the vane would be expected to be roughly proportional to the square of the liquid velocity. This was confirmed by plotting the relevant curve as shown in Fig. A.II (5). The experimental points are found to be very nearly on a straight line passing through the origin, and a calibration curve was drawn from points on this line.

This flow indicator proved reliable and its calibration did not change significantly during several months of use. It was free of several of the major shortcomings of the injection method; it gave continuous instead of intermittent readings; it involved no cooling or dilution of the working fluid, and it could be used at low velocities. But when the viscosity of the medium (stickwater concentrate or corn syrup) exceeded about 100 centipoises, i.e. when the flow pattern in the tube was fully laminar, the movement of the vane became irregular, and no longer corresponded to the calibration. This flow meter was nevertheless successfully used to measure the flow rates in Runs Nos. 111 to 132 (air-lift circulation with water).

REFERENCES

1. Foust, A.S., Baker, E.M. and Badger, W.L., Trans. Am. Inst. Chem. Engrs., 35, (1939), 45 - 71

2. Nachenius, R.J., "Method for Estimating the Liquid Velocity in Concentrator Tubes", Twelfth Annual Report, Fishing Industry Research Institute, University of Cape Town, Rondebosch (1958) p. 48.

APPENDIX III

SPECIMEN SETS OF READINGS AND CALCULATIONS

The tests which are described in this section have been selected because they illustrate the most important features of different groups of experiments.

Test (a) is a typical example of how the apparatus was operated in triple effect. Particulars are given of all readings and calculations, and of the method by which heat balances were established.

Test (b) is of interest because it shows that airlift circulation, even when applied only to the final effect of a triple effect plant, can cause a re-arrangement of the heat transfer pattern of the plant as a whole.

Test (c) illustrates how single - as opposed to triple - effect operation facilitates a closer control of the test variables, reduces the number of readings to be taken and produces more consistent results.

Test (d) shows how the isolation of individual thermocouples described in Fig. 8 Chapter IV, made it possible to measure the true tube wall temperatures, instead of the average temperature registered by the four thermocouples spaced equidistantly along the length of the tube. The readings in this test may be compared with the corresponding readings in test (c).

N.B. Although the runs within each test were made consecutively, the run numbers were subsequently re-arranged to suit the text.

Test (a)/

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N.B. Although the runs within each test were made consecutively, the run numbers were subsequently re-arranged to suit the text.

Test (a)/

RUN No. 47 (CONTROL) : READINGS

Temperatures and Pressures

Time	First Effect			Second Effect			Third Effect			
	Jacket		Liquid	Jacket		Liquid	Jacket		Liquid	
	Press psi	Temp °F	Temp °F	Press psi	Temp °F	Temp °F	Press, in. vac.	Temp °F	Press, in. vac.	Temp °F
10.00	29.5	277	239	9.0	236	193	11	192	28.1(1)	104
10.30	30.0	277	237	9.0	237	194	11	194	28.1	107
11.00	29.0	276	232	7.5	233	192	11	191	28.1	103
11.30	30.0	277	236	9.0	236	196	10	197	28.1	108
Ave. Temp. °F		277	236		236	194		194		106
Ave. Temp. Diff. °F		41			42			88		

Condensate Collected

Period	From First Effect Jacket	From Second Effect Jacket	From Third Effect Jacket	From Condenser
10.00 – 10.30	17.6 lbs.	11.4 lbs.	8.8 lbs.	8.8 lbs.
10.30 – 11.00	17.8 lbs.	11.0 lbs.	8.8 lbs.	9.3 lbs.
11.00 – 11.30	17.9 lbs.	11.2 lbs.	8.8 lbs.	8.8 lbs.
Average distillation rate: lbs/hr	35.5	22.4	17.6	17.9
"Flash": lbs/hr(2)	1.8	0.5	0.4	--
Total	37.3	22.9	18.0	17.9

Material Balance

Weight of Liquid(3)	First Effect	Second Effect	Third Effect
entering :	72 lbs/hr.	49 lbs/hr.	31 lbs/hr.
leaving :	49 lbs/hr.	31 lbs/hr.	13 lbs/hr.

Concentration (TSS% by Refractometer)

Liquid	Raw Stickwater	in First Effect	in Second Effect	in Third Eff.
% TSS	8.3	15	22	46

Average feed temperature : 190°F

Average ambient temperature : 69°F

Run No. 47 (contd.)

Velocity Measurements

	First Effect	Second Effect	Third Effect
Timed interval over five feet, except readings (a) to (e), which were timed over one foot, as velocity too low.	4.0 secs. 4.0 " 4.0 " 4.0 " 4.0 "	4.5 secs. 4.5 " 5.0 " 4.5 "	8.0 (?) secs. 9.0 (?) " 3.0 (a) " 2.5 (b) " 2.5 (c) " 3.0 (d) " 2.5 (e) "
Average	4.0 secs.	4.5 secs.	2.7 secs.
Reading converted to true velocity in heated tube by using calibration curve	1.95 ft/sec.	1.7 ft/sec.	0.6 ft/sec.

RUN No. 64 (TEST) : READINGS

Temperatures and Pressures

Time	First Effect			Second Effect			Third Effect			
	Jacket		Liquid	Jacket		Liquid	Jacket		Liquid	
	Press psi	Temp °F	Temp °F	Press psi	Temp °F	Temp °F	Press in. vac	Temp °F	Press in. vac	Temp °F
9.45	32.0	275	236	11.0	237	195	9.0	194	28.1	96
10.15	31.0	273	235	11.5	236	201	6.0	200	27.9	100
10.45	31.0	271	234	11.5	235	200	6.0	199	27.8	103
11.15	30.5	271	230	10.0	230	194	8.0	194	27.8	99
Ave. Temp. °F		273	234		234	198		197		99
Ave. Temp. Diff. °F		39			36				98	

Condensate Collected

Period	From First Effect Jacket	From Second Effect Jacket	From Third Effect Jacket	From Condenser
9.45 - 10.15	21.1 lbs.	13.4 lbs.	12.4 lbs.	13.2 lbs.
10.15 - 10.45	20.6 lbs.	13.1 lbs.	12.6 lbs.	13.7 lbs.
10.45 - 11.15	21.1 lbs.	13.0 lbs.	12.2 lbs.	13.0 lbs.
Average distillation rate: lbs/hr	41.9	26.3	24.6	26.6
"Flash": lbs/hr	2.1	0.6	0.5	-
Total	44.0	26.9	25.1	26.6

Run No. 64 (contd.)

Material Balance

Weight of Liquid	First Effect	Second Effect	Third Effect
entering:	95 lbs/hr.	68 lbs/hr.	45 lbs/hr.
leaving :	68 lbs/hr.	45 lbs/hr.	19 lbs/hr.

Concentration (TSS% by Refractometer)

Liquid	Raw Stickwater	in First Effect	in Second Effect	in Third Eff.
% TSS	8.1	11.7	19.5	46.7

Average Feed Temperature : 190°F

Average Ambient Temperature : 78°F

Velocity Measurements

	First Effect	Second Effect	Third Effect
Timed interval over five feet	4.5 secs. 4.5 "	5.5 secs. 5.5 " 5.0 " 5.0 "	5.5 secs. 6.0 " 6.0 " 6.0 "
Average	4.5 secs.	5.25 secs.	5.9 secs.
Reading converted to true velocity in heated tube by using calibration curve	1.7 ft/sec.	1.4 ft/sec.	1.22 ft/sec.

HEAT BALANCE

Run No. 47 (Control)

Heat Entering and Leaving[4] BTU/hour	First Effect IN	First Effect OUT	Second Effect IN	Second Effect OUT	Third Effect IN	Third Effect OUT
With stickwater feed and discharge[5]	10,650	10,000	9,300	5,020	4,870	960
Transmitted through heated surface[6]	43,750	26,550	26,550	20,600	20,600	19,800
with condensate from jacket[7]		9,150		4,670		2,920
Radiation "A"[8]		2,340		1,750		520
Radiation "B"[9]		6,240		2,170		1,630
Totals[10]	54,400	54,280	35,850	34,210	25,470	25,830
Heat Transmitted:						
(a) Calculated on condensate from steam jacket[11]	28,360		19,710		16,050	
(b) Calculated on condensed distillate[12]	28,220		17,390		16,290	
Average of (a) and (b)[13]	28,290		18,550		16,170	
Apparent U value[14] BTU/sq.ft. °F hr.	256		164		68	
Gross U value[15] BTU/sq.ft.°F.hr	136.5					

Run No. 64 (0.3% Rhozyme B6 added)

Heat Entering and Leaving BTU/hour	First Effect IN	First Effect OUT	Second Effect IN	Second Effect OUT	Third Effect IN	Third Effect OUT
With stickwater feed and discharge	14,050	13,850	13,200	7,500	7,250	1,270
Transmitted through heated surface	51,600	31,200	31,200	28,800	28,800	29,400
With condensate from jacket		10,600		5,430		4,140
Radiation "A"		2,180		1,680		290
Radiation "B"		5,850		2,030		1,560
Totals	65,650	63,680	44,400	45,440	36,050	36,660
Heat Transmitted:						
(a) calculated on condensate from steam jacket	35,150		23,740		23,100	
(b) calculated on condensed distillate	33,060		23,830		23,490	
Average of (a) and (b)	34,100		23,780		23,300	
Apparent U value BTU/sq.ft.°F.hr	324		244		88	
Gross U value BTU/sq.ft.°F.hr.	173					

NOTES

(Referring to Numbers on Data Sheets)

1. Barometer readings were only recorded from Run 77 onwards, and the absolute pressure in the third effect plenum is therefore not known. The tendency observed in later tests for the liquid temperature in the third effect receiver to be slightly higher than the saturation temperature of water at the corresponding absolute pressure, is due to the boiling point elevation of the concentrate discussed in Appendix I.

2. Flash in the first effect is due to the steam trap discharging jacket condensate to the atmosphere. In preliminary tests the temperature of the condensate just ahead of the steam trap was measured, and found to range from 260° to 265°F. The average temperature drop to atmosphere was therefore about 50°F, and "flash" was estimated as $\frac{\text{weight of condensate collected x 50}}{1,000}$ lbs/hr.

   The temperature of the condensate just ahead of the second effect steam trap was found to be about the same as in the jacket itself, and "flash" was therefore calculated as

   $$\frac{\text{weight of condensate collected x } (236 - 212)}{1,000} \text{ lbs/hr.}$$

   The temperature drop between the third effect jacket and the collecting flask, which was kept under about 17 in. Hg. vacuum, was estimated as 20°F, and "flash" was accordingly calculated as

   $$\frac{\text{weight of condensate collected x 20}}{1,000} \text{ lbs/hr.}$$

3. The weight of liquid entering the first effect was estimated by considering the whole apparatus as being operated as a single effect evaporator, i.e.

   $$= \text{Total weight of condensate} \times \frac{\text{\%TSS of Concentrate}}{\text{\%TSS of Concentrate} - \text{\%TSS of Raw Stickwater}}$$

   in this instance: $58.8 \times \frac{46}{46 - 8.3} = 72$ lbs/hr.

   The weights of liquid entering and leaving the second and third effects were calculated by subtracting or adding the weights of condensate collected from the corresponding steam jackets.

4. All sensible and total heats were calculated as from 32°F. As the sensible heats are in all cases small compared with the latent heats, the specific heat of all liquids was taken constant as 1 BTU/lb.

5. Calculated as (weight of liquid) x (Temperature − 32), allowance was made for cooling of the liquid in the feed lines. On the basis of a preliminary test, the temperature of the liquid entering the first effect was taken as 10°F below the average temperature in the feed tank. A similar allowance of 10°F was made for the liquid transferred to the second effect, and 5°F for the liquid transferred to the third effect.

6. Calculated as (Total weight of condensate) x (Total heat; obtained from Steam Tables).

7. Calculated as (Total weight of condensate) x (Steam jacket temperature − 32°F).

8. Radiation "A" signifies the convection and radiation heat loss from the body of the evaporator excluding the steam jacket. This was estimated by passing dry steam through the apparatus and collecting the condensate during an interval of one hour. The radiation loss thus measured as average of two tests was 14 BTU/°F.hr. Hence Radiation "A" = 14 x (Temperature of liquid − Ambient temperature).

9. Radiation "B" signifies the convection and radiation heat loss from the steam jackets and the steam lines feeding them. It was determined in a similar manner as Radiation "A". In the case of the first effect jacket including the steam line from the boiler, this heat loss was found to be 30 BTU/°F.hr. For the second and third effect jackets and steam lines this figure was 13 BTU/°F.hr.

10. Disagreement between the totals for heat entering and heat leaving individual effects are in part due to ignoring the changes in liquid densities and boiling points, with temperature and solids content. The significance of these and other possible errors are discussed in Chapter IV.

11. Calculated as : Total heat transmitted through heated surface <u>less</u> Heat in condensate <u>less</u> Radiation "B"

12. Calculated as : Heat in distillate (collected from the next jacket, or the condenser) x latent heat <u>plus</u> Radiation "A" <u>plus/minus</u> Heat to heat or cool the feed. E.g. for Run 47, first effect, this equals
$22.9 \times 955 + 14\,(236 - 69) + 236 - (190 - 10) \times 72$
$= 21{,}850 + 2{,}340 + 4{,}030 = 28{,}220$ BTU/hr.

13. As there was no reason to consider either 11 or 12 as the more accurate, the average was taken as the best working figure for heat transmitted.

14. Calculated/

14. Calculated as average figure for heat transmitted divided by the heating surface (2.7 sq. ft.) and the nominal temperature difference between steam and liquid.

15. Calculated as the sum of the heat transferred in the individual effects divided by the overall temperature gradient and the heating surface per effect (2.7 sq. ft.)

T E S T (b)

Triple Effect Operation with Stickwater

Run Numbers: 24 and 25 : (Table)

Purpose:	To determine the effect of injecting air into the lower end of the heated tube in the third effect.
Working fluid:	Stickwater prepared by dilution of pilchard concentrate stored at $0^{o}F$
Air injection:	was controlled by means of a needle valve which allowed air to be sucked into the base of the return bend joining the heated tube and the return tube of the third effect of the apparatus. The air flow was maintained at 2 litres per minute, measured at ambient temperature and pressure.
Condition of Plant:	The tubes were cleaned before the start of the control run, but not cleaned between runs.
Velocity measurement:	By cold water injection. See Appendix II.
Calculations:	The same notes regarding the data sheets and heat balance calculations apply as for the previous test.
Conclusion:	The injection of 2 litres of air per minute into the base of the heated tube of the third effect resulted in an increase of the gross U value of 46%. It is noteworthy that the improvement in U values was as great (approximately 100%) in the second as in the third effect, and that the increases in circulation velocity are of a different order.

RUN No. 24 (CONTROL) : READINGS

Temperatures and Pressures

Time	First Effect			Second Effect			Third Effect			
	Jacket		Liquid	Jacket		Liquid	Jacket		Liquid	
	Press psi	Temp ^oF	Temp ^oF	Press psi	Temp ^oF	Temp ^oF	Press in. vac.	Temp ^oF	Press in. vac.	Temp ^oF
3.00	30	275	251	15	248	204	5.0	204	25.5	133
3.30	30	275	251	15	250	202	7.0	201	25.5	126
4.00	30	273	251	15	247	197	9.0	196	25.5	121
4.30	30	272	251	15	247	198	8.0	198	25.5	128
Average Temp ^oF		274	251		248	200		200		127
Average Temp. ^oF Difference		23			48			73		

Condensate Collected

Period	From First Effect Jacket	From Second Effect Jacket	From Third Effect Jacket	From Condenser
3.00 – 3.30	13.55 lbs	8.40 lbs	6.40 lbs	5.8 lbs
3.30 – 4.00	13.40 lbs	6.75 lbs	5.50 lbs	4.9 lbs
4.0 – 4.30	12.75 lbs	7.00 lbs	5.30 lbs	4.9 lbs.
Average distillation rate: lbs/hr	26.5	14.8	11.5	10.4
"Flash": lbs/hr	1.3	0.6	0.2	–
Total	27.8	15.4	11.7	10.4

Material Balance

Weight of Liquid	First Effect	Second Effect	Third Effect
entering :	46	31	19
leaving :	31	19	9

Concentration (TSS% by Refractometer)

Liquid	Raw Stickwater	in First Effect	in Second Effect	in Third Effect
% TSS	9.0	15	23	48

Average Feed Temperature : 190^oF

Average Ambient Temperature : 78^oF

Run No. 24 (contd.)

Velocity Measurements

	First Effect	Second Effect	Third Effect
Timed interval over five feet (seconds)	5.5 5.5 5.4	5.5 5.7 5.8	11.5 10.5 10.7
Average	5.47 secs.	5.67 secs.	10.9 secs.
Reading converted to true velocity in heated tubes by using calibration curve	1.3 ft/sec	1.3 ft/sec	0.5 ft/sec

RUN No. 25 (TEST) : READINGS

Temperatures and Pressures

Time	First Effect			Second Effect			Third Effect			
	Jacket		Liquid	Jacket		Liquid	Jacket		Liquid	
	Press psi	Temp °F	Temp °F	Press psi	Temp °F	Temp °F	Press in vac	Temp °F	Press in vac	Temp °F
11.30	30	275	246	13	244	203	6.0	203	25.5	136
12.10	30	273	247	13	243	204	5.0	205	25.5	134
12.35	29.5	272	244	13	240	201	6.0	201	25.5	127
1.00	30	275	245	12	243	202	6.0	203	25.5	133
Average Temp °F		274	245		242	203		203		132
Average Temp. Difference °F		29			39			71		

Condensate Collected

Period	From First Effect Jacket	From Second Effect Jacket	From Third Effect Jacket	From Condenser
11.30 - 12.00	19.60 lbs	11.5 lbs	10.6 lbs	10.8 lbs
12.00 - 12.30	19.45 lbs	11.6 lbs	10.0 lbs	10.2 lbs
12.30 - 1.00	19.25 lbs	12.0 lbs	10.6 lbs	10.6 lbs
Average Distillation rate: lbs/hr	38.9	23.4	20.8	21.1
"Flash": lbs/hr	1.9	0.8	0.4	-
Total	40.8	24.2	21.2	21.1

Material Balance

Weight of Liquid	First Effect	Second Effect	Third Effect
entering :	80	56	35
leaving :	56	35	14

Run No. 25 (contd.)

Concentration (TSS % by Refractometer)

Liquid	Raw Stickwater	in First Effect	in Second Effect	in Third Effect
%TSS	8.5	12	21	50

Average Feed Temperature : 170°F

Average Ambient Temperature : 78°F

Velocity Measurements

	First Effect	Second Effect	Third Effect
Timed interval over five feet (secs)	3.3 3.5 3.4	4.1 4.2 4.0	4.4 3.6 4.2
Average	3.4 secs.	4.1 secs.	4.1 secs.
Reading converted to true velocity in heated tube by using calibration curve	2.3 ft/sec	1.8 ft/sec.	1.8 ft/sec.

HEAT BALANCE

Run No. 24 (Control)

Heat entering and leaving BTU/hr	First Effect IN	First Effect OUT	Second Effect IN	Second Effect OUT	Third Effect IN	Third Effect OUT
With stickwater feed and discharge	6,800	6,800	6,500	3,200	3,100	900
Transmitted through heated surface	32,700	17,900	17,900	13,400	13,400	11,600
With condensate from jacket		6,700		3,300		2,000
Radiation "A"		2,400		1,700		700
Radiation "B"		5,900		2,200		1,600
Totals	39,500	39,700	24,400	23,600	16,500	16,800
Heat Transmitted: (a) Calculated on Condensate from steam jacket	20,100		12,400		9,800	
(b) Calculated from Condensed Distillate	20,200		11,600		9,900	
Average of (a) and (b)	20,150		12,000		9,850	
Apparent U value BTU/sq.ft.°F.hr.	324		93		50	
Gross U Value BTU/sq.ft.°F.hr	105					

Run 25 - 2 L/min air injected

Heat entering and leaving BTU/hr	First Effect IN	First Effect OUT	Second Effect IN	Second Effect OUT	Third Effect IN	Third Effect OUT
With Stickwater feed and discharge	10,200	11,900	11,400	6,000	5,800	1,400
Transmitted through heated surface	47,900	28,100	28,100	24,300	24,300	23,600
With condensate from jacket		9,900		5,100		3,600
Radiation "A"		2,300		1,700		800
Radiation "B"		5,300		2,100		1,600
Totals	58,100	57,500	39,500	39,200	30,100	31,000
Heat Transmitted: (a) Calculated on Condensate from steam jacket	32,700		20,900		19,100	
(b) Calculated on Condensed Distillate	32,100		20,200		19,800	
Average of (a) and (b)	32,400		20,500		19,400	
Apparent U value BTU/sq.ft.°F.hr.	414		195		101	
Gross U value BTU/sq.ft.°F.hr.	153					

HEAT BALANCE

Run No. 24 (Control)

Heat entering and leaving BTU/hr	First Effect		Second Effect		Third Effect	
	IN	OUT	IN	OUT	IN	OUT
With stickwater feed and discharge	6,800	6,800	6,500	3,200	3,100	900
Transmitted through heated surface	32,700	17,900	17,900	13,400	13,400	11,600
With condensate from jacket		6,700		3,300		2,000
Radiation "A"		2,400		1,700		700
Radiation "B"		5,900		2,200		1,600
Totals	39,500	39,700	24,400	23,600	16,500	16,800
Heat Transmitted: (a) Calculated on Condensate from steam jacket	20,100		12,400		9,800	
(b) Calculated from Condensed Distillate	20,200		11,600		9,900	
Average of (a) and (b)	20,150		12,000		9,850	
Apparent U value BTU/sq.ft. °F.hr.	324		93		50	
Gross U Value BTU/sq.ft.°F.hr	105					

Run 25 - 2 L/min air injected

Heat entering and leaving BTU/hr	First Effect		Second Effect		Third Effect	
	IN	OUT	IN	OUT	IN	OUT
With Stickwater feed and discharge	10,200	11,900	11,400	6,000	5,800	1,400
Transmitted through heated surface	47,900	28,100	28,100	24,300	24,300	23,600
With condensate from jacket		9,900		5,100		3,600
Radiation "A"		2,300		1,700		800
Radiation "B"		5,300		2,100		1,600
Totals	58,100	57,500	39,500	39,200	30,100	31,000
Heat Transmitted: (a) Calculated on Condensate from steam jacket	32,700		20,900		19,100	
(b) Calculated on Condensed Distillate	32,100		20,200		19,800	
Average of (a) and (b)	32,400		20,500		19,400	
Apparent U value BTU/sq.ft. °F.hr.	414		195		101	
Gross U value BTU/sq.ft.°F.hr.	153					

T E S T (c)

Single Effect Operation with Stickwater Concentrate

Run Nos. 63 and 106 (Table)

Purpose :	To determine the effect of injecting air into the lower end of the heated tube.
Apparatus:	The improved assembly of heated tube and steam jacket described in Chapter IV was used; in addition to providing a sharper division of heated and unheated tube sections, this also permitted the average tube wall temperature to be measured.
Working Fluid:	Stickwater concentrate processed at Hout Bay Canning Co., 16/4/1962. T.S.S. (Refractometer) 63%. Had been used for earlier tests at 63% and 40% TSS (thinned down with water) Reconcentrated to 63% for tests 63 and 106 in laboratory evaporator.
Air Injection:	Air was drawn into the bottom of the return (heated) tube via a Rotameter and a $\frac{1}{8}$ in. diameter injection nozzle. The air flow was controlled by means of a clip on the rubber air hose. The air flow was maintained at 5 litres per minute, measured at ambient temperature and pressure.
Condition of Plant:	The plant had been operated for about 8 hours without cleaning before runs 63 and 106 were made.
Conclusion:	Air injection at the rate of 5 litres per minute resulted in an improvement in the average film coefficient of heat transfer of 30%. This was achieved in spite of a high (62.5%) solids content of the concentrate.

RUN No. 63 (CONTROL) : READINGS

Temperatures

Time from start of Run: Mins.		0	10	20	30	40	Averages
Liquid Temperature :	Top	2.02	1.95	1.94	1.96	1.97	1.94 millivolts
" "	Bottom	1.97	1.90	1.90	1.91	1.91	= 118°F.
Tube Wall Temperature:	Top	4.22	4.20	4.20	4.21	4.20	4.21 millivolts
" " "	Upper half	4.23	4.22	4.22	4.22	4.21	= 210°F.
" " "	Lower half	4.22	4.20	4.20	4.21	4.20	
" " "	Bottom	4.22	4.20	4.20	4.21	4.19	
Steam Temperature:	Top	4.32	4.30	4.30	4.31	4.30	4.31 millivolts
" "	Bottom	4.32	4.30	4.30	4.31	4.30	= 213°F.

Temperature Difference : Tube Wall – Liquid : 92°F.

Average Feed Temperature : 117°F. Feed Heat Gain or Loss: BTU/Hr. : Neg.

Average Ambient Temperature : 72°F. Radiation Loss : BTU/Hr. : 18 x 46 = 800

Miscellaneous

Vacuum : 27.25 in. Hg.; Barometer : 29.81 in. Hg. ; Plenum Pressure : 1.26 psi

TSS of Concentrate (By Refractometer) : 62.5%

Air Injection : Nil – Control

Condensate Collected

Time from Start of Run: Mins	0	10	20	30	40	Average L/Hr.	Average Lbs/Hr.
Litres	3.0	5.6	8.2	10.9	13.6	15.9	35.1
Litres/10 min. interval	0	2.6	2.6	2.7	2.7		

Heat of Evaporation : BTU/Hr. : 1025 x 35.1 = 35,900

Total Heat Transmitted : BTU/Hr. : 35,900 + 800 = 36,700.

$$h_a = \frac{36,700}{2.88 \times 92} = 138 \text{ BTU/sq.ft. °F. Hr.}$$

RUN No. 106 (TEST) : READINGS

Temperatures

Time from Start of Run: Mins		0	10	20	30	Averages
Liquid Temperature :	Top	1.94	1.91	1.93	1.94	1.90 millivolts
" "	Bottom	1.86	1.86	1.88	1.88	= 116°F
Tube Wall Temperature :	Top	4.20	4.18	4.17	4.17	4.17 millivolts
" " "	Upper half	4.18	4.18	4.17	4.17	= 208°F
" " "	Lower half	4.18	4.17	4.17	4.17	
" " "	Bottom	4.18	4.17	4.17	4.16	
Steam Temperature :	Top	4.29	4.28	4.29	4.28	4.29 millivolts
" "	Bottom	4.29	4.28	4.29	4.28	= 213°F

Temperature Difference : Tube Wall – Liquid : 92°F

Average Feed Temperature : 115°F. Feed Heat Gain or Loss : BTU/Hr.: Neg.
Average Ambient Temperature : 68°F. Radiation Loss : BTU/Hr.: 18 x 48 = 850

Miscellaneous

Vacuum : 27.25 in. Hg.; Barometer : 29.80 in. Hg.; Plenum Pressure : 1.25 psi.
TSS of Concentrate (By Refractometer) : 62.5%
Air Injection : 5.0 Litre/min.

Condensate Collected

Time from Start of Run : Mins.	0	10	20	30	Average L/Hr.	Average Lbs/hr.
Litres	2.4	5.6	9.0	12.7	20.6	45.5
Litres/10 min. Interval	0	3.2	3.4	3.7		

Heat of Evaporation : BTU/Hr. : 1027 x 45.5 = 46,750
Total Heat Transmitted : BTU/Hr.: 46,750 + 850 = 47,600.

$$h_s = \frac{47,600}{2.88 \times 92} = 180 \text{ BTU/sq.ft. } ^\circ\text{F. Hr.}$$

T E S T (d)

Single Effect Operation with Water

Run Nos. 39, 161 and 40 (Table XVII)

Purpose : To determine (a) The effect of throttling the flow of the circulating liquid
(b) The reproducibility of tests carried out with the improved apparatus

Apparatus : The tube wall thermocouples were electronically isolated from each other as illustrated in Fig. 8, Chapter IV, and true tube wall temperatures could thus be measured.

Throttling : The flow was throttled by shutting the valve in the return line, and feeding liquid into the bottom of the rising section via a $\frac{1}{8}$ in. gear pump driven by a varispeed D.C. motor. The throttled flow was equivalent to a liquid velocity at the inlet of the heated tube of 0.1 ft/sec.

NOTE: The liquid velocity during the control run was not measured; referring however to Runs 111 and 112 (Table XV) it would be expected to have been of the order of 1.5 ft/sec.

Condition of Apparatus : The tube was cleaned before the test was started.

Conclusions: (a): Throttling the circulation rate to 0.1 ft/sec. (as compared with an estimated natural circulation rate of 1.5 ft/sec.) increased the film coefficient of heat transfer by about 21%.

(b): The reproducibility of successive runs appears to be of the order of 1%.

(c): The rise and fall of the tube wall temperature along its length suggests that during natural circulation, boiling was confined to the upper half of the tube, whereas with throttled circulation it probably started close to the tube inlet.

RUN No. 39 – THROTTLED FLOW – WATER (CONTROL) READINGS

Temperatures

Time from Start of Run : Minutes		0	10	20	30	Average
Liquid Temperature	Top	3.34	3.36	3.34	3.36	3.35 millivolts = 175.5°F
" "	Bottom	3.34	3.34	3.34	3.34	3.34 millivolts = 175°F
Average temperature of liquid entering heated tube : 3.34 millivolts = 175°F						
Tube Wall Temperature	Top	4.20	4.20	4.18	4.18	4.19 millivolts = 209°F
" " "	Upper half	4.25	4.25	4.25	4.25	4.25 millivolts = 211.5°F
" " "	Lower half	4.22	4.24	4.23	4.23	4.23 millivolts = 210.5°F
" " "	Bottom	4.00	3.98	3.99	4.00	3.99 millivolts = 201°F
Average Tube Wall Temperature : 4.16 millivolts = 208°F						
Steam Temperature	Top	4.32	4.32	4.32	4.31	4.32 millivolts
" "	Bottom	4.32	4.32	4.32	4.31	= 214°F

Temperature Difference : Tube Wall – Liquid entering Heated Tube : ΔT = 33°F

Average Feed Temperature : 180°F ; Feed Heat Gain or Loss: –(38.4 x 5) = –200 BTU/hr

Average Ambient Temperature : 74°F; Radiation Loss: +(18 x 101) = 1,800 BTU/hr

Pressures

Vacuum : 15.95 in. Hg.; Barometer : 29.88 in. Hg.;

Absolute Plenum Pressure : 6.84 psi

Condensate Collected

Time from Start of Run : Minutes	0	10	20	30	Average Litre/hr.	Average Lbs./hr.
Litres	2.0	4.9	7.8	10.7	17.4	38.4
Litres/10 min. interval	0	2.9	2.9	2.9		

Heat of Evaporation : (994 x 38.4) = 38,200 BTU/hr.

Total Heat Transmitted : (38,200 + 1,800 – 200) = 39,800 BTU/hr.

$$h_a = \frac{39,800}{2.88 \times 33} = 418 \text{ BTU/sq.ft. } ^\circ\text{F. Hr.}$$

RUN No. 39 – THROTTLED FLOW – WATER (CONTROL) READINGS

Temperatures

Time from Start of Run : Minutes		0	10	20	30	Average
Liquid Temperature	Top	3.34	3.36	3.34	3.36	3.35 millivolts = 175.5°F
" "	Bottom	3.34	3.34	3.34	3.34	3.34 millivolts = 175°F
Average temperature of liquid entering heated tube : 3.34 millivolts = 175°F						
Tube Wall Temperature	Top	4.20	4.20	4.18	4.18	4.19 millivolts = 209°F
" " "	Upper half	4.25	4.25	4.25	4.25	4.25 millivolts = 211.5°F
" " "	Lower half	4.22	4.24	4.23	4.23	4.23 millivolts = 210.5°F
" " "	Bottom	4.00	3.98	3.99	4.00	3.99 millivolts = 201°F
Average Tube Wall Temperature : 4.16 millivolts = 208°F						
Steam Temperature	Top	4.32	4.32	4.32	4.31	4.32 millivolts
" "	Bottom	4.32	4.32	4.32	4.31	= 214°F

Temperature Difference : Tube Wall – Liquid entering Heated Tube : ΔT = 33°F

Average Feed Temperature : 180°F ; Feed Heat Gain or Loss: –(38.4 x 5) = –200 BTU/hr
Average Ambient Temperature : 74°F; Radiation Loss: +(18 x 101) = 1,800 BTU/hr

Pressures

Vacuum : 15.95 in. Hg.; Barometer : 29.88 in. Hg.;
Absolute Plenum Pressure : 6.84 psi

Condensate Collected

Time from Start of Run : Minutes	0	10	20	30	Average Litre/hr.	Average Lbs./hr.
Litres	2.0	4.9	7.8	10.7	17.4	38.4
Litres/10 min. interval	0	2.9	2.9	2.9		

Heat of Evaporation : (994 x 38.4) = 38,200 BTU/hr.

Total Heat Transmitted : (38,200 + 1,800 – 200) = 39,800 BTU/hr.

$$h_a = \frac{39,800}{2.88 \times 33} = 418 \text{ BTU/sq.ft. °F. Hr.}$$

RUN No. 161 – THROTTLED FLOW – WATER (TEST) READINGS

Temperatures

Time from Start of Run : Minutes		0	10	20	30	Average
Liquid Temperature	Top	3.33	3.33	3.33	3.33	3.33 millivolts = 175°F
" "	Bottom	3.28	3.27	3.27	3.26	3.27 millivolts = 172°F
Average temperature of liquid entering Heated Tube : 3.27 millivolts = 172°F						
Tube Wall Temperature :	Top	4.17	4.17	4.17	4.18	4.17 millivolts = 208°F
" " "	Upper half	4.18	4.18	4.18	4.19	4.18 millivolts = 208.5°F
" " "	Lower half	4.17	4.17	4.17	4.18	4.17 millivolts = 208°F
" " "	Bottom	4.16	4.16	4.16	4.17	4.16 millivolts = 208°F
Average Tube Wall Temperature : 4.17 millivolts = 208°F						
Steam Temperature :	Top	4.30	4.30	4.31	4.30	4.30 millivolts
" "	Bottom	4.30	4.30	4.31	4.30	= 213°F

Temperature Difference : Tube Wall – Liquid entering Heated Tube : ΔT = 36°F

Average Feed Temperature : 180°F ; Feed Heat Gain or Loss: –(50.8 x 6) = –300 BTU/hr.

Average Ambient Temperature: 74°F; Radiation Loss: + (18 x 100) = 1,800 BTU/hr.

Pressures, etc.

Vacuum : 15.95 in. Hg. ; Barometer : 29.86 in. Hg.;

Absolute Plenum Pressure : 6.82 psi; Throttling Pump Speed : ± 100 rpm

Liquid Velocity in 35 mm Tube (from Pump calibration) : 0.1 ft/sec.

Condensate Collected

Time from Start of Run : Mins	0	10	20	30	Average Litre/hr.	Average Lbs./hr.
Litres	16.5	20.3	24.2	28.0	23.0	50.8
Litres/10 min. interval	0	3.8	3.9	3.8		

Heat of Evaporation : (994 x 50.8) = 50,500 BTU/Hr.

Total Heat Transmitted : (50,500 + 1,800 – 300) = 52,000 BTU/Hr.

$$h_s = \frac{52,000}{2.88 \times 36} = 502 \text{ BTU/sq. ft. }^\circ\text{F. Hr.}$$

$\therefore$ % improvement over average control = $\frac{86}{416} \times 100$ = ± 21%

RUN No. 40 - THROTTLED FLOW - WATER -(CONTROL) READINGS

Temperatures

Time from Start of Run : Minutes		0	10	20	30	40	Average
Liquid Temperature	Top	3.35	3.35	3.33	3.32	3.32	3.34 = 175°F
" "	Bottom	3.33	3.33	3.32	3.31	3.31	3.32 = 174°F
Average Temperature of liquid entering Heated Tube : 3.32 millivolts = 174°F.							
Tube Wall Temperature :	Top	4.18	4.20	4.20	4.21	4.21	4.20 = 209°F
" " "	Upper half	4.27	4.27	4.29	4.30	4.30	4.29 = 213°F
" " "	Lower half	4.25	4.24	4.25	4.26	4.26	4.25 = 211°F
" " "	Bottom	4.02	4.01	4.00	4.03	4.01	4.01 = 201.5°F
Average Tube Wall Temperature : 4.19 millivolts = 209°F							
Steam Temperature :	Top	4.32	4.32	4.32	4.32	4.32	4.32 = 214°F
" "	Bottom	4.32	4.32	4.32	4.32	4.32	

Temperature Difference : Tube Wall - Liquid entering Heated Tube : $\Delta T = 35^{\circ}F$

Average Feed Temperature : 179°F; Feed Heat Gain or Loss: -(40.5 x 5) = -200 BTU/Hr.

Average Ambient Temperature: 74°F; Radiation Loss: +(18 x 100) = 1,800 BTU/Hr.

Pressures

Vacuum : 15.95 in. Hg.; Barometer : 29.84 in. Hg.

Absolute Plenum Pressure : 6.80 psi.

Condensate Collected

Time from Start of Run : Mins.	0	10	20	30	40	Average Litre/Hr.	Average Lbs/Hr
Litres	1.0	4.0	7.1	10.1	13.2	18.3	40.4
Litres/10 min. interval	0	3.0	3.1	3.0	3.1		

Heat of Evaporation : (994 x 40.4) = 40,100 BTU/Hr.

Total Heat Transmitted : (40,100 + 1,800 - 200) = 41,700 BTU/Hr.

$$h_a = \frac{41,700}{2.88 \times 35} = 414 \text{ BTU/sq.ft. } ^{\circ}\text{F. Hr.}$$

APPENDIX IV

REDUCING THE VISCOSITY OF STICKWATER BY ENZYME TREATMENT

McBride et al (1) have reported on the remarkable reduction in the viscosity of herring solubles that can be effected by treatment with certain enzyme preparations. Fig. A.IV (1), extracted from a publication by the Fishing Industry Research Institute (2), illustrates that similar effects can be obtained with pilchard and maasbanker concentrate.

Different percentages of commercial enzyme preparations were added to samples of stickwater obtained from a local factory. Earlier tests at F.I.R.I. (3) had indicated that homogenized pilchard viscera was also a powerful hydrolyzing agent. This material, freshly frozen and stored at $0^{o}F$, was therefore also included in the list of preparations that were tested.

The temperature of each sample of stickwater was maintained at approximately $130^{o}F$ for one hour after the enzyme had been added. The viscosity of the liquid was determined before and after treatment by standard U-tube, as described in Appendix I. The percentage of enzyme added was calculated on the dry matter in the stickwater, as determined by refractometer. The moisture content of pilchard gut was determined and found to be roughly 75%. The percentage of dry pilchard gut added was therefore taken as one quarter of its wet weight.

The following conclusions can be drawn from an inspection of Fig. A.IV (1) :

a) variations in the viscosity reduction of stickwater by enzyme treatment may cover the entire range from 90% to 10%;
b) the treatment is about twice as effective with maasbanker stickwater as with pilchard stickwater;
c) there is little to choose between individual commercial enzyme preparations. The choice would primarily be one of cost; and
d) homogenized pilchard gut can be quite as effective as any of the commercial preparations, when compared on a dry basis.

The considerable variation in the effectiveness of enzyme treatment is probably due to the same factors which were responsible in the case of the herring solubles (1), i.e. seasonal variations in the composition of the raw fish; differences in the potency of the additives from batch to batch (particularly in the case of pilchard gut); and pre-history of the raw fish between catch and process.

The cost of the most popular commercial enzyme preparation is

currently/

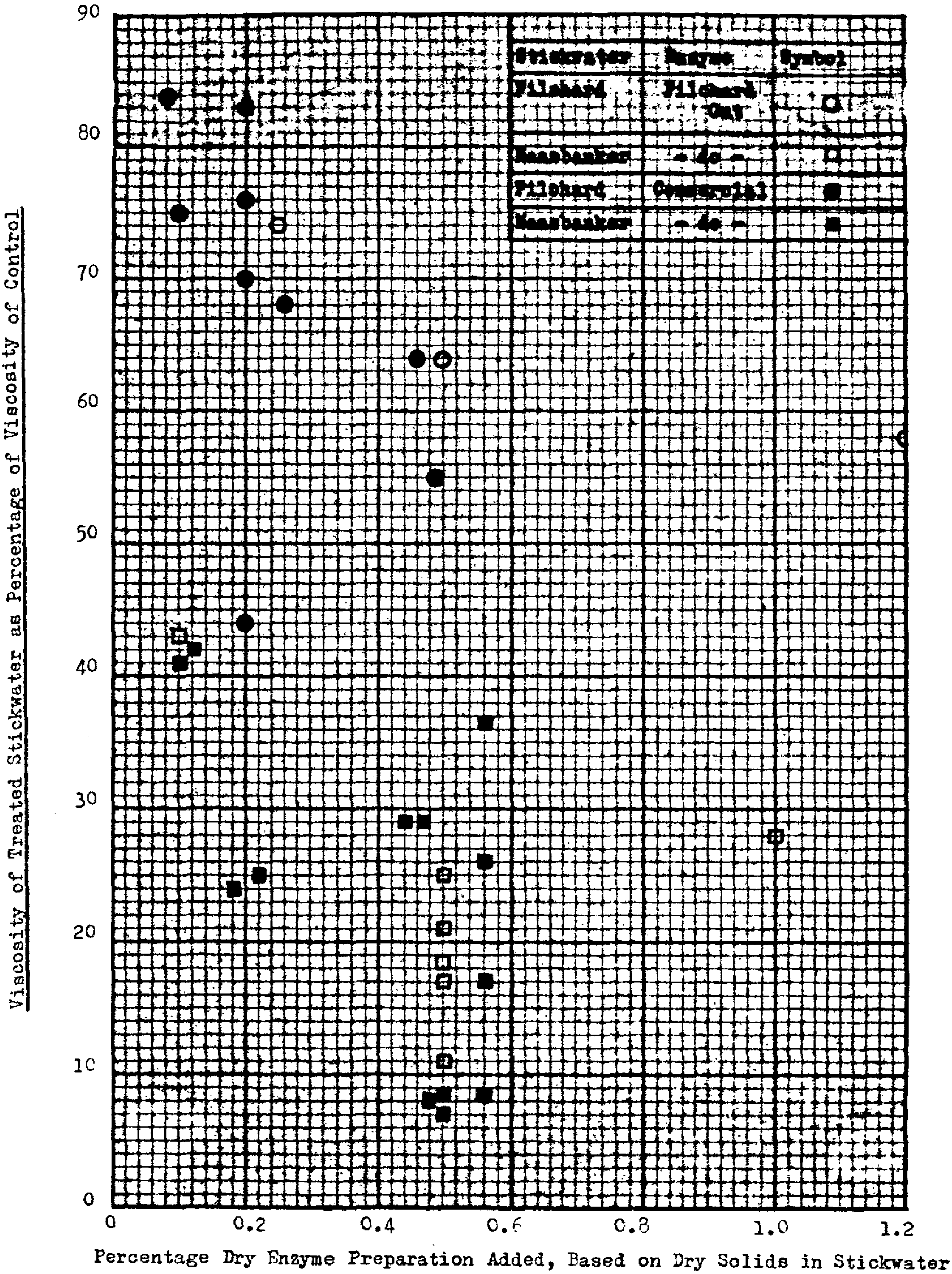

FIG. A.IV (1) : REDUCTION OF VISCOSITY OF PILCHARD AND MAASBANKER STICKWATER BY ENZYME TREATMENT.

currently 40 cents per pound. If added in the proportion of ½% of the dry matter in the stickwater, its use would therefore add about R2 to the price of a ton of dried stickwater solubles, or say 50 cents to the price of a ton of full meal.

Treatment of stickwater with unpreserved pilchard gut involves only the cost of homogenisation. A homogeniser large enough to keep up with the rate at which fish are eviscerated in a large cannery would have to have a capacity of about one ton per hour, would require about 25 Horse Power and cost say R2,000. It is estimated that processing the gut with a machine of this size would cost not more than about R2 per ton or say 0.1 cents per pound. The addition of say 5% of wet gut to the stickwater based on its dry matter content would therefore increase its price by only 10 cents per ton, which is virtually negligible.

Adequate supplies of fresh pilchard gut are readily available in certain localities e.g. at Walvis Bay. In 1960 a total of 345,000 short tons of pilchards were landed at this fishing centre. Of this, about 95,000 short tons was canned. All canned fish is degutted. The average weight of pilchard gut is 10% of the whole fish (4); hence a total of roughly 9,500 short tons of gut would be available to treat the stickwater from 250,000 short tons of fish converted into fish meal.

The surplus gut from each factory could be preserved by freezing. It is estimated that freezing might add a maximum of 0.15 cents, and storage at 0°F for say 3 months might add another 0.3 cents per pound to the cost of the material. Thus, at a maximum cost of about 2 cents per pound on a dry basis, pilchard gut remains vastly more economical to use than any of the commercial preparations.

In most factories the stickwater leaves the centrifuges at about 180°F, which is too hot for enzyme treatment. Current practice when treating maasbanker stickwater is to allow the stickwater to cool to the optimum temperature (130°F to 150°F), which involves a slight loss in thermal efficiency. This can be avoided by feeding the enzyme into that effect of the plant which operates within the optimum temperature range. If this is done the earlier effects do not benefit from the treatment, but this disadvantage may be outweighed by other considerations as discussed in Chapter VIII.

Appendix IV

REFERENCES

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3. Georgala, D.L., "Reduction of Viscosity of Concentrate by Fish Gut Enzymes", Twelfth Annual Report, Fishing Industry Research Institute, University of Cape Town, Rondebosch (1958) p. 56

4. Rowan, A.N., and Stander, G.H., "Salt Content of Fresh Pilchards", Ninth Annual Report, Fishing Industry Research Institute, University of Cape Town, Rondebosch (1955) p. 20

APPENDIX V

THE PRESSURE DROP IN THE BOILING SECTION OF EVAPORATOR TUBES

In Chapter VII the following expression was derived for the loss of head over a finite length of tube :

$$-V_L \frac{\Delta P}{\Delta L} = \frac{1 + Rm}{1 + RV} + \frac{1}{1 + RV}\left(\frac{\Delta H}{\Delta L}\right)_{FL} + \left(\frac{V_L \Delta V_L}{g} + Rm\,\frac{V_G \Delta V_G}{g}\right)\frac{1}{(1 + RV)dL}$$

To illustrate the use of this expression under practical conditions, the same assumptions are made as for the worked example in Chapter VII :

Tube diameter : 0.115 ft.

Density of liquid : 70 lbs/cubic ft.

Specific volume of vapour : 200 cubic ft./lb.

Maximum and minimum viscosity of liquid : 500 cp and 1 cp.

Assume also that vapour is being generated at a typical rate of 50 lbs/hr., and that the liquid velocity in the non-boiling section is 2.0 ft/sec. at the lower, and 0.4 ft/sec. at the higher viscosity. The corresponding liquid flow rates are 1.45 lbs/sec. and 0.29 lbs./sec.

Consider first the hydrostatic term at the maximum flow rate :

	At Tube Outlet	At Midpoint of Boiling Section	1/10 of Boiling Section	1/20 of Boiling Section
w_G, lbs/sec.	0.014	0.007	0.0014	0.0007
w_L, lbs/sec.	1.45	1.45	1.45	1.45
$Rm = \frac{w_G}{w_L}$	0.0097	0.0048	0.00097	0.00048
$RV = Rm\,\frac{V_G}{V_L}$	136	68	13.6	6.8
$\frac{1 + Rm}{1 + RV}$	0.0073	0.0145	0.069	0.128

These values of $\frac{1 + Rm}{1 + RV}$ have been plotted as shown in Fig. A.V. (1) and it will be noted that :

a) except at the very commencement of the boiling section the term is small; i.e. over practically the whole extent of the boiling section the density of the liquid-vapour mixture is much less than that of the liquid only ;

b) even major changes in the assumed flow rates w_L and w_G cannot significantly alter this condition.

The calculation of $\frac{1 + Rm}{1 + RV}$ for the reduced flow rate of 0.29 lbs/sec. if therefore omitted.

Consider/

Consider next the Friction Loss term $\frac{1}{1 + RV}\left(\frac{\Delta H}{\Delta L}\right)_{FL}$

A useful correlation of data on the pressure drop due to friction for two-phase systems flowing in pipes is due to Lockhart and Martinelli (1). Their findings are presented as sets of curves, as reproduced by McAdams (2), linking the two key parameters.

$$\Phi = \sqrt{\frac{\left(\frac{\Delta P}{\Delta L}\right)_{TP}}{\left(\frac{\Delta P}{\Delta L}\right)_{L}}} \quad \text{and} \quad X = \sqrt{\frac{\left(\frac{\Delta P}{\Delta L}\right)_{L}}{\left(\frac{\Delta P}{\Delta L}\right)_{G}}}$$

where $\left(\frac{\Delta P}{\Delta L}\right)_G$ = Pressure drop due to vapour imagined flowing by itself. lb/(sq.ft.) (ft.)

$\left(\frac{\Delta P}{\Delta L}\right)_L$ = Pressure drop due to liquid imagined flowing by itself. lb/(sq.ft.) (ft.)

$\left(\frac{\Delta P}{\Delta L}\right)_{TP}$ = Pressure drop for two-phase flow. lb/(sq.ft.) (ft.)

The term $\left(\frac{\Delta P}{\Delta L}\right)_{TP}$ as defined by Lockhart and Martinelli incorporates specific flow conditions, and is synonymous with $\frac{1}{1 + RV}\left(\frac{\Delta H}{\Delta L}\right)_{FL} \times \rho L$

If, therefore, representative values for $\left(\frac{\Delta P}{\Delta L}\right)_L$ and $\left(\frac{\Delta P}{\Delta L}\right)_G$ can be derived, reference to Lockhart and Martinelli's curves will enable the corresponding values of $\left(\frac{\Delta P}{\Delta L}\right)_{TP}$ to be estimated, and hence $\frac{1}{1 + RV}\left(\frac{\Delta H}{\Delta L}\right)_{TP}$

The point values of these parameters at the tube outlet, the mid-point and the commencement of the boiling section will be considered.

At the lower viscosity level and with a flow rate of 1.45 lb/sec., the Reynolds Number of the liquid stream imagined flowing by itself is :

$$\frac{2 \times 0.115 \times 70 \times 10^4}{1.0 \times 6.72} = 24{,}000$$

For a smooth tube surface the corresponding Friction Factor is 0.006.

Hence $\left(\frac{\Delta P}{\Delta L}\right)_L = \frac{4 \times 0.006 \times 2^2 \times 70}{0.115 \times 2 \times 32} = 0.91$ lb/(sq.ft.) (ft.)

At the tube outlet the vapour velocity $= \frac{50 \times 200}{0.0104 \times 3{,}600} = 270$ ft/sec

At 120°F the viscosity of the vapour is about 0.01 cp.

The Reynolds Number for the vapour stream imagined flowing by itself is

$$\frac{270 \times 0.115 \times 1 \times 10^4}{0.01 \times 6.72 \times 200} = 23{,}000$$

The corresponding Friction Factor is 0.006, and

$$\left(\frac{\Delta P}{\Delta L}\right)_G = \frac{4 \times 0.006 \times 270^2 \times 1}{0.115 \times 2 \times 32 \times 200} = 1.19 \text{ lb/(sq.ft.) (ft.)}$$

Hence $X = \sqrt{\frac{\left(\frac{\Delta P}{\Delta L}\right)_{L}}{\left(\frac{\Delta P}{\Delta L}\right)_{G}}} = \sqrt{\frac{0.91}{1.19}} = 0.87$

Referring to Martinelli's curve for Turbulent-Turbulent flow, the

corresponding/

corresponding $\Phi = 4.5$

and hence $\left(\frac{\Delta P}{\Delta L}\right)_{TP} = 0.91 \times 4.5^2 = 18.5$ lb/(sq.ft.) (ft.)

or $\frac{18.5}{70} =$ 0.26 ft. liquid/ft.

At the mid-point of the boiling section $\left(\frac{\Delta P}{\Delta L}\right)_L = 0.91$ lb/(sq.ft.) (ft.) as before, but the vapour flow rate can be taken as one half that at the tube outlet; i.e. $Re_G = 12,000$

$f_G = 0.0075$

and the superficial vapour velocity = 135 ft/sec.

Therefore $\left(\frac{\Delta P}{\Delta L}\right)_G = \frac{4 \times 0.0075 \times 135^2 \times 1}{0.115 \times 2 \times 32 \times 200} = 0.37$ lb/(sq.ft.) (ft.)

$X = \sqrt{\frac{0.91}{0.37}} = 1.57$

Φ (from Martinelli's curve) $= 3.3$

$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = 0.91 \times 3.3^2 = 9.9$ lb/(sq.ft.) (ft.)

$= \frac{9.9}{70} =$ 0.14 ft. liquid/ft.

At the point at which boiling commences

$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = \left(\frac{\Delta P}{\Delta L}\right)_L = 0.91$ lb/(sq.ft.)(ft)

or $\frac{0.91}{70} =$ 0.013 ft. liquid/ft.

By similar reasoning and calculation, the corresponding figures for a viscosity of 500 cp. and a circulating velocity of 0.4 ft/sec. are :

at tube outlet : 0.65 ft. liquid/ft.
at mid-point : 0.47 ft. liquid/ft.
at origin : 0.145 ft. liquid/ft.

These friction loss figures have also been plotted in Fig. A.V (1). It is to be noted that :

a) the effect of the 500:1 variation in viscosity far outweighs the changes within practical limits of any of the other physical properties of the liquid;

b) the friction head loss calculated for a viscosity of 500 cp. and a (purposely high) circulating velocity of 0.4 ft/sec. is close to the maximum that can be conceived for normal operating conditions.

Consider finally the acceleration term $\left(\frac{v_L \Delta v_L}{g} + Rm \frac{v_G \Delta v_G}{g}\right) \frac{1}{(1 + RV)\Delta L}.$

It is possible to evaluate this expression step by step, starting from the commencement of the boiling section. In the limit the resulting curve would be parabolic in shape.

For the purpose of estimating the average head loss, however, it is sufficient to calculate the total acceleration head and to divide by the length of the boiling section.

Lockhart and Martinelli (1) provide another correlation between the

liquid/

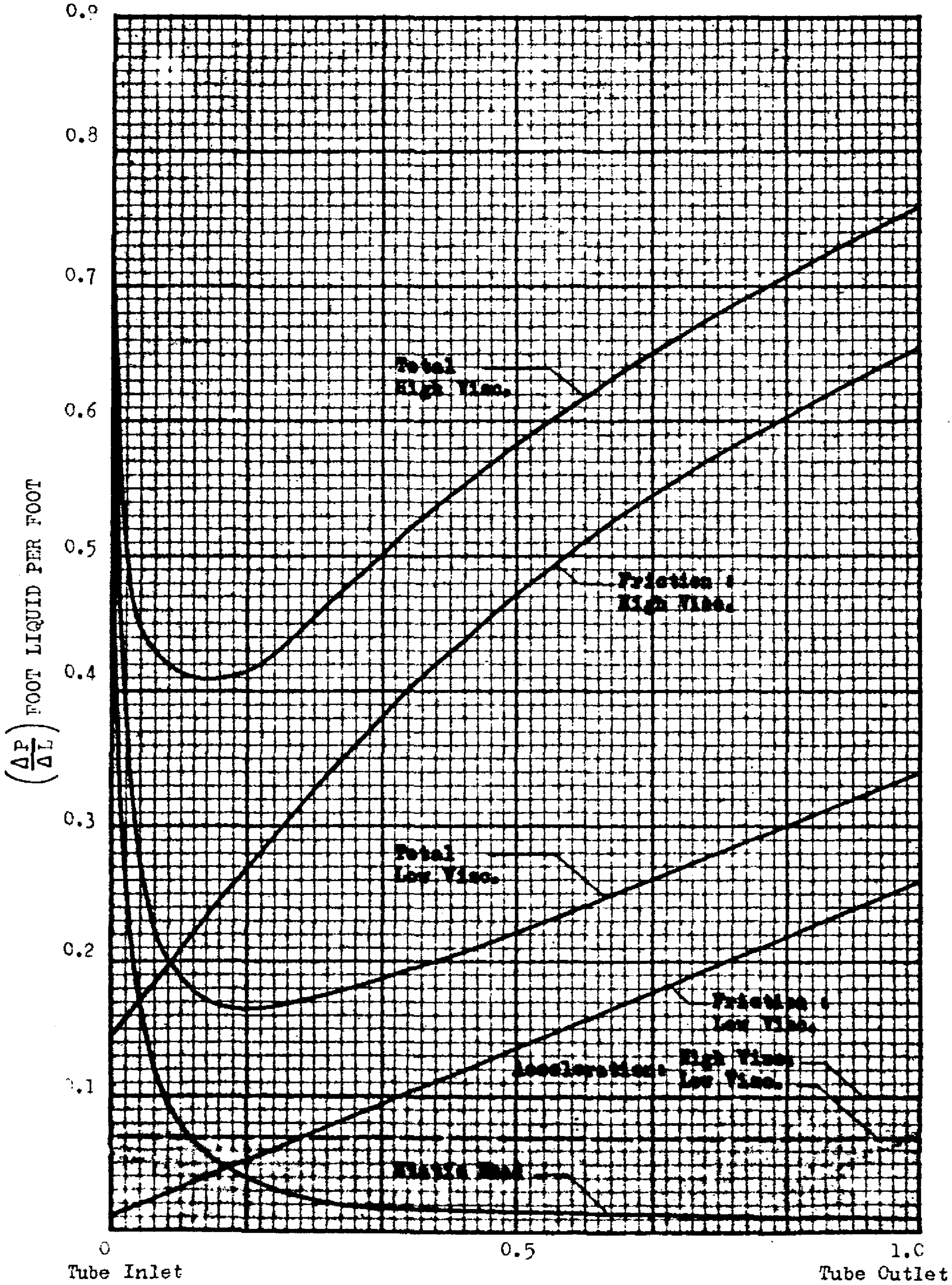

FIG. A.V (1) : ESTIMATED HEAD LOSS IN BOILING SECTION FOR TURBULENT AND VISCOUS FLOW been plotted as the sum of the respective static, friction and acceleration terms. It will be observed that the average total losses are roughly 0.25 and 0.55 ft. of liquid per foot.

Govier et al (4) reported on experiments concerning the pressure drop and hold-up of air-water mixtures flowing vertically upwards in tubes. In a typical case, using a tube of $1\frac{1}{2}$ in. internal diameter and water velocities of 0.869 and 2.5, they measured average pressure drops of about 0.3 and 0.5 ft. per ft. respectively. The agreement of the calculated figures with such experimental data as are available therefore appears reasonable.

However, in view of the conjectural nature of the pressure drop estimated for the high viscosity range, and bearing in mind that Lockhart and Martinelli's correlation was developed for horizontal rather than for vertical flow, it appears wise to allow for a generous margin of error. Even so, it seems reasonably safe to assume that the average pressure drop in the boiling section of industrial evaporators can hardly be less than 0.1 or more than 0.9 ft. liquid per foot.

This average pressure drop may be considered as a fraction "a_2" of the hydrostatic head.

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NOMENCLATURE

A	:	Cross Sectional Area of Tube	sq. ft.
a_1, a_2, a_3 ...		Constants	Dimensionless
c	:	Specific Heat	BTU/lb.
D	:	Tube Diameter	ft.
f	:	Friction Factor	Dimensionless
g	:	Acceleration due to gravity	32 ft./sec.^{2}
H	:	Head of Liquid	feet
H_{FL}	:	Friction Head Loss of Liquid	feet
H_{FG}	:	Friction Head Loss of Gas or Vapour	feet
h	:	Coefficient of Heat Transfer between Tube Wall and Bulk of Liquid	BTU/sq.ft. ^oF. Hr.
		h_a Average coefficient over length of Heated Tube	
		h_1 Average coefficient over length of Non-Boiling Tube Section	
		h_2 Average coefficient over length of Boiling Tube Section	
k	:	Thermal Conductivity	BTU.ft/sq.ft. ^oF. Hr.
L	:	Length of Non-Boiling Section of Tube	feet
L_o	:	Total Length of Heated Tube	feet
ΔL	:	Finite Element of Length	ft.
P	:	Pressure	lb./sq. in.
P_H	:	Hydraulic Pressure Drop in Boiling Section	lbs./sq. in.
P_F	:	Friction Pressure Drop in Boiling Section	lbs./sq. in.
P_A	:	Acceleration Pressure Drop in Boiling Section	lbs/sq. in.
ΔP	:	Finite change of Pressure	lbs./sq. in.
q	:	Heat Flow Rate	BTU/hr.
		q_1 Heat Flow Rate in Non-Boiling Section	
		q_2 Heat Flow Rate in Boiling Section	
R_m	:	Vapour : Liquid Mass Ratio	$\frac{W_G}{W_L}$
R_v	:	Vapour : Liquid Volume Ratio	$\frac{W_G}{W_L} \times \frac{V_G}{V_L}$

Nomenclature (contd.)

T	:	Temperature	Deg. F.
	ΔT	Temperature Gradient between Tube Wall and Bulk of Liquid	
	ΔT_o	Temperature Gradient between Steam and Bulk of Liquid	
	δT	Temperature Rise of Liquid in Non-Boiling Tube Section	
t	:	Time	Hours
U	:	Overall Heat Transfer Coefficient between Steam and Bulk of Liquid	BTU/sq.ft. ^{o}F. Hr.
	U_a	Average overall Heat Transfer Coefficient over Length of Heated Tube	
	U_G	Gross overall Heat Transfer Coefficient assuming Single Effect Operation of a Multi-Effect Plant	
V	:	Specific Volume	Cubic ft./lb.
	V_L	Specific Volume of Liquid	
	V_G	Specific Volume of Gas or Vapour	
v	:	Velocity	ft./sec.
	v_L	Velocity of Liquid	
	v_G	Velocity of Gas or Vapour	
W	:	Work performed per Pound of Air	Ft.lbs/lb.
w	:	Mass Flow Rate	lbs/sec.
	w_L	Mass Flow Rate of Liquid	
	w_G	Mass Flow Rate of Gas or Vapour	
γ	:	Ratio of Specific Heats of Air	Dimensionless
ρ	:	Density	lb/cubic ft.
	ρ_L	Density of Liquid	
	ρ_G	Density of Gas or Vapour	
μ	:	Viscosity	Centipoises
	μ_w	Viscosity of Water	
σ	:	Surface Tension	lb./sq.ft.
λ	:	Latent Heat	BTU/lb

Nomenclature (contd.)

T	:	Temperature	Deg. F.
	ΔT	Temperature Gradient between Tube Wall and Bulk of Liquid	
	ΔT_o	Temperature Gradient between Steam and Bulk of Liquid	
	δT	Temperature Rise of Liquid in Non-Boiling Tube Section	
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	U_G	Gross overall Heat Transfer Coefficient assuming Single Effect Operation of a Multi-Effect Plant	
V	:	Specific Volume	Cubic ft./lb.
	V_L	Specific Volume of Liquid	
	V_G	Specific Volume of Gas or Vapour	
v	:	Velocity	ft./sec.
	v_L	Velocity of Liquid	
	v_G	Velocity of Gas or Vapour	
W	:	Work performed per Pound of Air	Ft.lbs/lb.
w	:	Mass Flow Rate	lbs/sec.
	w_L	Mass Flow Rate of Liquid	
	w_G	Mass Flow Rate of Gas or Vapour	
γ	:	Ratio of Specific Heats of Air	Dimensionless
ρ	:	Density	lb/cubic ft.
	ρ_L	Density of Liquid	
	ρ_G	Density of Gas or Vapour	
μ	:	Viscosity	Centipoises
	μ_w	Viscosity of Water	
σ	:	Surface Tension	lb./sq.ft.
λ	:	Latent Heat	BTU/lb

Nomenclature (contd.)

Re	:	Reynolds Number	$\frac{D.v.\rho}{\mu}$
Nu	:	Nusselt Number	$\frac{h.D}{k}$
Pr	:	Prandtl Number	$\frac{c.\mu}{k}$
Gr	:	Graetz Number	$\frac{w.c.}{k.L}$

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